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Evaluation Of Subcooling Control In Residential Heat Pumps Through Experimental And Model Analysis

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ABSTRACT

Reversible systems present an issue in properly optimizing the charge for both cooling and heating operation to achieve maximum efficiency. Subcooling control can prevent this problem by regulating the expansion device and provide the best performance in both cooling and heating. This paper focuses on subcooling control in residential heat pumps with both large and small heat exchangers (HX) and evaluating the potential improvement from this expansion control strategy as a function of the heat exchanger sizes. The work is divided into an experimental investigation and a model analysis of the effect of both indoor and outdoor heat exchanger's UA values on the performance increase obtained through subcooling control. Two 2-Ton (7 kW) off-the-shelf residential systems were evaluated: a high-efficiency large-HX system with variable compressor speed and superheat-controlled electronic expansion valve; and a small-HX system with fixed compressor speed and orifice piston tube. The results showed an increase in heating performance factor of +4.1% and +6.2% for the large-HX and small-HX systems at the rating condition, respectively. Although subcooling control allows the evaporator to operate with no superheat increase its saturation pressure and resulting in lower compressor power, there's a reduction on the desuperheating heat transfer which leads to decreased heating capacity, as observed with the small-HX system. A system model was validated using the experimental data, and its results show that a decrease in condenser size is not strongly correlated to an increase in the potential improvement in performance from subcooling control. But the improvement is substantial regardless of heat exchanger size when considering that most systems' charge is not optimized for heating operation.

The control scheme for heat pumps also does not match the predicted curve use in air conditioning which is based on the difference between condenser saturation temperature and condenser air inlet temperature. The deviation in subcooling control behavior for heat pumps is possible due to its large variation in pressure ratios or maldistribution introduced by the installation of the EXV. A linear correlation for the pressure ratio does provide better agreement, especially for the small-HX system, but further investigation is required to develop a simple control strategy to maximize efficiency for heating without the need for pressure measurement.

1. INTRODUCTION

The increasing pressure for environmentally friendly residential HVAC systems along with rising natural gas prices has boosted the demand for residential heat pump systems. Since most residential systems focused heavily on air conditioning the optimization always favored the cooling operation. Strupp et al. (2010) proposed using a low-side accumulator to absorb refrigerant charge fluctuations and keep the compressor suction quality at a safe level while the expansion valves regulated the condenser subcooling and showed the existence of an optimum subcooling through system modeling. Carvalho and Hrnjak (2020ab) showed experimentally that subcooling control could allow for maximum efficiency at both A/C and H/P operation while using the accumulator as a low-pressure charge receiver.

Pöttker and Hrnjak (2015a) have previously shown that controlling subcooling can potentially increase system performance by 2.7%-8.4% in simulations comparing different refrigerants. The authors also experimentally validated these findings by determining COP-maximizing subcooling values for different mobile air conditioning systems using R134a and R1234yf. While the system was identical and the capacity was kept at 4.1 kW by controlling compressor speed the optimal subcooling values found for R134a and R1234yf differed with values of 9 K and 11 K, respectively. Subcooling control in residential heat pumps has not been thoroughly investigated yet. Some research has been done in heat pump water heater and optimal expansion control through keeping the condenser approach at a fixed value attempting to match the maximum efficiency at different conditions by relying on the optimization based on the subcooling heat transfer region to minimize exergy destruction in the condenser (Pitarch et al., 2017abc; Hervas-Blasco et al., 2018/2019).

Menken et al. (2014) also evaluated a discharge superheat control with a high-pressure receiver and compared to a subcooling controlled system using an accumulator. The authors concluded that discharge superheat control using a receiver showed slightly higher COP and HPF than the subcooling control with accumulator. Although the high-pressure receiver would require modification to operate in both cooling and heating mode while the subcooling controlled accumulator system can be used in both operations without any modification. The comparison also uses the same refrigerant charge for both systems which could have a bias on the results obtained.
Carvalho and Hrnjak (2020b) provided an initial analysis of the potential improvement from subcooling control in residential heat pumps, observing an improvement in HPF ranging from +1.9% to +4.2% at high load conditions and +7.1 to +19.1% for low load conditions.

This paper aims to build up on some of the previous research done on the effect of subcooling in HVAC systems' performance, specifically focused on the heat pump operation by evaluating two separate systems experimentally with different heat exchangers, compressor and expansion devices and comparing their results when subcooling control is implemented. A system model is used to corroborate the experimental results and provide further insight into the behavior of heat pump operation in a reversible residential air-conditioning system.

