Performance Comparison of Hydronic Secondary Loop Heat Pump and Conventional Air-Source Heat Pump

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ABSTRACT

In residential heat pump systems, the motivation for secondary loop systems is to allow for the use of flammable or toxic refrigerants with lower global warming potentials than the currently employed HFC refrigerants. The addition of radiant panels as integral building components (embedded in concrete at construction) is becoming more common. Combining the large surface area of the radiant panel and an efficient primary loop, the hydronic secondary loop heat pump system can greatly outperform the conventional air-to-air heat pump. The improvement in coefficient of performance is as much as 38% over the conventional air-to-air heat pump when the secondary loop hydronic system is employed. Due to the large area of the radiant panel, the condensing temperature of the primary loop for the hydronic secondary loop heat pump can be reduced by as much as 5°C at high ambient temperatures.

1. INTRODUCTION

Radiant heating and cooling systems are becoming more and more popular around the world. As of 2002, radiant heating and cooling were employed as many of 90% of new constructions in Korea, and 30-50% in Germany (Olesen, 2002). Many of these systems employ electric water heaters as the heat source, which while easy to implement, is not the most efficient solution.

The conventional air-to-air heat pump is quite ubiquitous. 1.8 million split-type air-source heat pumps were produced by U.S. manufacturers in 2011, and 20 million units between 1992 and 2011 (AHRI, 2012). Many of the modern heat pump units operate with the refrigerant R410A which has a global warming potential (GWP) of 2100, versus a GWP for propane of 20 (Calm, 2007). Regulatory pressure in Europe and elsewhere is pushing towards systems that do not use HFC refrigerants like R410A, and regulatory bodies are reconsidering the use of flammable refrigerants like propane.

Radiant heating systems are employed not just for their performance benefits. They can also improve occupant comfort due to a more even temperature profile in the occupied space (Olesen, 2002). In addition, it has been suggested that the air temperature setpoint can be reduced due to the use of radiant heat, although that effect has not been investigated here.

In principle, the hydronic heating system could also be coupled with a solar collector and use the solar energy as the heat source for the heat pump.

2. SYSTEM DESCRIPTIONS

2.1 Systems Under Consideration

In this study, two different types of systems are under consideration for the heating of a residential structure. These systems are a conventional air-to-air heat pump that operates with an HFC-based working fluid (here refrigerant R410A) and a secondary loop heat pump system that employs radiant panels for heating of the occupied space. The secondary loop system can use flammable working fluids like propane that would not be acceptable for direct expansion systems.

Figure 1a shows a schematic of the air-to-air direct expansion heat pump system. In the direct expansion system, refrigerant exits the evaporator (the outdoor heat exchanger) at state point 1. The compressor then compresses the refrigerant from state point 1 to state point 2. The refrigerant is heated against the ambient air up to state point 3 as it passes through the vapor line on its way to the condenser. In the condenser, heat is delivered to the heated space by the condensation of the refrigerant, and the refrigerant condenses from state point 3 to state point 4. In general, the refrigerant is subcooled at the outlet of the condenser. At the outlet of the condenser, the refrigerant passes through the liquid line from state point 4 to state point 5. The refrigerant next passes through the expansion device which throttles the refrigerant to state point 6, the inlet to the evaporator. Then the refrigerant re-enters the evaporator and the cycle continues.

