University of Illinois at Urbana-Champaign

ACRC

Air Conditioning and Refrigeration Center   A National Science Foundation/University Cooperative Research Center

Suitability of Polymer Heat Exchangers
for Air Conditioning Applications

T. Malik and C. W. Bullard

ACRC TR-237   March 2005

Abstract

The project compares the non-metallic/polymer heat exchangers to the current technology in metallic heat exchangers. A review of state of the art of polymer heat exchangers is presented and technological problems are identified. Simulation models were developed and used to explore the thermal-hydraulic, packaging and weight tradeoffs associated with polymer heat exchangers. Finally, the results are used to provide essential background for deciding if and where to allocate resources to more detailed analyses and prototype development.

1.1 Introduction and literature review

The conductivity of non-metallic/polymer tubes in heat exchangers is ~1000 times less than their metallic counterparts. But polymer/non-metallic heat exchangers could offer such benefits as design freedom, low thermal expansion, chemical resistance etc. This project analyzes the thermal-hydraulic and weight tradeoffs associated with polymer heat exchangers, to provide essential background for deciding if and where to allocate resources to more detailed analysis and prototype development

A review of the available literature revealed that most of the commercially available polymer heat exchangers are used in a corrosive environment [1] or in low temperature applications (ice storage or solar heating of domestic hot water and swimming pools [2]). The literature available on the University of Minnesota website [3] and teleconferences with Prof. J. Davidson (University of Minnesota), Ms. Diana Salvadori (DuPont, Canada) and Mr. M. Rubio (Fafco) led to a decision to focus on primary surfaces due to the potential complexity and inefficiency of many kinds of integral fin concepts. The teleconferences also called attention to several other applications of polymer heat exchangers which are under development for applications like air-to-air turbochargers for diesel cars in Europe, car radiators and turbine air inlet coolers.

It is clear that the state of the technology of the polymers, which can be used with standard refrigerants like R134A, R22 or R410A, is still in its premature stage because of issues relating to diffusion of the refrigerant and high pressure requirements. Laminated tubes composed of layers impermeable to refrigerant, water and other gases would probably be needed – perhaps similar to those needed to prevent diffusion through vacuum panel insulation where the pressure differentials are actually much smaller. Preliminary tests on these laminated tubes, at the University of Minnesota, have shown that benefits obtained from them don't warrant the complexity and cost of making these. Therefore the analysis in this report considers a R410A vapor compressor system with a flat plate evaporator and condenser and a secondary loop using water as the secondary refrigerant

Detailed analysis of compact water chillers, with focus on charge minimization, has been addressed by Bullard and Barnes [6]. Modeling of these chillers were carried out at ARI 550/590 standard rating conditions (0.054 L/s per kW at 29.4°C inlet condenser water, 0.043 L/s per kW at 6.7°C outlet evaporator water) and the COP_{sys} was found out to be ~4. This project focuses on analysis of the air coils, while assuming a simplified model for the chiller that neglects the pressure drop in its evaporator and condenser.

This review also highlighted the importance of issues like long-term pressure and temperature capabilities; and mechanical issues like fitting and ease of repairs etc. in designing polymer heat exchangers. These issues are out of the scope of this project but they should be kept in mind while analyzing the feasibility of using polymers in heat exchangers.

1

Work at the ACRC has focused on some interesting applications of special polymers, like DuPont's Kapton®, in very thin (<250mm) and extremely lightweight (<0.7 kg/m^2) microchannel heat exchangers [4]. Kapton is renowned for its superior thermal stability, mechanical toughness, high strength, and chemical resistance and thus is very suitable for this application. Kapton heat exchangers are made from thermally bonded laminated sheets. The use of Kapton also allows design flexibility so that the heat exchangers can have many different integral channel, header and manifold configurations which can be optimized for different applications. Because fabrication issues are only in the earliest stages of research, in this particular project we focus on more conventional materials and applications.

The following could be some of the possible future applications for polymer heat exchangers -

1. As refrigerators become more efficient, we can possibly replace the steel skin (wrapper) of refrigerators with structural polymers, to reject condenser heat via natural convection. Water or refrigerant could possibly flow through the polymer (polycarbonate) in these hot wall condensers, if such plates could be designed to have high conductivity perpendicular to surface. Currently available rigid polymers are designed to have high conductivity parallel to the plate to diffuse the heat (e.g. in headers of car radiators) and are thus not suitable for use in refrigerators and building walls. But, in future if structural polymers with high conductivity perpendicular to the surface could be designed then these hot wall condensers could provide lot of design flexibility in rejecting the heat. Because of their dual structural/thermal use these heat exchangers might be much lighter, than the current ones, too.

2. Polymer tubes could be used as water loops, indoors and outdoor (from evaporator and condenser), with ultra-compact chillers having tiny charge of toxic or flammable refrigerant which has zero global warming potential (e.g. hydrocarbons, ammonia). This probably means having tube banks with small diameter tubes, thin walls and creative header design.

3. Similarly for automotive air conditioning, using propane with a water loop inside car could be another application. The same loop could be used for heating as well, using pumped heat or engine waste heat, or a combination. Currently in vans and SUV's it is hard to get oil returned from evaporator in back seat, so water loop might possibly be useful for R134a systems too, if the energy penalty were not too large.

Thus, the range of applications to be targeted, has been narrowed down and so the analyses in this report are focusing on some simple geometries like cross-counterflow tube banks. It is these low pressure applications (where the tube thickness is small) that the performance of polymer tubing can approach that of conventional metallic ones, and use of polymers could be viable. The following results will be expressed in terms of their effects on system efficiency, and the tradeoff with package volume and material requirements.

A comparison of liquid-to-liquid polymer heat exchangers with their metallic counterparts, based mainly on work conducted by Davidson et al., is presented in section 1.2.1 and 1.2.2. In section 1.3, a nylon polymer-to-air heat exchanger, used in secondary loop with a vapor compression system, is simulated and compared to a microchannel condenser.

1.2 Liquid-to-liquid heat exchangers
Two types of liquid-to-liquid heat exchangers have been considered in this section–

1. Tube-in-shell heat exchangers
2. Immersed heat exchangers

1.2.1 Tube-in-shell heat exchangers

Davidson et al. compared a commercial copper counterflow tube-in-shell liquid-to-liquid heat exchanger with its polymer counterpart. Simulation models were developed in EES and critical technical issues associated with using polymer heat exchangers were identified. A brief summary of their work has been presented in the following paragraphs and details can be found in [1].

The arrangement and the number of tubes, shell dimensions, flow rate of the liquids and the required heat rate in an external tube-in-shell heat exchanger were fixed and the length of the tube required was calculated. Two kinds of polymer tubes were analyzed viz. PEX (cross-linked polyethylene), a low strength polymer, and nylon (similar to DuPont tubing), a high strength polymer. The thermal conductivities for PEX and nylon were similar, with PEX being slightly more conductive (~0.38 W/m K) as compared to nylon (~0.31 W/m K). The PEX heat exchanger used a standard PEX tube with outer diameter 9.53 mm with a wall thickness of 1.78 mm, while the nylon heat exchanger used smaller tubes of outer diameter 3.81 mm and only 0.2 mm thick walls.

The analysis highlighted the fact that the thermal resistance of the wall was a dominant limitation of the PEX heat exchanger, while it was not so for the nylon one. The authors noted that at a typical flow rate of 5.7 L/min, the ratios for inside, outside and wall thermal resistance were 24%, 34% and 42% , of the total, respectively for the PEX heat exchanger and 49%, 26% and 25% of the total, respectively for the nylon counterpart. Also, the required lengths of the tube for the nylon heat exchanger were ~75% less than the PEX heat exchanger. This is mostly due to decreased thermal resistance because of thinner walls in nylon heat exchanger.