2. THEORETICAL ANALYSIS

2.1 Ideal cycle analysis of potential benefits from subcooling control in heat pump

To evaluate the potential improvement from subcooling in heat pump systems the relative increase in specific capacities and work must be defined based on the T-h diagram showed in figure 1. Equations 1, 2, 3, 4 and 5 show the relative change in specific cooling capacity, specific heating capacity, specific work, COP and HPF, respectively.

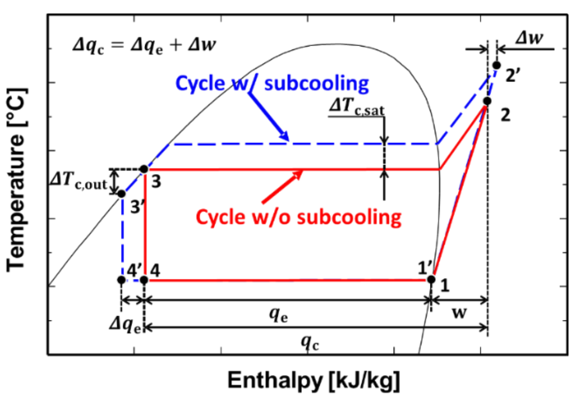

Figure 1: Subcooling boosts specific capacity, q_e, and specific work, w, at different rates leading to a possible efficiency maximum

$$\frac{\Delta q_e}{q_e} = \frac{h_3 - h_{3'}}{h_1 - h_3} \tag{1}$$

$$\frac{\Delta q_c}{q_c} = \frac{\Delta q_e + \Delta w}{q_e + w} = \frac{h_3 - h_{3'} + h_{2'} - h_2}{h_2 - h_3} \tag{2}$$

$$\frac{\Delta w}{w} = \frac{h_{2'} - h_2}{h_2 - h_1} \tag{3}$$

$$\frac{\Delta COP}{COP} = \frac{COP'}{COP} - 1 = \frac{\frac{q_e + \Delta q_e}{w + \Delta w}}{\frac{q_e}{w}} - 1 = \frac{1 + \frac{\Delta q_e}{q_e}}{1 + \frac{\Delta w}{w}} - 1 \tag{4}$$

$$\frac{\Delta HPF}{HPF} = \frac{HPF'}{HPF} - 1 = \frac{\frac{q_c + \Delta q_c}{w + \Delta w}}{\frac{q_c}{w}} - 1 = \frac{1 + \frac{\Delta q_c}{q_c}}{1 + \frac{\Delta w}{w}} - 1 \tag{5}$$

This analysis builds on the work from Carvalho and Hrnjak (2020b) which evaluated the potential improvement in HPF for different refrigerant. To compare the potential of subcooling control in cooling vs. heating operation it's important to note that if there exists an increase in COP coming from subcooling then equation 6 must be true.

$$\frac{\Delta q_{\mathrm{e}}}{q_{\mathrm{e}}} > \frac{\Delta w}{w} \tag{6}$$

And if equation 6 holds true and for $q_{\mathrm{e}} > w$ then mathematically equation 7 is also true, meaning that for the same operating conditions the relative improvement in COP will be greater than in HPF because the baseline specific heating capacity is always greater than specific cooling capacity.

$$\frac{\Delta q_{\mathrm{e}}}{q_{\mathrm{e}}} > \frac{\Delta q_{\mathrm{c}}}{q_{\mathrm{c}}} \therefore \frac{\Delta COP}{COP} \geq \frac{\Delta \mathrm{HPF}}{HPF} \tag{7}$$

Table 1 shows the calculated results for this analysis assuming an increase of 6 K in subcooling results in a 1 K boost in condensation temperature. The initial condensation temperature and fixed evaporation temperature were set at 45°C and 5°C, respectively. The results corroborate the mathematical assumptions that the relative improvement in HPF is lower or equal to its COP counterparts. The main dependency for improvements in efficiency by increasing subcooling is the ratio of enthalpy of vaporization to liquid isobaric specific heat, defining the baseline specific cooling capacity vs the rate of decrease in evaporator inlet enthalpy promoted by subcooling.