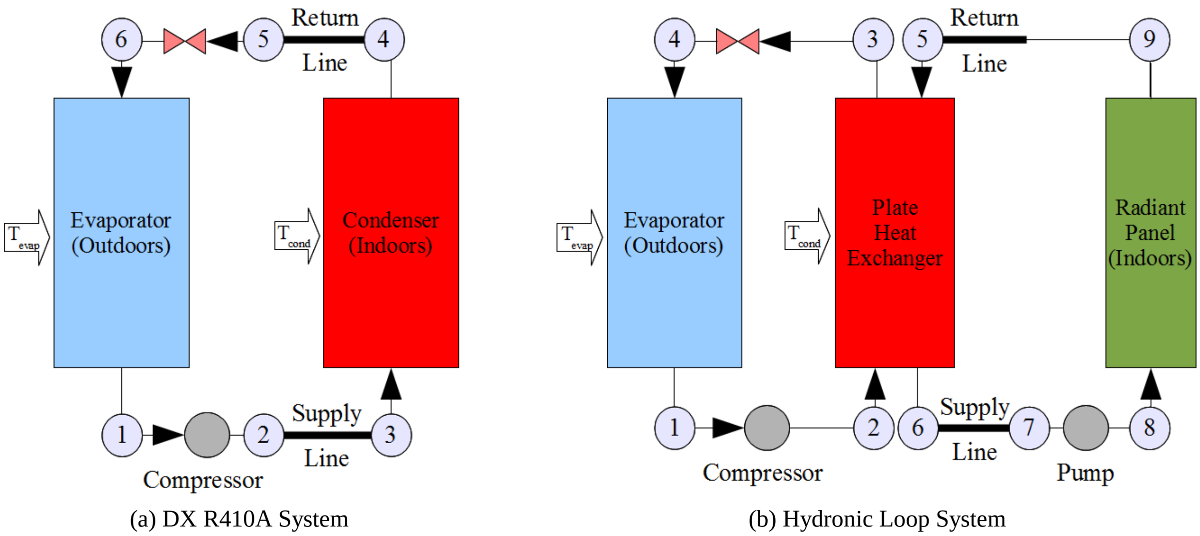


Figure 1: Schematics of DX and Secondary Loop systems

In the secondary loop heat pump system considered here, there are a few modifications to the conventional air-to-air heat pump. Figure 1b shows a schematic of the secondary loop system. The primary refrigerant loop (state points 1 to 4) is essentially the same, except that the refrigerant delivers its heat into a secondary loop in the plate heat exchanger as it condenses, and there is are no vapor or liquid lines since the whole compressor-expansion device-plate heat exchanger loop can be close-coupled, decreasing greatly the piping pressure losses and the refrigerant charge.

In the secondary loop, a secondary working fluid is heated from state point 5 to state point 6 in the plate heat exchanger against the condensing refrigerant. The warmed secondary working fluid then passes through a pump which is required to overcome the pressure drops from the heat exchangers and the lines. The secondary working fluid passes through the supply pipe and is then delivered to the radiant panels. In the radiant panels, the secondary working fluid is cooled and delivers its heat to the heated space. The cooled secondary working fluid is then returned to the plate heat exchanger through the return pipe.

For the secondary loop system, there are a number of options for the secondary working fluid, but water is the most obvious choice. It has the highest mass specific heat of any liquid, and one of the highest densities of any liquid. In addition, its low viscosity results in low pressure drops. For radiant systems applied to heating applications, water would be the secondary working fluid.

2.2 Air-Conditioning Heat Pump Model (ACHP)

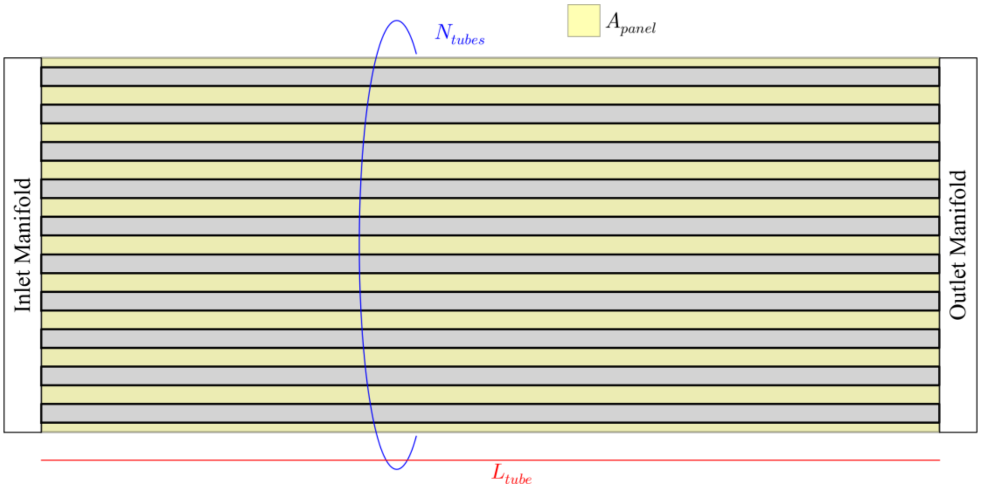
A specialized model has been developed to analyze direct-expansion and secondary loop heat pump and air-conditioning systems (ACHP) that is freely available online, including source code. The details of the ACHP model are provided in the documentation available online[1].