Further, the analysis of the copper heat exchanger, with tubes of outer diameter 6.35 mm and wall thickness 0.5 mm, showed that its wall thermal resistance was negligible and its performance was limited only by the low inside the tubes, the percents of inside, outside and wall thermal resistances, at typical flow rate of 5.7 L/min, being 76%, 24% and 0.04% respectively.

A comparison of the polymer heat exchanger with copper one is given in Table 1.2.1.1. It is interesting to note that the thin-walled nylon heat exchanger has a very similar performance as the copper heat exchanger.

Table 1.2.1.1 Heat transfer surface areas for tube-in-shell HX at 5.7L/min [1]

Heat transfer capacity	PEX	Nylon	Copper
3000 W	1.78 m^2	0.50 m^2	0.50 m^2
6000 W	7.78 m^2	2.16 m^2	2.19 m^2

1.2.2 Immersed heat exchangers

Similar to the tube-in-shell configuration, an analysis of immersed heat exchangers was also done by Davidson et al. The immersed heat exchanger has a bundle of horizontal tubes laid in a water tank with liquid flowing through the tubes only. There is no flow in the tank, only natural convection.

The outer diameters and wall thicknesses of the tubes for PEX and nylon immersed tube-bank configuration was same as that for the tube-in-shell configuration. The results of the analysis were very similar to the tube-in-shell configuration and have been summarized in Table 1.2.2.1. Here the copper immersed heat exchanger was modeled as a single 15.88 mm outside diameter tube.

Table 1.2.2.1 Heat transfer surface areas for immersed HX [1]

Heat transfer capacity	PEX	Nylon	Copper
3000 W @ 5.7L/min	4.21 m^2	1.89 m^2	1.10 m^2
6000 W @ 11.4 L/min	8.42 m^2	3.78 m^2	2.03 m^2

Again, it is clear from the results that heat transfer areas in nylon and copper heat exchangers were significantly lower than PEX heat exchanger.

Thus in liquid-to-liquid heat exchangers, thin walls in polymer designs are necessary to achieve thermal performance levels comparable to their metallic counterparts. As will be seen below, this is not as important in the case of air-to-liquid heat exchangers where the air side heat transfer resistance is the limiting factor.

1.3 Air-to-liquid heat exchangers

A brief summary of polymer liquid-to-liquid heat exchangers was presented in the previous section. It was noted that, in liquid-to-liquid polymer heat exchangers, thin walls are necessary to achieve thermal performance levels comparable to their metallic counterparts. This section presents an analysis of air-to-liquid heat exchangers to explore the effect of replacing metal tubes by polymer ones, on the heat transfer characteristics and the overall performance of the system.

The analysis in this section considers a R410A vapor compressor system with a flat plate condenser and a secondary loop using water, for heating, as the secondary refrigerant. Also, just to investigate feasibility, we have used the properties of a commonly used material - DuPont's Nylon (Zytel HTN 51G15) [5], for the polymer tubes, because it represents the best case of "k/t" (conductivity/wall thickness) amongst the currently available materials in described in the literature. However in its current form, DuPont's nylon is not suitable for heat exchangers using glycol or water (due to permeability) and has very poor performance in oxidative environments [7]. Thus, nylon represents a design target, of 'k/t', for the future materials even though in its current state, it is unsuitable for use in this kind of heat exchanger application. Laminated nylon tubes could provide the necessary protection against diffusion, but benefits from lamination don't currently warrant the cost and complexities involved. A discussion of some other polymers potentially suitable for use in heat exchangers is presented in Appendix A. The analysis of vapor compression system with a secondary water loop for heating (heating coil) is presented in the following paragraphs.

1.3.1 Air-to-liquid heat exchangers – Analysis of a heating coil

In the heating coil design, water was first heated by the heat rejected from the condenser of the vapor compression system, and then used to heat the air using a polymer heat exchanger. A schematic of the circuiting is shown in Figure 1.3.1.1.
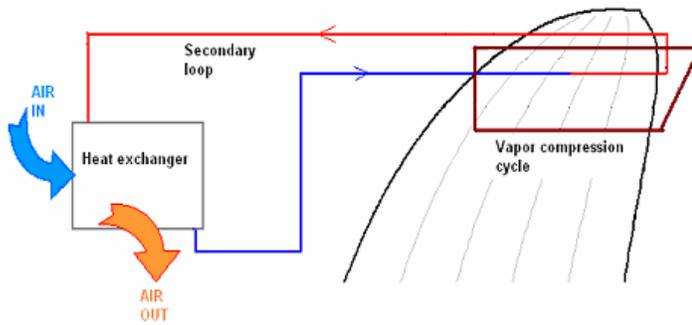
Figure 1.3.1.1 Schematic of the heating loop

The configuration of the polymer heat exchanger is shown in Figure 1.3.1.2 and Figure 1.3.1.3.
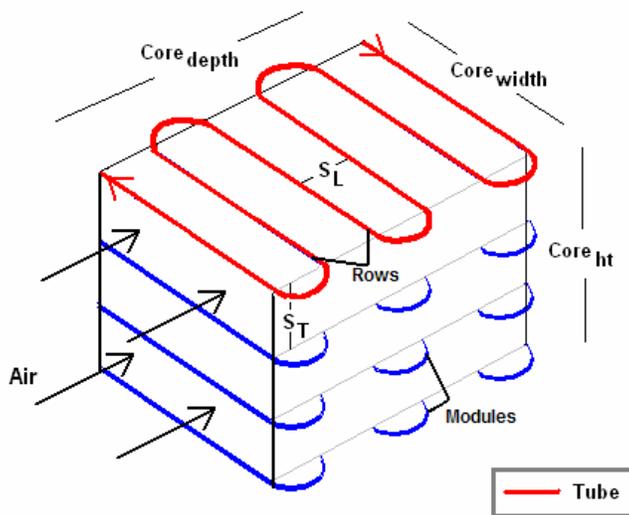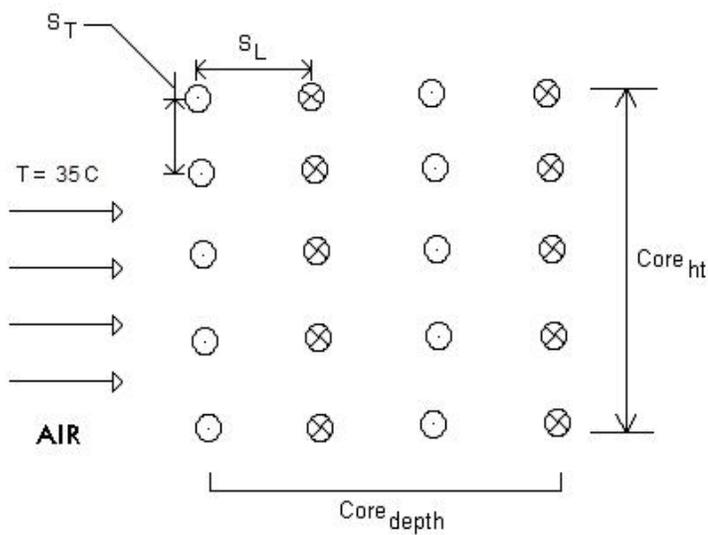

Figure 1.3.1.2 Heat exchanger – 3D view

Figure 1.3.1.3 Heat exchanger - Front view

5

A NXN system model, shown in Figure 1.3.1.3, was developed in EES to explore the design space for an optimal polymer heat exchanger configuration which could then be compared to a typical metallic counterpart. For ease of analysis, the polymer heat exchanger has been modeled as a simple counterflow configuration. The modeling and analysis has been done for a vapor compression a/c system having one ton cooling capacity. Also, the tubes have been assumed to be made of a commonly used material - DuPont's Nylon (Zytel HTN 51G15) [1].