Table 1: Ideal cycle analysis of performance improvement from subcooling for a heat pump

Refrigerant	$c_{\mathrm{pl,c}}$	$h_{\mathrm{fg,e}}$	$h_{\mathrm{fg,c}}$	$\frac{h_{\mathrm{fg,e}}}{c_{\mathrm{pl,c}}}$	$\frac{\Delta w}{w}$	$\frac{\Delta q_{\mathrm{e}}}{q_{\mathrm{e}}}$	$\frac{\Delta q_{\mathrm{c}}}{q_{\mathrm{c}}}$	$\frac{\Delta COP}{COP}$	$\frac{\Delta \mathrm{HPF}}{HPF}$
[-]	[kJ kg^{-1}K^{-1}]	[kJ kg^{-1}]	[kJ kg^{-1}]	[K]	[%]	[%]	[%]	[%]	[%]
R717	5.0	1244	1076	232	2.5	2.8	2.8	0.3	0.3
R22	1.4	201	161	131	2.4	5.4	5.0	2.9	2.6
R32	2.3	307	224	116	2.5	5.7	5.4	3.2	2.9
R600a	2.6	350	305	127	2.2	6.1	5.5	3.8	3.3
R454B	2.1	254	185	104	2.4	6.7	6.2	4.2	3.7
R134A	1.5	195	158	115	2.2	6.6	6.0	4.3	3.6
R290	3.0	368	296	111	2.2	6.8	6.2	4.5	3.8
R407C	1.7	205	157	105	2.3	7.0	6.4	4.6	4.0
R410A	2.1	215	148	88	2.4	8.0	7.3	5.6	4.9
R1234yf	1.5	160	127	95	2.1	8.5	7.5	6.3	5.3

Figures 2 and 3 show the effect of condensation and evaporation temperatures on the relative change in HPF for the refrigerants listed in table 1. R410A and R1234yf show the highest potential in both cases with R1234yf showing high improvement for condensation temperatures below 52°C. Since R1234yf has a higher $\frac{h_{\mathrm{fg,e}}}{c_{\mathrm{pl,c}}}$ than R410A other factors have some effect on the potential HPF improvement from subcooling, such as isentropic compression curves and the shape of the saturation bell.

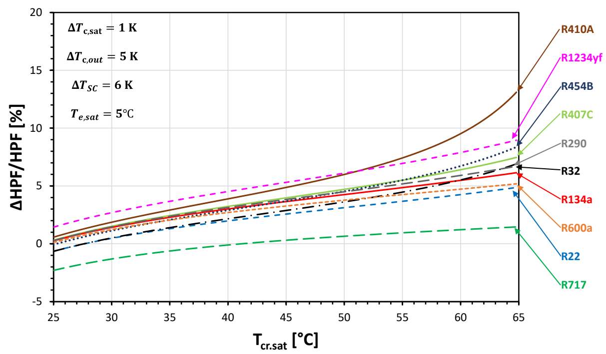

Figure 2: Relative changes HPF for several refrigerants at a fixed $T_{e,sat}$

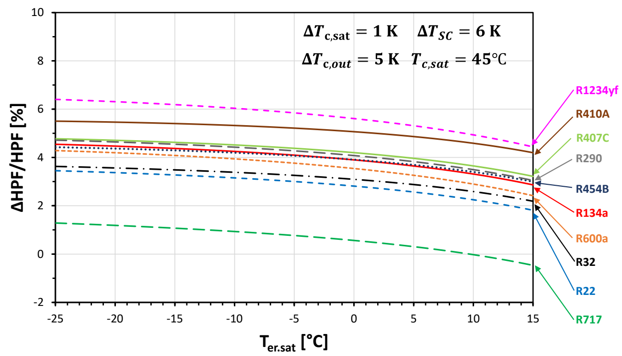

Figure 3: Relative changes in HPF for several refrigerant at a fixed $T_{c,sat}$

3. FACILITY

Two environmental chambers simulate indoor and outdoor conditions and can keep temperatures within ± 0.5°C and absolute humidity ±2%. Compressor, heaters and blowers power measurement is within ±0.2% and the expanded uncertainty for air-side and refrigerant side capacity calculations of approximately ±4%. Heating performance expanded uncertainty is estimated to be around ±5%.