In ACHP, each of the heat exchanger models are based on moving boundary formulations. Essentially, the moving boundary heat exchanger model is based on using a numerical solver to find the locations where the refrigerant changes phase, and then solve each of the portions of the heat exchanger separately. The addition of partially-wet/partially-dry air-side surfaces analysis is also included in the evaporator.

The compressor model is based on 10-coefficient compressor performance map with appropriate correction for compressor inlet superheat. Models are also available for the plate heat exchanger, line sets and other components. The refrigerant and humid air properties are based on a reference-quality property database developed in parallel with ACHP[2].

Coupling all the component models together, a multi-dimensional numerical solver is used to find the evaporation and condensation saturation temperatures for the refrigerant loop and enforce a few energy balances. Either charge or refrigerant subcooling can be imposed, though subcooling was imposed for all the work carried out here. The expansion device is an idealized device that can achieve a given evaporator outlet superheat. In the case of the secondary loop, the solver is also used to find a secondary loop temperature.

2.3 Radiant Panel Model



The radiant panel is formed of a number of tubes with inlet and outlet manifolds. In practice, the tubes may be bent to fit the contours of the space it is installed in, but for the purposes here, the tubes are assumed to all be straight. The radiant panel is then sized based on the area available for the panel. If the total area available for the panel is given by A_{panel}, then the length of the tubes can be obtained from

$$L_{tube} = \frac{A_{panel}}{w_{tube-tube} N_{tubes}} \tag{1}$$

where $w_{tube\text{-}tube}$ and N_{tube} are the tube-to-tube centerline distance (in meters) and the number of tubes forming the panel respectively. For instance, the average floor space of American homes in 2010 was 222 m^2 [2,392 ft^2][3]. If 80% of the available area is used for the radiant paneling and there are 10 tubes with 0.254 m [10 inch] tube-to-tube centerline distance, then the tubes would need to be 69.9 meters long.

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[1] ACHP 1.3: http://achp.sf.net

[2] CoolProp: Fluid properties for the masses. http://coolprop.sf.net/

[3] http://www.census.gov/const/C25Ann/sftotalmedavgsqft.pdf

If the spreading heat conduction thermal resistance in the radiant panel is neglected, there are two thermal resistances that govern the heat transfer from the radiant panel to the surroundings. The overall heat transfer conductance can therefore be given by

$$UA = \left(\frac{1}{\bar{h}_{air} A_{panel}} + \frac{1}{\bar{h}_{water} N_{tubes} \pi D_{i,tube} L_{tube}} \right)^{-1} \tag{2}$$

which includes the convective thermal resistance in the tubes as well as the combined radiant and convective air-side thermal resistance. The overall air-side heat transfer coefficient is usually on the order of 10 W/m^2/K.

With the value for *UA* known, the water outlet temperature can then be obtained from (Bergman, 2011)

$$\frac{T_\infty - T_{w,o}}{T_\infty - T_{w,i}} = \exp\left(-\frac{UA}{\dot{m}_{water} c_p} \right) \tag{3}$$

and the heat transfer rate is then given by

$$Q = -\dot{m}_{water} c_p (T_{w,i} - T_{w,o}) \tag{4}$$

The pressure drop in the radiant panel is governed by the internal flow pressure drop relations as described in the ACHP documentation.