To provide a realistic system context for the optimization, the evaporator capacity, inlet air and water temperatures into the polymer hx, geometry variables like tube diameter, ratio of the outside diameter to the tube thickness (SDR – Standard diametric ratio), ratio of tube diameter to transverse/longitudinal pitch (D_{out}/S_L) etc. were specified and system performance parameters like COP, outlet air and water temperatures etc. were calculated. Initially, the face area was specified as 0.4 m^2, which is a typical value of the face area per ton for metallic heat exchangers. The aspect ratio (AR) was taken as 1, representing a square geometry

The following correlations were used in the model –

HEAT TRANSFER CORRELATIONS

Water side: Gneilinski (1976): $0.6 < Pr_{ref} < 10^5$, $1300 < Re_{ref} < 5 \times 10^6$ [10]

                          $Nu_w = 4.3$; $Re < 1300$

Air Side: Zukauskas (1976): $Re < 2 \times 10^5$ [12]

PRESSURE DROP CORRELATIONS

Water side: Churchill (1977) [11]

Air Side: Zukauskas (1976)

The NXN system model is shown in Figure 1.3.1.4.

A fixed value of the UA_{cond} for the refrigerant-to-water condenser was specified in the model. To calculate UA_{cond}, a nominal heat rejection rate of 3.75 kW and a condensing temperature of 45°C were assumed. Also, the approach temperature difference at the condenser was taken as 2°C (Figure 1.3.1.5) and the inlet temperature of the water into the condenser was specified at 37°C. With the above assumptions, the UA_{cond} was found to be ~ 0.86 kW/C.
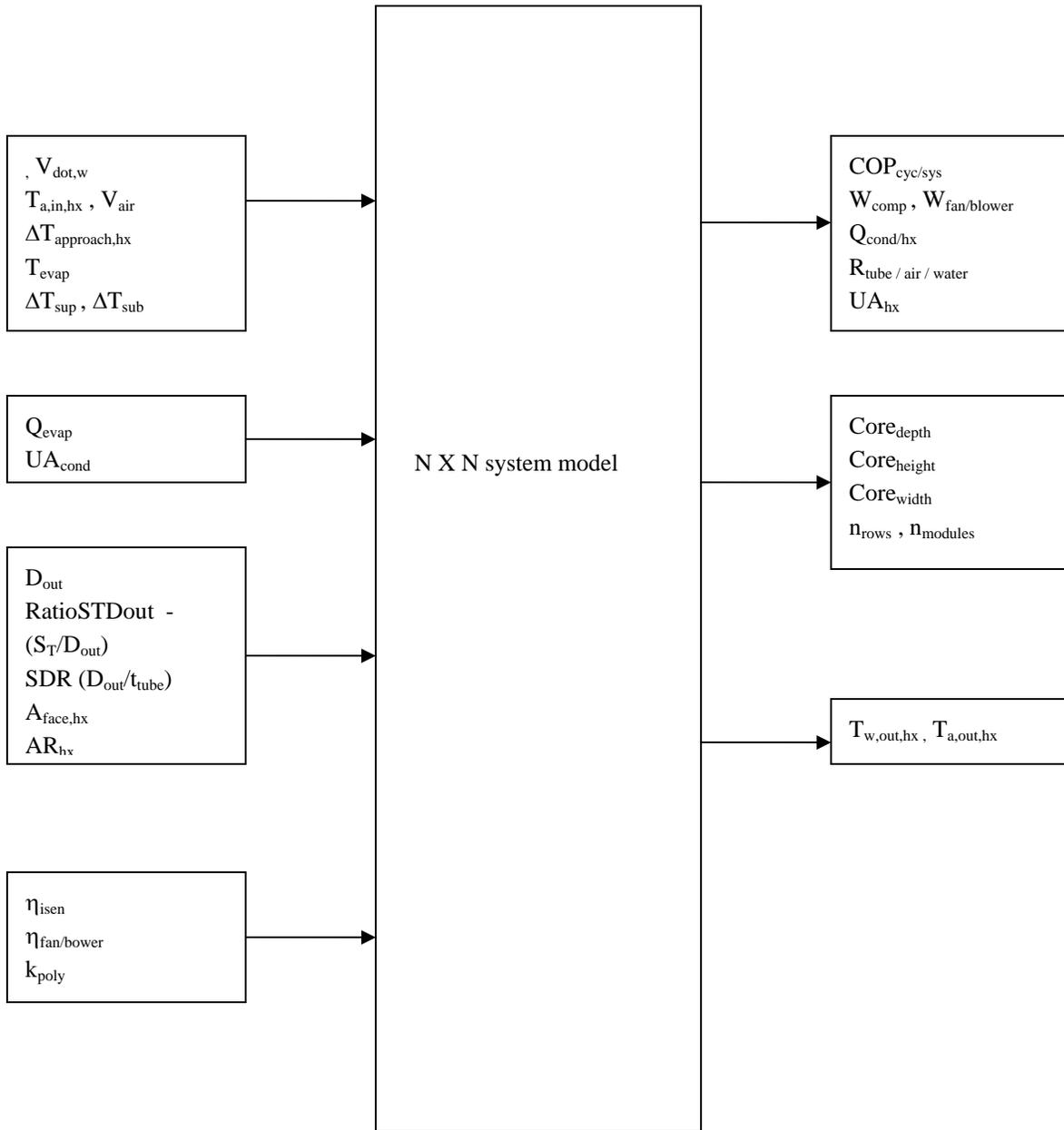
Figure 1.3.1.4 Inputs and outputs in the heat exchanger model
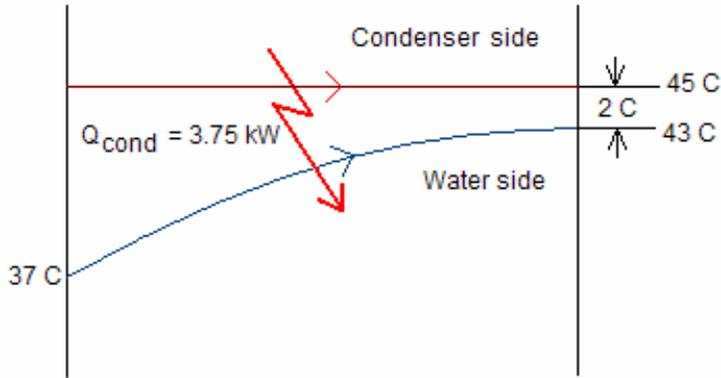
Figure 1.3.1.5 Base case for calculation of UA_{cond}

The approach temperature difference for the polymer heat exchanger, $\Delta T_{approach,hx}$, was fixed at 2°C. The $\Delta T_{approach,hx}$ is defined as shown in Figure 1.3.1.6.
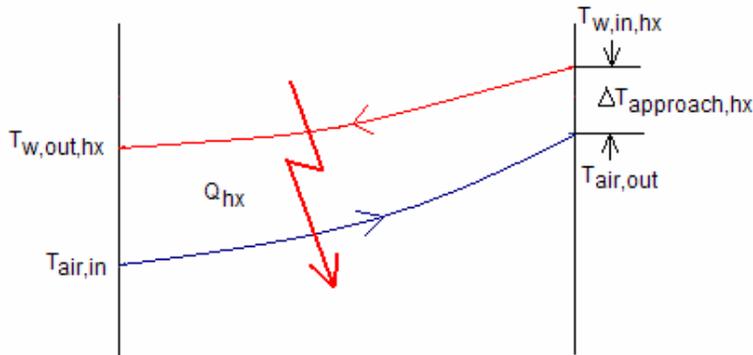
Figure 1.3.1.6 $\Delta T_{approach,hx}$ in polymer heat exchanger

This vapor compression system efficiency was combined with the other component efficiencies to express the system COP as –

$$COP_{sys} = \frac{Q_{evap}}{(W_{comp} + \dfrac{W_{fan}}{\eta_{fan}} + \dfrac{W_{pump}}{\eta_{pump}})}$$
(Eq. 1.3.1.1)

The values of the input variables for a typical run are shown in Table 1.3.1.1. Table 1.3.1.2 shows the outputs from the simulation.