Two off-the-shelf reversible residential 2 Ton (7kW) R410-A systems with a round-tube A-coil indoor heat exchangers and a round-tube horizontal condensers were used in this investigation. The High-SEER system has a variable capacity (30-117 Hz) scroll compressor with a displacement volume of 21.5 cm^3 using an electronic expansion valve (EXV) for its original heat pump setup. The Low-SEER system uses a fixed speed (58.33 Hz) scroll compressor with a displacement volume of 19.0 cm^3 and an orifice piston tube as its heat pump expansion device. Separate EXVs were installed on the outdoor units to allow full control over the expansion process. The charge compensator in the High-SEER system was closed off to prevent it from destabilizing subcooling control. More information and schematics of this facility, and specifications for the indoor and outdoor heat exchangers for both systems are available in the conference paper by Carvalho and Hrnjak (2022) (ID 2282). The High-SEER system has indoor and outdoor heat exchanger with an air-side surface area 4.5 and 5.5 times that of the Low-SEER systems HXs.

4. RESULTS AND DISCUSSION

4.1 Charging and evaluation methods

The concept behind subcooling control is to simply install an electronic expansion valve in the system and control its subcooling by varying the valve opening to achieve maximum COP or HPF. The system also requires an accumulator large enough to account for the charge migration as subcooling is decreased. Ideally the system should always operate just below saturation at the evaporator outlet improving the overall heat transfer coefficient on the evaporator while the subcooling is controlled and the accumulator holds the excess charge. All charging procedures used the AHRI 210/240 A condition (26.7°C DB, 19.4°C WB indoor / 35°C DB, 23.9 °C WB outdoor). The High-SEER baselines was charged until the subcooling measured around 5 K, while the evaporator superheat should be between 3-5 K, while the Low-SEER baseline was charged following the manufacturer's recommended values based on the length of the liquid lines.

The charging procedure in subcooling control is slightly different. The objective in this case is to have enough charge so that under any condition the system will operate at its COP/HPF-maximizing subcooling while simultaneously having saturated evaporator outlet or a liquid level in its accumulator. Ideally as charge is added to the system the COP-maximizing subcooling should be determined and this step should be repeated until a plateau in COP is found. This should happen because any further addition in charge is directly stored in the accumulator which should keep the evaporator pressure almost constant and lead to no change in the COP. The final refrigerant charges determined for each system are shown in table 3. The subcooling controlled high-SEER and low-SEER systems operate with 28.8% and 15.6% more charge than their baseline counterparts. This is due to the accumulators being able to hold 5500 g and 2450 g of saturated liquid at 0°C, for the high-SEER and low-SEER systems, respectively.

Table 3: Refrigerant charges for each system tested

	High-SEER system baseline (H/P-original EXV)	High-SEER system SC control (H/P-EXV)	Low-SEER system baseline (H/P-piston tube)	Low-SEER system SC control (H/P-EXV)
Refrigerant charge [g]	**6600 g**	**8500 g**	**3288 g**	**3800 g**

To evaluate the HSPF improvement obtained by subcooling control the AHRI 210/240 heating conditions were used. The conditions H1 (rating) and H3 (low temperature) were used for individual performance comparison between systems. The indoor air flow rate for the High-SEER and Low-SEER systems at these conditions are 900 and 870 CFM, respectively. At each condition the electronic expansion valve will be used to sweep through the subcooling and define the COP/HPF-maximizing subcooling as well as observe how the systems behave as the condenser subcooling is varied with a fixed refrigerant charge. The results can be then compared to the baseline systems using their respective refrigerant charges.

4.2 Effect of subcooling control on the heat pump performance of a low and high SEER system

The performance characteristics for the H1 condition (21.1°C DB, 15.5°C WB indoor / 8.3°C DB, 6.1 °C WB outdoor) are shown in figure 5, with data for the HPF, heating capacity and compressor power. The high-SEER system uses an electronic expansion valve with its proprietary control that seems to focus on the compressor suction superheat, while the low-SEER system has an orifice piston tube designed to operate well in this condition. Figure 5 shows that the low-SEER system achieves an improvement of 6.2% and can match the HPF of the high-SEER system albeit at reduced capacity (-9.5%).