2.2 Air-To-Air Heat Pump System Analysis

The required model parameters for the R410A air-to-air heat pump system are summarized in Table 1 (heat exchangers) and Table 2 (other parameters). The heat exchanger parameters were obtained from the analysis of Shen (2006). This system is a nominal 3-ton cooling capacity system.

Table 1: Heat Exchangers for R410A system based on Bo Shen

	Evaporator	Condenser
Tubes per bank [-]	41	32
Number of bank [-]	1	3
Number of circuits [-]	5	6
Length of tube [m]	2.286	0.452
Tube OD [m]	0.007	0.009525
Tube ID [m]	0.0063904	0.0089154
Longitudinal tube pitch [m]	0.0191	0.0254
Transverse tube pitch [m]	0.0222	0.0219964
Fin Type	Wavy Lanced	Wavy Lanced
Fins/inch [1/in]	25	14.5
Twice fin amplitude [m]	0.001	0.001
½ period of fin waviness [m]	0.001	0.001
Fin thickness [m]	0.00011	0.00011
Fin conductivity [W/m/K]	237	237
Humid air volume flow rate [m^3/s]	1.7934	0.5663
Atmospheric pressure [kPa]	101.3	101.3
Relative humidity [-]	0.51	0.51
Fan power [W]	160	438

Table 2: Other parameters for R410A system

Parameter	**Value**
Refrigerant	R410A
Condenser outlet subcooling [K]	7.0
Evaporator outlet superheat [K]	5.0
Line set length [m]	7.6

2.3 R410A Heat Pump Performance Results

ANSI/AHRI Standard 210/240 governs the rating of unitary heat pump units, and provides a few rating points for which heat pump manufacturers must provide performance data. The ASHRAE standard rating point H1 is used in this study as the conventional rating point. The H1 rating point employs an 8.33°C [47°F] air inlet temperature to the evaporator, and a 21.1°C [70°F] air inlet temperature to the condenser. Standard 210/240 provides rating points as low as -8.33°C [17°F] air inlet temperature to the evaporator (rating point H3).
Table 3 summarizes the results for the three rating points. Both COSP and Capacity are very nearly linear with evaporator air inlet temperature.

Table 3: Modeling Results for the R410A Heat Pump

Rating Point	Condenser Air Inlet Temp.	Evaporator Air Inlet Temp.	COSP	Capacity
H1	21.1°C [70°F]	21.1°C [47°F]	3.44	10000 W
H2	21.1°C [70°F]	1.66°C [35°F]	2.96	8341 W
H3	21.1°C [70°F]	-8.33°C [17°F]	2.29	6242 W

2.4 Secondary Loop System

In order to provide a fair comparison between the two systems, the same operating conditions have been used for both systems. In the secondary loop system, the condenser is replaced with a plate heat exchanger described by the geometry in Table 4.

Table 4: Geometry of Plate Heat Exchanger for Secondary Loop System

Parameter	Value
Number of plates [-]	46
B_p [m]	0.117
L_p [m]	0.300
Plate Amplitude [m]	0.001
Plate Thickness [m]	0.0003
Plate Conductivity [W/m/K]	15
Plate Wavelength [m]	0.0628
Inclination Angle [deg]	60

The saturation pressure of propane at 0°C (474 kPa) is quite a bit lower than that of R410A (800 kPa). As a result, the number of circuits in the evaporator must be increased in order to have a well-controlled pressure drop in the evaporator. The number of circuits in the evaporator was increased to 12, otherwise all the parameters of the secondary loop evaporator are the same as the evaporator in Table 1. The water flow-rate through the secondary loop system was set at 0.38 kg/s.

Furthermore, the compressor displacement of the propane secondary loop system was scaled slightly in order to yield the same capacity as the R410A system at the H1 rating point. The propane compressor displacement (and therefore mass flow rate and electrical power) was decreased by 5.3% when the scaling parameter was introduced.