Table 1.3.1.1 Input variables for typical hot HX simulation

Variable	Value	Notes
Q_{evap}	3.5 kW	1-ton
UA_{cond}	0.86 kW/K	Assumed value
D_{out}	4 mm	Initial guess
SDR (D_{out}/t_{tube})	15	Initial guess
RatioSTDout	2	Initial guess
A_{face}	0.4 m^2	Typical of air cooled cond.
$\Delta T_{approach,hx}$	2 C	Initial guess
V_w	0.00015m^3/s (2.5 gpm)	Initial guess
$V_{face,air}$	1.0 m/s	Typical of air cooled cond.
$T_{air,in}$	35 C	Rating condition
T_{evap}	12 C	Ensure dehumidification
ΔT_{sup}	5 C	TXV or EEV
ΔT_{sub}	5 C	Typical at rating condition
$\eta_{pump/fan}$	0.2	Assumed pump/fan efficiency
η_{isen}	0.7	Assumed isentropic efficiency
k_{tube}	0.31 W/mK	Nylon conductivity

Table 1.3.1.2 Outputs from hot HX simulation

Output	~Value
t_{tube}	0.26 mm
$Core_{width}$	0.63 m
$Core_{height}$	0.63 m
$Core_{depth}$	0.30 m
n_{rows}	38
$n_{modules}$	79
n_{total}	3010
$T_{w,,in,hx}$	45.8 C
$T_{w,out,hx}$	39.6 C
$T_{air,out}$	43.8 C
UA_{hx}	1.3 kW/K
COP_{cyc}	4.28
COP_{sys}	4.06

At this point, it would be appropriate to establish a benchmark COP for a better understanding of the effect of different parameters on the polymer heat exchanger performance and overall system performance. Also, this allows us to quantify the inefficiencies incurred by adding a secondary heating loop instead of using the condenser to directly heat the air. Consider the vapor compression cycle shown in Figure 1.3.1.7 in which there is no pressure drop in the condenser and the condenser temperature is equal to the ambient temperature. The remainder of the cycle is characterized by 5°C superheat, a 12°C evaporator temperature, and an isentropic efficiency of the compressor of

0.7. The COP_{cyc} of such an idealized cycle was found out to be ~6.8. This is the thermodynamic limit to the COP_{cyc} which can be achieved by within the given evaporator and the specified ambient temperature, allowing for the possibility of near-infinite condenser air flow rate.
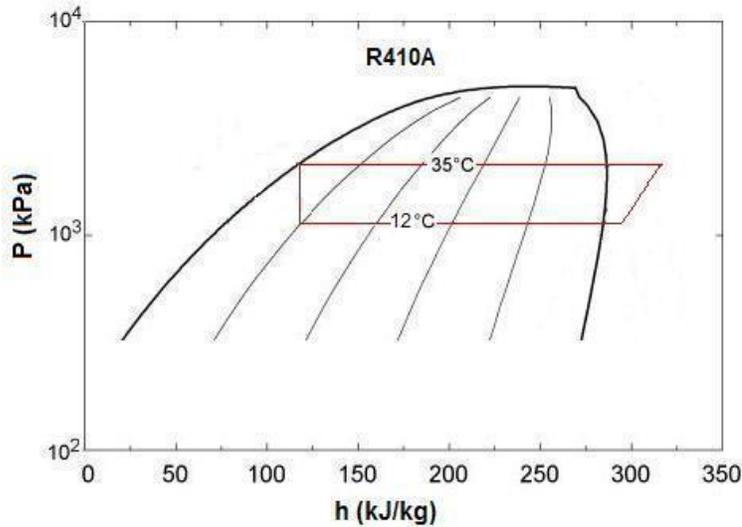
Figure 1.3.1.7 Idealized vapor compression cycle

Optimal heat exchanger configuration was found by letting four parameters (D_{out}, RatioSTD$_{out}$, V_{air}, $V_{dot,w}$ and $\Delta T_{approach,hx}$) vary simultaneously. The bounds on the variable are given in Table 1.3.1.3. It is clear than thinner tube walls imply lower thermal resistance and hence a better thermal performance. Therefore, 0.2 mm thin nylon HTN tubes were used in determining the optimal geometry. The value - 0.2 mm corresponds to the minimum tube thickness which can be produced from nylon, due to manufacturing considerations. The resulting optimal configuration and performance parameters have been summarized in Table 1.3.1.4.

Table 1.3.1.3 Parameter constraints for multi-dimensional optimization

Variables	Bounds	Notes
D_{out}	3 – 6 mm	Lower bound – Manuf. limit
RatioSTDout	1.5 - 3.5	Typical range
V_{air}	0.5 m/s - 2.0 m/s	Typical range
$V_{dot,w}$	Lower: 0.00012m^3/s - 0.00015m^3/s Upper - 0.00027 m^3/s	Lower limit varies according to $\Delta T_{approach,hx}$
$\Delta T_{approach,hx}$	2°C – 5°C	Typical range

Table 1.3.1.4 Results for optimal configuration

Variables	~Value
D_{out}	3.2 mm
RatioSTDout	2.0
V_{air}	1.19 m/s
$V_{dot,w}$	0.00015* m^3/s (2.5* gpm)
$\Delta T_{approach,hx}$	2°C*
Outputs	~Value
$Core_{width}$	0.63 m
$Core_{height}$	0.63 m
$Core_{depth}$	0.24 m
$Core_{volume}$	0.1 m^3
N_{rows}	37
$N_{modules}$	98
N_{total}	3010
$T_{w,in,hx}$	44.5°C
$T_{w,out,hx}$	38.3°C
$T_{air,out}$	42.5°C
UA_{hx}	1.54 kW/K
COP_{cyc}	4.51
COP_{sys}	4.13
$Volume_{polymer}$	0.0045 m^3
$mass_{hx}$	5.1 kg
$mass_{water}$	19.35 kg
$A_{air,side}$	23.47 m^2
$A_{ref,side}$	20.57 m^2

* Parameter at lower bound

We can see from Table 1.3.1.4 that there exist internal optimal values of tube diameter, RatioSTD$_{out}$ and V_{air} while $\Delta T_{approach}$ and $V_{dot,w}$ reach their lower bounds. The optimal tube diameter is representative of the tradeoff between minimizing the water side pressure drop (for large D_{out}) and maximizing air side area (for small D_{out}). Similarly, the optimal RatioSTDout and V_{air} result from the tradeoff between minimizing the air side pressure drop (for low V_{air} and high RatioSTDout) and maximizing the air side heat transfer coefficient (for high V_{air} and low RatioSTDout). A small $\Delta T_{approach}$ keeps the condenser temperature low thus minimizing the compressor work. Also, a low water flow rate keeps the water side pressure drop small. Note that the approach temperature constraint was selected arbitrarily; it could be relaxed at the cost of adding size and cost.

The percent of heat transfer resistance on air side, water side and tube wall were ~86.6, ~8.0 and ~4.9 respectively. This shows that the air side resistance dominates the heat transfer characteristics of the polymer heat exchanger. Thus, using high conductivity material (like metals), for the tubes, would affect the thermal performance

only marginally. The major difference in performance between this polymer heat exchanger and its metal counterpart is due to its lack of extended surfaces.