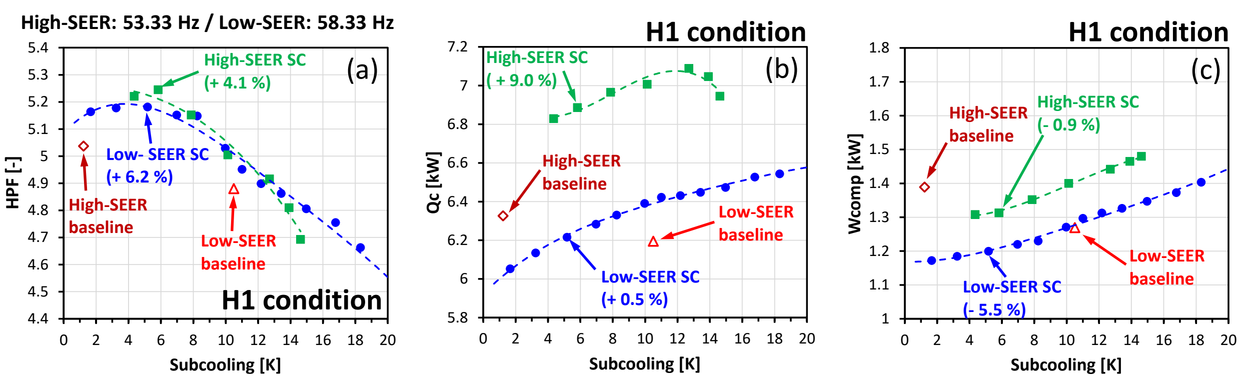

Figure 5: HPF (a), Q_c (b) and W_{comp} (c) results for the A/C H1 condition

Both baselines have focused on keeping the compressor suction superheat around 3-5 K, but the high-SEER system provides very minimal subcooling possibly due the use of a charge compensator which should remove some of the active charge in the system when running in heating operation since the system is usually charge optimized for air conditioning applications. The systems also show very close HPF-maximizing subcooling for this condition indicating that the efficiency-maximizing subcooling is directly affected by the capacity as well as the condenser size. Figure 6 shows the T-h diagrams for both results and the ΔT_{cra} (equation 8) values which are within approximately 2 K of each other indicating why both systems have similar HPF-maximizing subcooling. The trend here is less clear as the high-SEER system shows a lower ΔT_{cra} but has an overall higher HPF-maximizing subcooling.

$$\Delta T_{cra} = T_{cr,sat} - T_{cai} \tag{8}$$

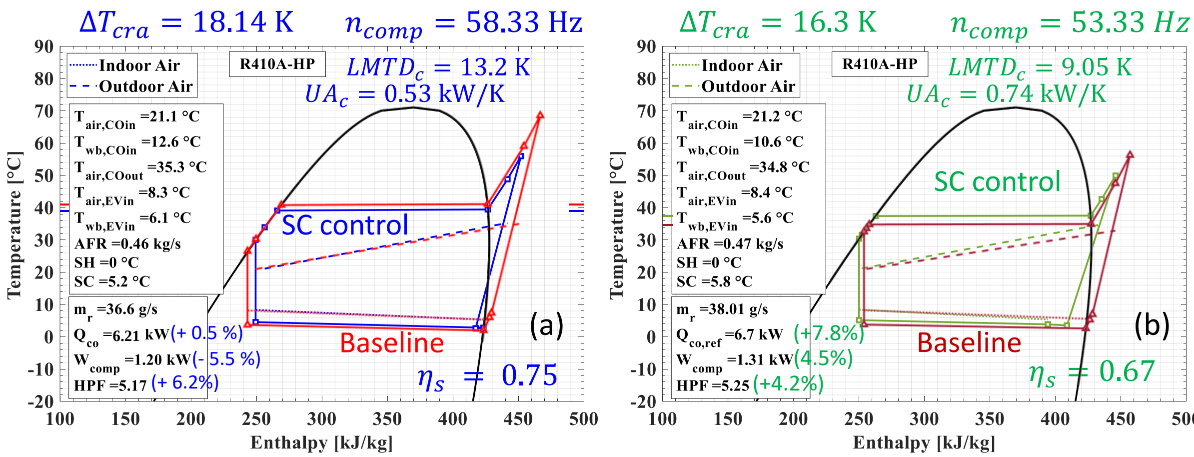

Figure 6: T-h diagrams for the low-SEER system (a), high-SEER system (b), baselines vs. COP-maximizing SC for the H1 condition

In both cases the evaporation temperatures between baselines and subcooling-controlled points are within 1 K of each other, meaning the improvement from properly controlling subcooling is providing most of the HPF increase for the H1 condition.