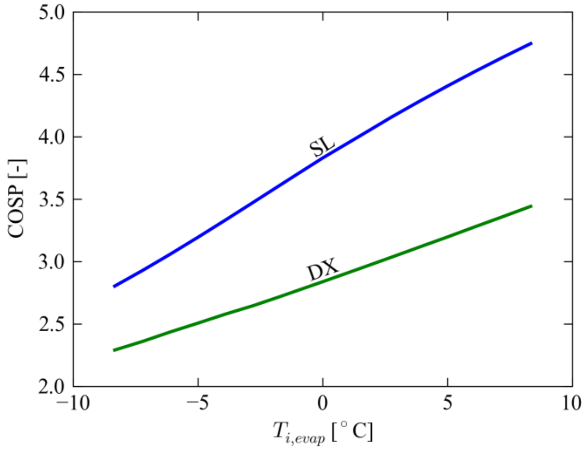

Figure 2: COSP of the systems as a function of the air inlet temperature to the evaporator

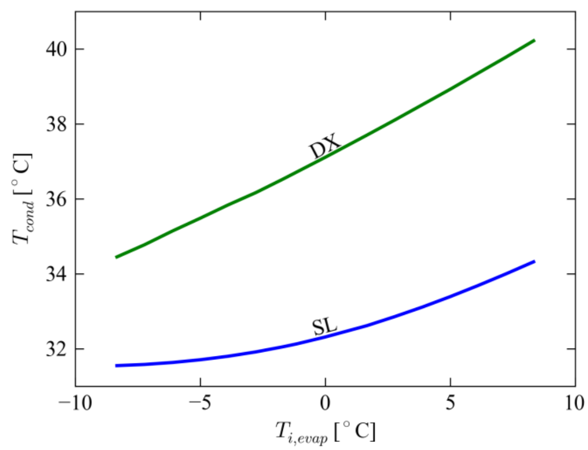

Figure 3: Condensing temperature of the systems as a function of the air inlet temperature to the evaporator

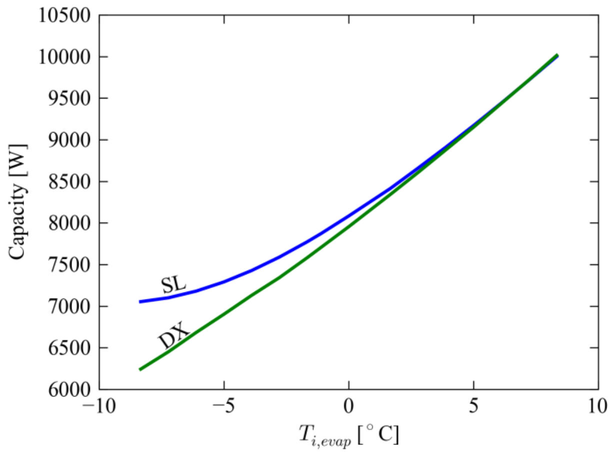

Figure 4: Heating capacity of the systems as a function of the air inlet temperature to the evaporator

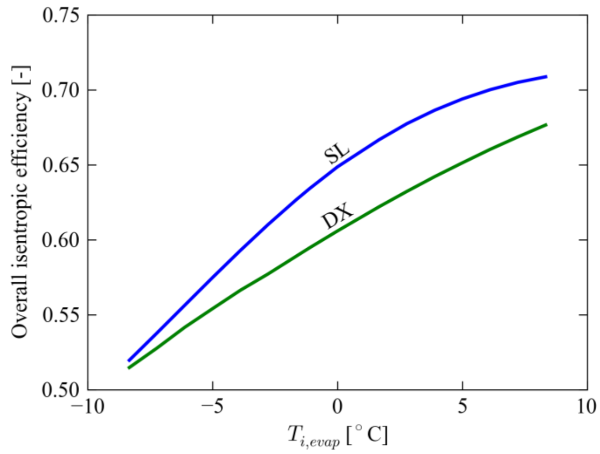

Figure 5: Compressor overall isentropic efficiency for the systems as a function of the air inlet temperature to the evaporator

As can be seen from Figure 2 to Figure 5, by every important performance metric, the performance of the propane secondary-loop system is better than that of the R410A air-to-air heat pump.