The optimized tube bundle configuration can be compared to a typical microchannel heat condenser in a one-ton air conditioner. A comparison of the performance and dimensions of the polymer and metallic heat exchanger is given in Table 1.3.1.5.

It can be seen by comparing Tables 1.3.1.4 and 1.3.1.5 that for almost equal system performance, the metallic heat exchanger requires 20 times less core volume than the polymer heat exchanger, which is composed of ~3000 small thin tubes. Specifically, the volume of the material used in the polymer heat exchanger is ~0.0045m^3, which is much more than the volume of metal in microchannel heat exchanger (~ 0.0011 m^3) and ~40 % heavier than the metallic one. When the mass of water is added, the polymer hx becomes almost six times as massive as the microchannel refrigerant-to-air condenser.

Table 1.3.1.5: Typical one-ton microchannel condenser simulation results

Parameter	Metallic hx (R410A system)
Q_{evap}	3.5 kW
A_{face}	0.4 m^2
$\Delta T_{approach}$	2°C
$Core_{width}$	6.05 m
$Core_{height}$	0.077 m
$Core_{depth}$	0.011 m
$Core_{volume}$	0.0054 m^3
$Volume_{metal}$	0.0011 m^3
$mass_{metal}$	3.06 kg
$A_{air,side}$	9.38 m^2
$A_{ref,side}$	1.15 m^2

1.3.2 Air-to-liquid heat exchangers – Analysis of a cooling coil

In this design, water is first cooled by the evaporator of a vapor compression system, and then used to cool the air using a polymer heat exchanger. A schematic of the circuiting is shown in Figure 1.3.2.1.
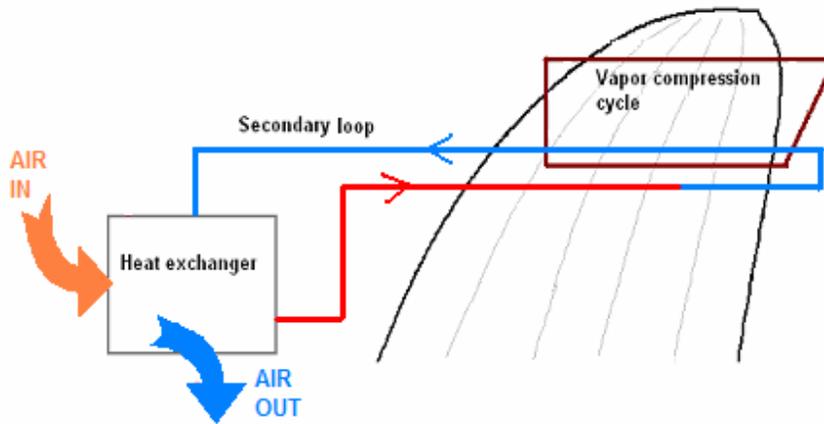
Figure 1.3.2.1 Schematic of the cooling loop

The configuration of the polymer heat exchanger is similar to the heating loop, as shown in Figure 1.3.1.2 and Figure 1.3.1.3.

Similar to the heating loop analysis, EES was used to solve the NxN set of simultaneous equations to explore the design space for an optimal polymer heat exchanger configuration, which was then compared to a typical metallic counterpart. For ease of analysis, the polymer heat exchanger was modeled as a counterflow configuration, a reasonable approximation when the number of passes is greater than 3. Again, the modeling and analysis was done for a vapor compression system with a capacity of one ton and the properties of DuPont's Nylon (Zytel HTN 51G15) were assumed for the tube material.

In the model (Figure 2), the condenser temperature, inlet air and water temperatures into the polymer hx, geometry variables like tube diameter, ratio of the outside diameter to the tube thickness (SDR – Standard diametric ratio), ratio of tube diameter to transverse/longitudinal pitch (D_{out}/S_L) etc. were specified and system performance parameters like COP, outlet air and water temperatures etc. were calculated. The face area was specified as 0.4 m^2, which is a typical value of the face area per ton for metallic evaporators of the type used in ductless mini-split systems. In ducted split systems, the face area can be significantly smaller. The aspect ratio (AR) was taken as 1, representing a square geometry

The following correlations were used in the model –

HEAT TRANSFER CORRELATIONS

Water side: Gneilinski (1976): $0.6 < Pr_{ref} < 10^5$, $1300 < Re_{ref} < 5 \times 10^6$ [10]

                         $Nu_w = 4.3$; $Re < 1300$

Air Side: Zukauskas (1976): $Re < 2 \times 10^5$ [12]

PRESSURE DROP CORRELATIONS

Water side   Churchill (1977) [11]

Air Side: Zukauskas (1976)

Inputs (left boxes):

Box 1:
V_{dot_w}
$T_{a,in,hx}$, $RH_{a,in,air}$
$\Delta T_{approach,hx}$
T_{cond}
ΔT_{sup} , ΔT_{sub}

Box 2:
$Q_{indoor/evap}$, SHR,
UA_{evap}

Box 3:
D_{out}
RatioSTDout -
(S_T/D_{out})
SDR (D_{out}/t_{tube})
$A_{face,hx}$
AR_{hx}

Box 4:
η_{isen}
$\eta_{fan/bower}$
k_{poly}

Center: N X N system model

Outputs (right boxes):

Box 1:
$COP_{cyc/sys}$
W_{comp} , $W_{fan/blower}$
Q_{eond}
$R_{tube / air / water}$
UA_{hx}