To investigate how subcooling control can perform at low outdoor temperature condition both systems were run at H3 with the results shown in figure 7. This condition is interesting as the main benefit from running with subcooling controls shifts from just increasing HPF to boosting the capacity to reduce the auxiliary heating required. The relative improvement in HPF is diminished in this case when compared to H1 possibly due to the higher latent heat of vaporization on the evaporator side, which according to the approximation shown in equation 9, defined by Pottker and Hrnjak (2015a), can lead to a decrease in performance benefit from subcooling.

$$\frac{\Delta q_e}{q_e} \cong \frac{\Delta T_{c,out}}{\frac{h_{fg,e}}{c_{pl,c}} + (T_c - T_e)_{sat}} \tag{8}$$

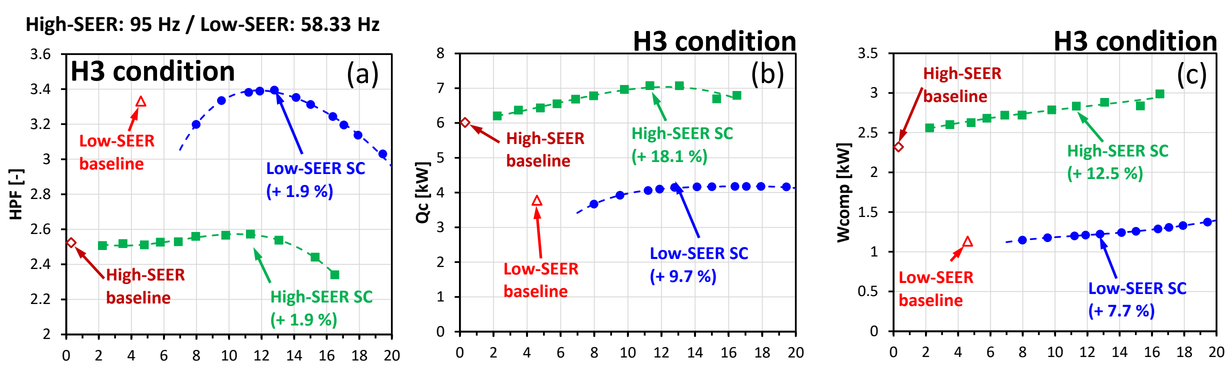

Figure 7: HPF (a), Q_c (b) and W_{comp} (c) results for the A/C H3 condition

Figure 8 shows the T-h diagrams for both systems with baselines and HPF-max. subcooling. At this condition due to decreased isentropic efficiencies the potential for subcooling control improvement is reduced. Both systems show only +1.9% increase in HPF. But on the other hand, there's an increase in 9.7% and 18.1% in capacity for the small and large systems, respectively. Also, in this condition both system's expansion devices cannot properly keep the evaporator superheat as the charge selected based on the air-conditioning operation is too high for their baseline expansion device setups. HPF improvement is mostly coming from condenser subcooling here since evaporator operation is nearly identical between baselines and subcooling controlled. The High-SEER has more potential for performance improvement by increasing its compressor speed, but it's restricted by the discharge temperature to prevent oil degradation.

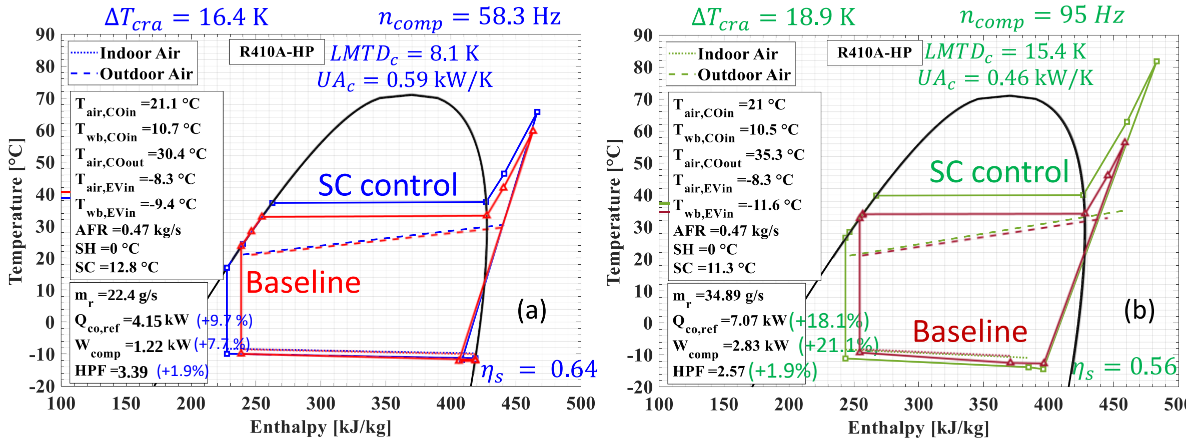

Figure 8: T-h diagram comparing the baseline system with a subcooling control-based system for $H1_{Low}$