Figure 2 shows the COSP for both the secondary loop heat pump and the direct expansion heat pump. These results show that for all the evaporator air inlet temperatures, the COSP of the secondary loop system is better than that of the direct expansion system. There a number of factors contributing to the improved performance with the secondary loop system. For one, in the secondary loop system, the fan power required for the condenser (which is quite significant) is removed. In addition, the condensing temperature of the secondary loop system can be reduced, further improving the compressor electrical power input. Figure 3 shows the condensing temperature of the systems.

Figure 4 shows the heating capacity of the system as a function of the air inlet temperature to the evaporator. For all temperatures investigated here, the capacity of the secondary loop system is better than the direct expansion system. Although the compressor for the secondary loop system was not specifically selected in order to match the lower condensing temperatures, the compressor map predicts a higher overall isentropic efficiency, as seen in Figure 5.

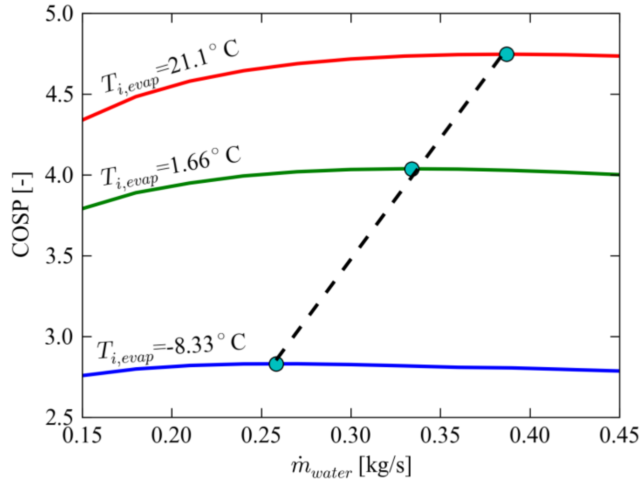

Figure 6: COSP of secondary loop systems as a function of water mass flow rate

Figure 6 shows the COSP of the secondary loop systems for a range of water mass flow rates and evaporator air inlet temperatures. The system COSP is not very sensitive to the mass flow rate of water; as a result, a fixed mass flow rate of water can be used over the full operating envelope with very little decrease in system efficiency. All other parameters were unchanged from the comparisons above.

Further optimization of the mass flow rate of the secondary loop can be achieved by determining the optimal mass flow rate for a selection of evaporator air inlet temperatures (shown with the circular markers in Figure 6) and then conducting a linear fit of the optimal mass flow rate versus the evaporator air inlet temperature. The optimal mass flow rate for this particular system as a function of evaporator air inlet temperature can therefore be given by

$$\dot{m}_{water,opt} = 0.004146 \cdot T_{i,evap} + 0.3064 \tag{5}$$

where $T_{i,evap}$ is in °C and $\dot{m}_{water,opt}$ is in kg/s.

Figure 7 shows that as long as the water flow rate is high enough, there is very little impact on the heat pump capacity. As a matter of fact, the use of the optimal water flow rate from Equation (5) would yield a good solution for the water flow rate, and would allow for a simple control strategy if the secondary loop pump speed were controllable.

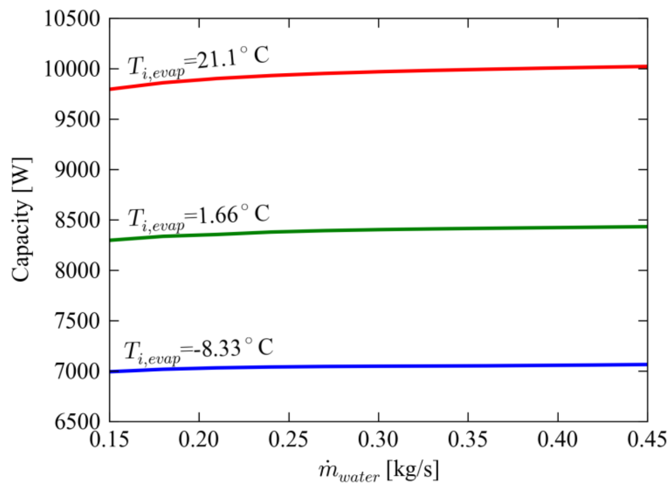

Figure 7: Capacity as a function of secondary loop water flow rate and air inlet temperature to evaporator

3. SYSTEM COMPARISONS

It is useful to consider the energy flows and other system parameters for the rating point H1 in order to clarify the differences between the secondary loop options.