Box 2:
$Core_{depth}$
$Core_{height}$
$Core_{width}$
n_{rows} , $n_{modules}$

Box 3:
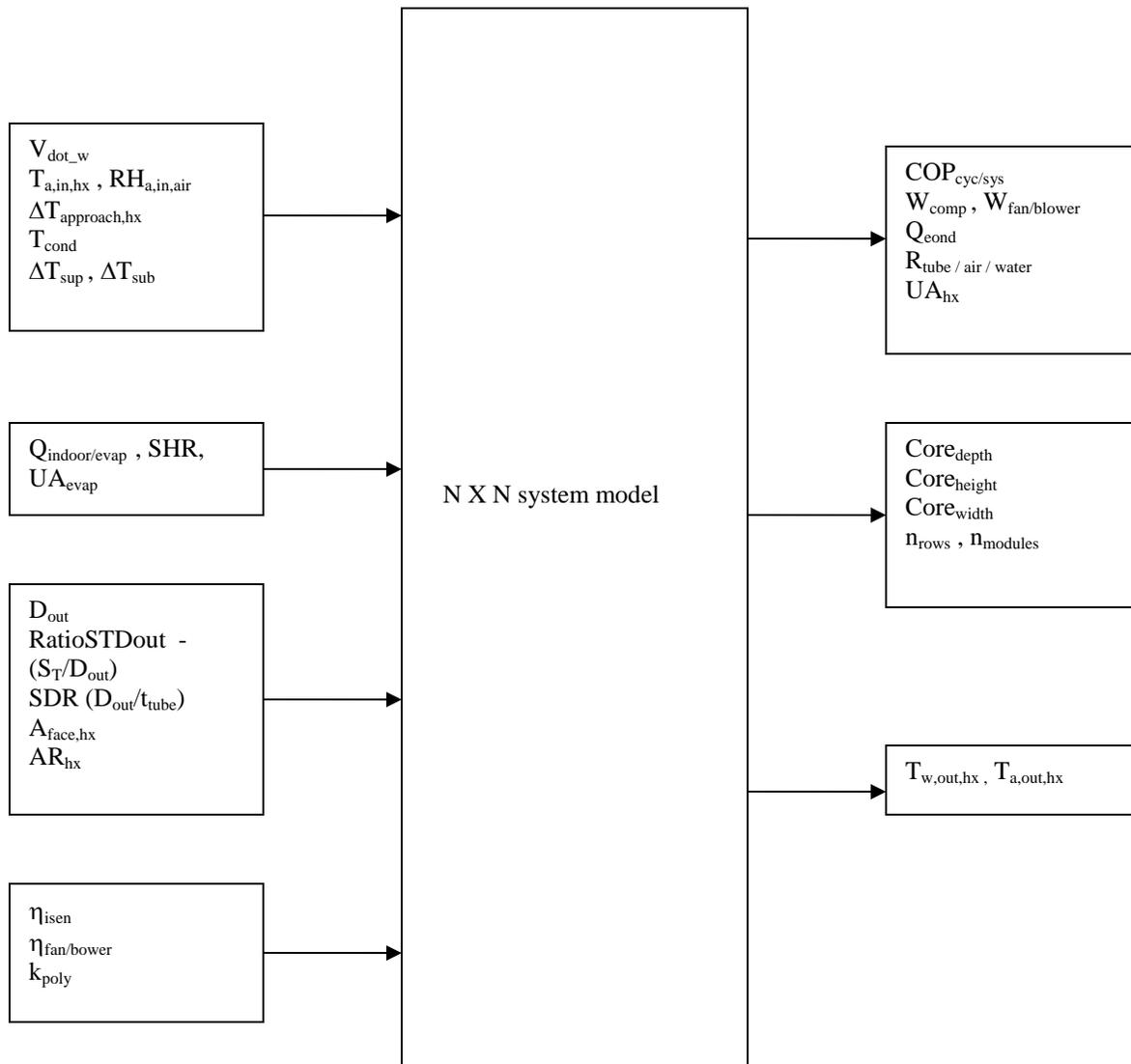$T_{w,out,hx}$, $T_{a,out,hx}$

Figure 1.3.2.2 Inputs and outputs in the heat exchanger model

Then there was a choice between running wet vs. dry – preliminary simulation results show that running partially dry (with a substantially larger polymer heat exchanger) makes it possible to downsize the evaporator by about 30%, and to reduce water flow by more than a factor of two. For simplicity, it has been assumed that the coil is completely wet, so the results from the simulation provide a lower bound on polymer heat exchanger size.

A fixed value of the UA_{evap} = 0.8 kW/C for the refrigerant-to-water evaporator was specified in the model. It was selected to be able to provide 3.5 kW at water supply/return temperatures of 12 and 20°C, respectively, with an approach temperature difference at evaporator outlet of 2°C.
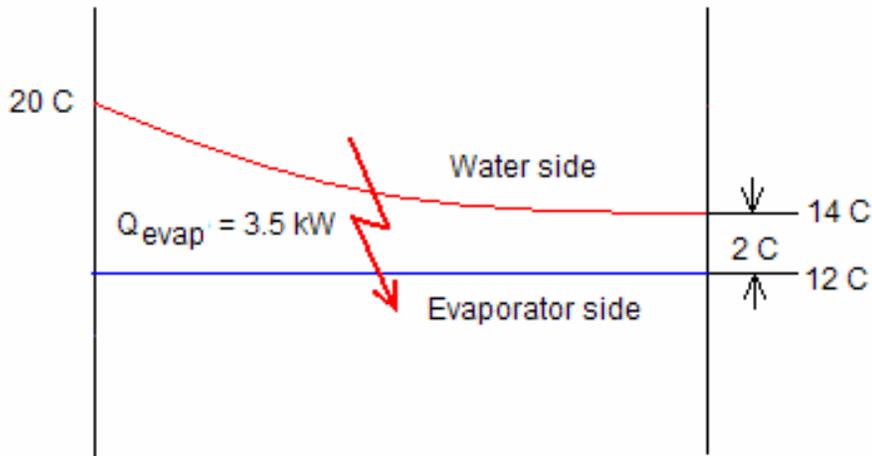
Figure 1.3.2.3 Base case for calculation of UA_{evap}

The $\Delta T_{approach,hx}$ is defined as shown in Figure 1.3.2.4.
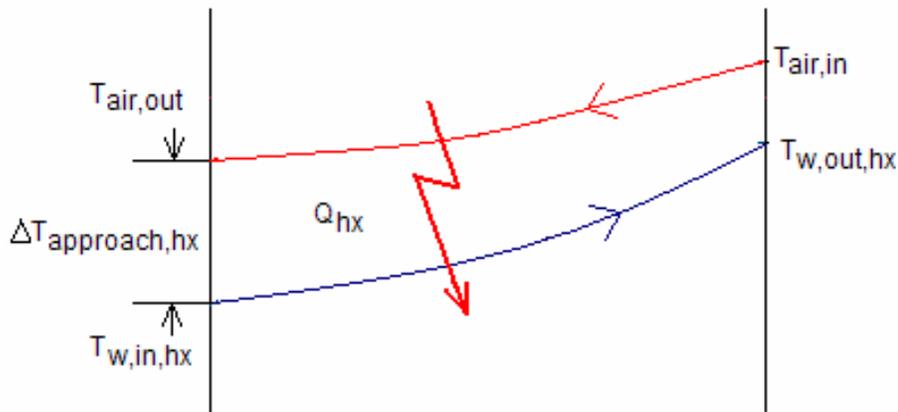
Figure 1.3.2.4 $\Delta T_{approach,hx}$ in polymer heat exchanger

This vapor compression system efficiency was combined with the other component efficiencies to express the system COP as –

$$COP_{sys} = \frac{Q_{evap}}{(W_{comp} + \dfrac{W_{fan}}{\eta_{fan}} + \dfrac{W_{pump}}{\eta_{pump}})}$$

(Eq. 1.3.2.1)

The values of the input variables for a typical run are shown in Table 1.3.2.1. Table 1.3.2.2 shows the outputs from the simulation.

Table 1.3.2.1 Input variables for a typical simulation

Variable	Value	Notes
Q_{evap}	3.5 kW	Fixed evap. capacity
SHR	0.7	Assumed value
UA_{evap}	0.80 kW/K	Assumed value
D_{out}	4 mm	Initial guess
SDR (D_{out}/t_{tube})	15	Initial guess
RatioSTDout	2	Initial guess
A_{face}	0.4 m^2	Typical of air cooled evaporator
$\Delta T_{approach,hx}$	2°C	Initial guess
$V_{w,gpm}$	0.00015m^3/s (2.5 gpm)	Initial guess
$T_{air,in}$ /$RH_{air,in}$	26.7°C / 0.5	Rating condition
T_{cond}	40°C	
ΔT_{sup}	5°C	TXV or EEV
ΔT_{sub}	5°C	Typical at rating condition
$\eta_{pump/fan}$	0.2	Assumed pump/fan efficiency
η_{isen}	0.7	Assumed isentropic efficiency
k_{tube}	0.31 W/mK	Conductivity of Nylon

Table 1.3.2.2 Outputs from the simulation

Output	~Value
t_{tube}	0.26 mm
$Core_{width}$	0.63 m
$Core_{height}$	0.63 m
$Core_{depth}$	0.21 m
n_{rows}	26
$n_{modules}$	79
n_{total}	2062
$T_{w,,in,hx}$	9.9°C
$T_{w,out,hx}$	15.2°C
$T_{air,out}$	11.9°C
UA_{hx}	0.47 kW/K
COP_{cyc}	5.0
COP_{sys}	4.9