Following AHRI 210/240 the seasonal heating performance factor (HSPF) for region IV is calculated and shown in table 4 for both high and low SEER systems tested. Subcooling control can achieve a much higher improvement in the high-SEER system. This is mainly due to the requirement of a partial load condition $H0_{Low}$ (16.7C DB, 13.6C WB outdoor; 21.1C DB, 15.5C WB indoor), shown in figure 9, for variable capacity systems. In this condition the HPF-maximizing subcooling along with no dryout in the evaporator increases the HPF by 19.1% over the baseline. This indicates that even though partial load condition usually needs low COP/HPF-maximizing subcooling the actual improvements can be very significant and have a big impact on the seasonal performance of the system. Although further investigation should be done since most of the improvement comes from an increase in heating capacity which would lead to more frequent on/off cycling to match the partial load at this condition, possibly rendering the use of subcooling control unfeasible.

Table 4: SEER and HSPF results for baseline and SC-controlled low-SEER and high-SEER systems

HSPF			
Baseline High-SEER	**SC control High-SEER**	**Baseline Low-SEER**	**SC control Low-SEER**
12.0	**14.3 (+19.2%)**	**8.5**	**9.4 (+10.6%)**

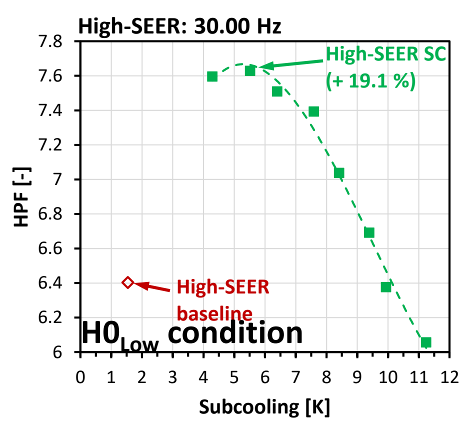

Figure 9: HPF results for the H/P $H0_{Low}$ condition

4.3 Subcooling control scheme for heat pumps

The HPF-maximizing subcooling values for both systems are used to define a control curve following the scheme developed and previously evaluated in Carvalho and Hrnjak (2020b). Figure 10a shows the HPF maximizing subcooling curve as a function of ΔT_{cra}, shown in equation 9. The heat pump control function cannot be properly described by this scheme, specifically for the Low-SEER system. Possible causes for this are the large variation in pressure ratios, ranging from 1.8 to 5.0, and the Low-SEER system presented distribution issues in its evaporator when the EXV was installed upstream of the distributor to control subcooling. A control curve may be better defined as a function of the pressure ratio, as shown in figure 10b, with an improved agreement with the data. Further investigation on subcooling control curves for the heating operation must be performed to define a cost-effective solution.

$$\Delta T_{SC}\left(HPF_{maximizing}\right) \cong a\Delta T_{cra}\ (K) + b \quad (6)$$

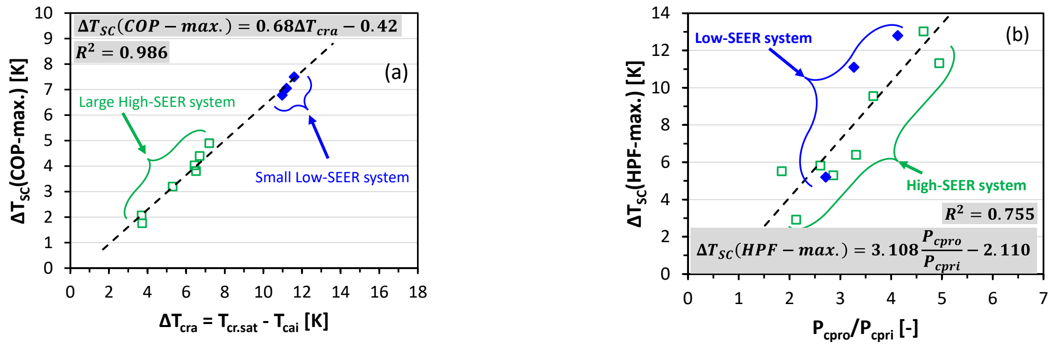

Figure 10: Heat pump subcooling control scheme based on experimental data

4.4 System model validation of experimental results

Overall, the use of subcooling control in both systems showed similar performance improvement at the rating condition H1 and low-temperature condition H3. The HPF-maximizing subcooling values also weren't significantly affected by the discrepancy in heat exchanger sizes between both systems. To further confirm this conclusion a system model was developed python using a CoolProp (Bell et al, 2014) wrapper with REFPROP (Lemmon et al 2018). The model uses an ε-NTU method for the heat exchangers and a 37-coefficient semi-empirical compressor correlation for the High-SEER variable capacity compressor with Rice and Dabiri's (1981) correction. Table 5 shows the correlations used in the system model.