Table 5: Summary of system parameters for operation at rating point H1

		Air-Source DX	Secondary Loop
Energy Flows	Compressor Power [W]	2311	1820
	Evaporator Fan Power [W]	160	160
	Condenser Fan Power [W]	438	-
	Pump Power [W]	-	27
	Net Power [W]	2909	2007
	System Capacity [W]	10000	10000
Part Sizing	Compressor pumped vol. rate [m^3/s][4]	6.121	10.29
	Evaporator air-side area [m^2]	105.34	105.34
	Condenser air-side area [m^2]	36.8	-
	Radiant panel area [m^2]	-	132

With the use of the secondary loop system, the relatively high condenser fan power of the DX system is traded for a much lower secondary loop pump power, which reduces the energy consumption by 411 W. The decreased condensing temperature of the secondary loop system results in a further 491 W reduction in compressor input power.

The compressor pumped volumetric rate for the secondary loop system is 68% larger than that of the R410A compressor, due to the fact that the density of the propane is lower than that of R410A. Thus, at the same evaporation temperature, and for the same heating capacity, a larger volumetric flow rate is required for the propane compressor.

The cause of the lower condensing temperature for the secondary loop system can be clearly seen from a comparison of the condenser air-side area of the DX system and the radiant panel dimensions of the secondary loop system. The available area for heat transfer of the radiant panel is 3.6 times greater than that of the condenser installed in the ductwork.

3.1 Other considerations

The use of a secondary loop system is a natural first step to a multi-zoned heating system. Robust zone temperature control can therefore be achieved through the use of variable water flow rates to the zones, each zone getting a radiant panel. Since water is used as the fluid flowing to the zones (rather than a refrigerant), the pressure drop can be well-controlled for long line sets

The secondary loop system could also be used in cooling mode, though that is not considered here. The performance of the secondary loop systems in cooling mode have been shown to be competitive with, if not better than, the performance of direct expansion R410A systems.

CONCLUSIONS

The benefits to system efficiency through the use of a secondary-loop hydronic heat pump are quite significant. This suggests that it should be straightforward to design a hydronic heat pump that can easily achieve the same seasonal performance as a conventional HFC-based air-source heat pump. Furthermore, this technology would be easily adapted for cooling mode operation or multi-zone systems.

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[4] Based on the compressor map; pumped mass flow rate divided by density at compressor inlet. Includes the volumetric efficiency implicitly since the volumetric efficiency is built into the compressor map

NOMENCLATURE

Parameter	Units	Description
A_{panel}	m^2	Area of panel
c_p	J/kg/K	Mass specific heat
$D_{i,tube}$	m	Inner diameter of tube
$\bar{h}_{air}$	$W/m^2/K$	Air mean heat transfer coefficient
$\bar{h}_{water}$	$W/m^2/K$	Water mean heat transfer coefficient
L_{tube}	m	Length of panel
$\dot{m}_{water}$	kg/s	Water mass flow rate
$\dot{m}_{water,opt}$	kg/s	Optimal water mass flow rate
N_{tubes}	-	Number of tubes
$T_{i,evap}$	K	Evaporator air inlet temperature
T_{cond}	K	Condensing temperature
$T_{w,i}$	K	Water inlet temperature
$T_{w,o}$	K	Water outlet temperature
T_∞	K	Ambient temperature
$w_{tube\text{-}tube}$	m	Tube-tube centerline distance
UA	W/K	Overall heat transfer conductance

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