     To establish a benchmark or "ideal" COP, the evaporating temperature is assumed to be equal to the temperature of the water coming in. The water inlet temperature into the evaporator, for this ideal case, is taken to be the dew point of the inlet air into the cooling coil (as the cooling coil is assumed to be completely wet) . Thus, with the assumed inlet conditions of 26.7°C/50% RH, the dew point (15.42°C) is assumed to be the ideal evaporating temperature. The remainder of the cycle is characterized by 5°C superheat, a condensing temperature of 40°C and an

isentropic efficiency of the compressor of 0.7. Also, it is assumed that such a cycle has no pressure drops. The COP_{cyc} of such an idealized cycle was found out to be ~6.3. This is the thermodynamic limit to the COP_{cyc} which can be achieved by the system at the specified ambient inlet temperature and humidity, allowing for the possibility of near-infinite evaporator and cooling coil area.
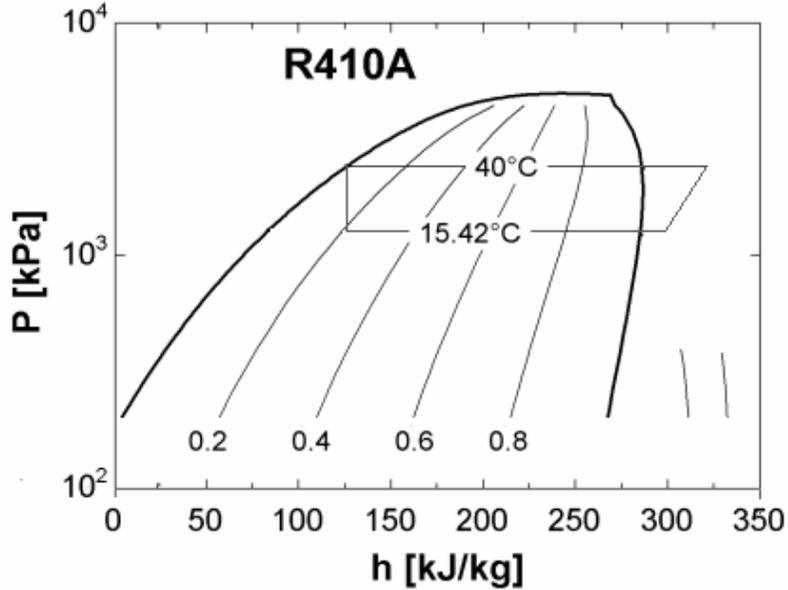

Figure 1.3.2.5 Idealized vapor compression cycle

Similar to the optimization for heating loop, four parameters (D_{out}, RatioSTD$_{out}$, $V_{dot,w}$ and $\Delta T_{approach,hx}$) were allowed to vary simultaneously and COP_{sys} was maximized. The bounds on the variables are given in Table 1.3.2.3. A fixed value of (D_{out}/t) was used for the nylon HTN tubes in determining the optimal geometry – a simple way of holding burst pressure constant. The resulting optimal configuration and performance parameters have been summarized in Table 1.3.2.4.

Table 1.3.2.3 Parameter constraints for multi-dimensional optimization

Variables	Bounds	Notes
D_{out}	3 – 6 mm	Lower bound – Manuf. limit
RatioSTDout	1.5 – 3.5	Typical range
$V_{dot,w}$	Lower: 0.00012m³/s - 0.00015m³/s Upper - 0.00027 m³/s	Lower limit varies according to $\Delta T_{approach,hx}$
$\Delta T_{approach,hx}$	2°C – 5°C	Typical range

Table 1.3.2.4 Results for optimal configuration

Variables	~Value
D_{out}	6* mm
RatioSTDout	1.5*
$V_{dot,w}$	0.00041 m^3/s (6.5 gpm)
$\Delta T_{approach,hx}$	3.5°C
Outputs	**~Value**
Core$_{width}$	0.63 m
Core$_{height}$	0.63 m
Core$_{depth}$	0.20 m
Core$_{volume}$	0.08 m^3
N_{rows}	23
$N_{modules}$	70
N_{total}	1600
$T_{w,in,hx}$	13.0°C
$T_{w,out,hx}$	15.0°C
$T_{air,out}$	16.6°C
UA$_{hx}$	0.56 kW/C
COP$_{cyc}$	5.42
COP$_{sys}$	5.28
Volume$_{polymer}$	0.007 m^3
mass$_{hx}$	8.0 kg
mass$_{water}$	21.5 kg

* Parameter at lower/upper bound

We can see from Table 1.3.2.4 that D_{out} and RatioSTD$_{out}$ reach their upper and lower bounds, respectively. The optimal value of D_{out} is representative of the tradeoff between minimizing the water side pressure drop (for large D_{out}) and maximizing air side area (for small D_{out}). Similarly, the optimal RatioSTD$_{out}$ results from the tradeoff between minimizing the air side pressure drop (for large tube pitch: high RatioSTD$_{out}$) and maximizing the air side heat transfer coefficient (for low RatioSTD$_{out}$). It seems that maximum benefit is obtained by reducing the water side pressure drop (for large D_{out}) and increasing the air side area at the expense of air side pressure drop (for small RatioSTD$_{out}$).

Also, Table 1.3.2.4 shows that there exist optimal values for $\Delta T_{approach}$ and $V_{dot,w}$. A small $\Delta T_{approach}$ keeps the evaporating temperature high thus minimizing the compressor work. However, a low $\Delta T_{approach}$ comes with added size and thereby increased water side pressure drop. Similarly, a high $V_{dot,w}$ keeps the evaporating temperature high but increases the water side and the air side pressure drop. The optimal values of $\Delta T_{approach}$ and $V_{dot,w}$ are representative of this tradeoff.

The percent of heat transfer resistance on air side, water side and tube walls were ~70% , ~20% and ~ 10% respectively. This shows that the air side resistance dominates the heat transfer characteristics of a polymer heat exchanger. Thus, using high conductivity material (like metals), for the tubes, would affect the thermal performance

only marginally. The major difference in performance between this polymer heat exchanger and its metal counterpart is due to its lack of extended surfaces.

The optimized tube bundle configuration can be compared to a typical tube-fin evaporator in a one-ton air conditioner with the same inlet air conditions and the same approach temperature difference (Table 1.3.2.5).

It can be seen by comparing Tables 1.3.2.4 and 1.3.2.5 that for the same cooling capacity, the metallic heat exchanger requires ~10 times less core volume than the polymer heat exchanger, which is composed of ~1600 small thin tubes. Specifically, the volume of the material used in the polymer heat exchanger is ~0.0006 m^3 which is about three times the volume of metal in tube-fin heat exchanger (~ 0.0002 m^3 - including both aluminum fins and copper tubes) and is ~3 times heavier than the metallic one. When the mass of water is added, the polymer hx becomes almost eight as massive as the tube-fin refrigerant-to-air evaporator.

Table 1.3.2.5 Typical one-ton tube fin simulation results

Parameter	Metallic hx (R410A system)
Q_{evap}	3.5 kW
$T_{air,in}$ /$RH_{air,in}$	26.7°C / 0.5
SHR	0.7
$\Delta T_{approach}$	3.5°C
A_{face}	0.12 m^2
$Core_{width}$	0.34 m
$Core_{height}$	0.35 m
$Core_{depth}$	0.07 m
$Core_{volume}$	0.0083 m^3
$Volume_{metal}$	0.0006 m^3
$mass_{metal}$	2.6 kg
$A_{air,side}$	8.61 m^2
$A_{ref,side}$	0.30 m^2

Conclusions

The state of the art of polymer heat exchangers was reviewed, and their suitability for HVAC&R applications assessed. Currently such heat exchangers are used primarily in corrosive environments. These are small-volume applications where customized tradeoffs can be made between material properties and thermal performance and cost penalties. Another class of applications is low-temperature water heating, where the thermal resistance of the polymer is a relatively large part of the total and thin walls are necessary to achieve thermal performance levels comparable to their metallic counterparts. Potential applications in HVAC&R, however, are likely to be air-side dominated, so thin walls may not be the most critical factor. Two other factors were found to limit the thermal-hydraulic performance potential: 1) conductivity is too low for polymers to be used for extended surfaces; and 2) conductivity in structural polymer materials (e.g. polycarbonate) can be enhanced relatively easily in the direction parallel to the plate surface but with great difficulty perpendicular to it. That is why the use of such structural polymers have been limited to diffusing heat from hot spots (e.g. in auto radiator headers) and they would be unsuitable for wall panels or refrigerator wrappers.

A simulation model was developed to get a quantitative estimate of the thermal and material tradeoffs incurred by using polymers in a heating coil tube bundle. Results for a hypothetical 1-ton a/c system showed that systems with optimally-configured polymer heat exchangers could achieve thermal performance similar to their metallic counterparts. But, because of lack of extended surfaces, exchangers are ~ 6 times more massive, and occupy ~ 20 times more volume than conventional microchannel heat exchangers.

The simulation model was also used to explore the potential for using polymer materials to construct tube bundles for cooling coils and cooling tower applications. Such heat exchangers could conceivably be used in secondary loops to facilitate charge minimization in chillers and in systems using toxic or flammable refrigerants, e.g. in vehicles and residential applications. Again, the results showed that an optimally-configured 1 ton a/c system with polymer heat exchangers could achieve thermal performance similar to their metallic counterparts. But, the polymer heat exchanger would be ~ 8 times more massive and occupy ~10 times more volume than conventional tube and fin coils. This estimate is based on a 'fully wet' coil and represents a lower bound on such heat exchangers. Actual coils, which might be running fully/partially dry, would be even more massive.

These results were obtained assuming the thermal, structural and (wall thickness) manufacturability characteristics of nylon. However nylon has other characteristics (e.g. oxidation, moisture permeability) that would have to be overcome by advanced materials before polymer heat exchangers could achieve the performance calculated here.

Another problem that would need to be solved is that of connecting hundreds of tubes to headers. Until these outstanding issues are resolved, polymer heat exchangers are unlikely to find widespread use in the HVAC&R industry.

REFERENCES
1. Davidson, J. and S. Mantell, "Use of polymers in liquid-to-liquid heat exchangers; Applied to low-cost solar water heating systems", Task III Report, University of Minnesota, 1998.
2. Heating, L. W., J. H. Davidson, R. Raman and S. C. Mantell, "Thermal and economic analysis of plastic heat exchangers for solar water", *Proceedings SOLAR '99*, ASES, Portland, ME.
3. http://www.me.umn.edu/divisions/environmental/solar/research.htm
4. Shannon, M. A., M. L. Philpott, N. R. Miller, C. W. Bullard, D. J. Beebe, A. M. Jacobi, P. S. Hrnjak, T. Saif, N. Aluru, H. Sehitoglu, A. Rockett and J. Economy, "Integrated mesoscopic cooler circuits (IMCCS)," *Proceedings of the 1999 International Mechanical Engineering Congress and Exposition*, Nashville, TN, AES-Vol. 39, pp. 75-81, November 1999.
5. http://www.microplastics.com/Materials/NYLON%2066.htm
6. Barnes, P.R. and C. W. Bullard, "Minimizing TEWI by charge reduction in a compact chiller", ACRC TR-176, August 2000.
7. Davidson, J., University of Minnesota, personal communication, 2004.
8. Rubio, M., Fafco Co., personal communication, 2004.
9. Gneilinski, V., "New equations for heat and mass transfer in turbulent pipe and channel flow", *Int .Chem. Eng.*, Vol. 16, pp. 359-368, 1976.
10. Churchill, S. W., "Friction-factor equations spans all fluid flow regimes", *Chemical Engineering*, pp. 91-92, 1977.
11. Zukauskas, A. and Ulinskas, R. "Efficiency parameters for heat transfer in tube banks", Heat Transfer Engineering, 6:2 pp 19-25, 1985.

Appendix A. Polymers for Use in Heat Exchangers

The literature review was conducted by contacting companies involved in design and production of polymer heat exchangers (like DuPont, Fafco, Ametek etc.), and the government agency funding such research (DoE's National Renewable Energy Laboratory and its contractor, the Univ. of Minnesota).   This provided the basis for identifying promising polymer materials suitable for use with liquid-to-liquid heat exchangers with a design pressure requirement of ~1.1 MPa. A brief summary of their work is presented here. Polymer materials for air-to-air and air-to-liquid heat exchangers can be selected on similar guidelines. A more detailed discussion of liquid-to-liquid heat exchangers is out of the scope of this project.

A review of the available literature revealed that many of the commercially available polymer heat exchangers are made of rather expensive fluoropolymers (polyvinylidine fluoride (PVDF), Teflon), or materials like polypropylene (PP), which are not suitable for use with water over 60° C.

It was recognized that the selection of appropriate material/s for heat transfer application depends on the following six factors:

1. Thermal conductivity
2. Strength and stiffness
3. Temperature limits (thermal index, glass transition temperature etc.)
4. Refrigerant/Water absorption and diffusion properties
5. NSF codes and standards for the targeted application and
6. Cost

Issues related to permeability of the polymer to water, chemical resistance to chlorinated water, ions found in potable water, mating to piping and tanks, ease of repairs, and compliance with applicable plumbing codes were also addressed.

The polymers in Table A.1 were identified as suitable for use with water/glycol.

The polymers were classified on the basis of their tensile strength and flexural modulus. Polymers having tensile strength >80 MPa and flexural modulus >2500 MPa were classified as 'high strength' and others were categorized as 'low strength' polymers. All the polymers, in Table A.1, absorbed < 1% of water by weight after 24 hrs of being immersed in water.

Of all the high strength polymers chosen, nylon polyamide (NP) had the lowest cost while cross-linked polyethylene (PEX) was the least costly of the low strength polymers.

High strength polymers can be used to make thin-walled components for heat exchangers (e.g. thin tubes and sheets) while low strength polymers require thick walls, which hinder their heat transfer performance. But low strength polymers are usually cheaper and can be used in heat exchanger components which require bending (e.g. coiled tubes), as they have low flexural modulus.

Also, if the operating pressure and temperatures are low, then cheap plastics like PolyEthylene (PE) and PolyPropylene (PP) can also be used. However, these materials are not resistant to propylene glycol, which is commonly used on food-related applications. Polypropylene works well in water and its life can be extended by

adding suitable additives [8]. However it has a low temperature limit. PolyButylene has shown some promise as a suitable polymer for heat exchangers but still testing is being done in high temperature and oxidative environments [7].

In summary, it can be said that the polymer to be used in a particular heat exchanger depends on the structural and thermal requirements as well as cost. Also, standards and codes provided by NSF, ASTM (American Society for Testing and Materials) and PPI (Plastic Pipe Institute) should be followed when polymer heat exchangers are being designed for potable water system or toxic environments.

Table A.1 Summary of polymers used in heat exchangers [1]

Polymer	Conductivity W/mK	Classification	Notes
NP/HTN (High temperature grade Nylon Polyamide)	0.31	High strength	Can be make thin walled tubes of ~0.2 mm thickness
PEEK (PolyEther-Ether-Ketone)	0.43	High strength	High temperature processing required
PLS (PolySulfone)	0.26	High strength	Very stable in water. Can make thin walled tubes
PPA (PolyPhthalAmide)	0.35	High strength	
PPS (PolyPhenylene Sulfide)	0.5-0.59	Low strength	Extremely dimensionally stable in water
FEP (perFluoro Ethylene-Propylene)	0.195	Low strength	
PEX (Cross linked PolyEthylene)	0.38	Low strength	Good history in hot water applications.
PFA (perFluoro Alkoxy Alkane)	0.195	Low strength	
PTFE (PolyTetra FluoroEthylene)	0.24	Low strength	