Table 5: Correlations used in the system model

System fluid / components		Correlations
Refrigerant	Single phase HTC	Gnielisnki, 1976
	Condensation HTC	Dobson & Chato, 1998
	Evaporation HTC	Gungor & Winterton, 1986
	Single phase DP	Churchill, 1977
	Two-phase DP	Friedel, 1979
	Void fraction	Rouhani & Axelsson, 1970
Air	Air-side fin-tube HTC	Chang and Wang, 2000
	Air-side fin-tube DP	Chang and Wang, 2000

Using the model, the effect of heat exchanger size on the HPF improvement from subcooling control was evaluated. Figure 11 shows the HPF results for the model at the rating condition H1 using the High-SEER system compressor model with speed control to keep the heating capacity fixed. The air flow rates through indoor and outdoor heat exchanger matched each systems rate values. The results show that the High-SEER system on average twice the improvement when comparing COP-maximizing value at xero = 0.92 to ΔT_{SC} = 0 K at SHero = 4 K. But most of the improvement in this case is from the improvement in evaporator performance, since with a larger evaporator the air-side bottle neck is reduced giving more potential for improvement to the High-SEER system. If the baseline here is set as ΔT_{SC} = 0 K at xero = 0.92 then the relative improvement from subcooling control is greater by a small margin for the Low-SEER system at 3 kW and 5 kW, and comparable at 7 kW. These latter results indicate that the effect of heat exchanger size in residential system on the performance improvement solely from subcooling control is not significant, corroborating the experimental data for both H1 and H3 conditions. Overall, because most system's charge is not optimized for heating operation, the improvement from subcooling control is substantial for both large and small HX systems.

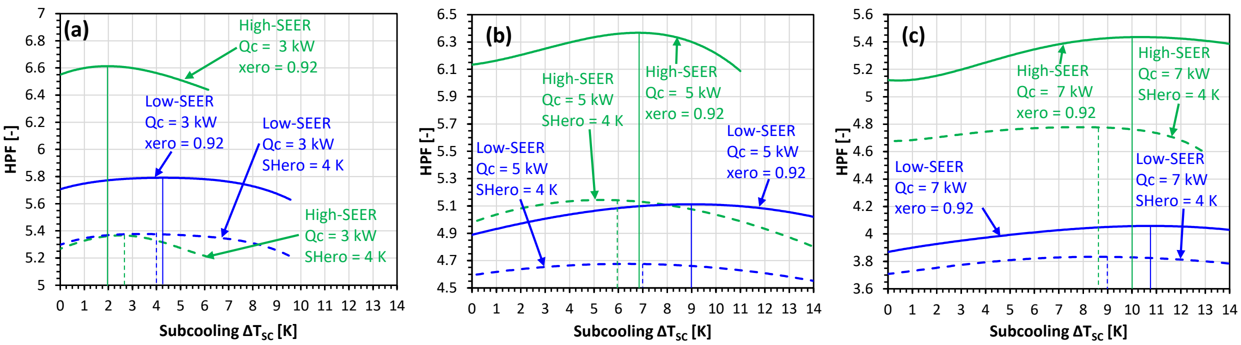

Figure 11: System model results on the effect of subcooling control at 3 kW (a), 5 kW (b) and 7 kW (c) of heating capacity for the High-SEER and Low-SEER system heat exchangers using the same compressor

5. CONCLUSIONS

This paper evaluated the performance of two different residential heat pump system using subcooling control. The results show an increase in HPF of +4.1% and +6.2% for the for the High-SEER and Low-SEER systems at H1, respectively. The HSPF improvement for the High-SEER and Low-SEER systems were of +19.2% and +10.6%, respectively. Although the greater increase in efficiency of the High-SEER system is attributed to the increase in HPF for the partial load condition which was not used for the Low-SEER system's calculations.

The control curve previously proposed by Carvalho and Hrnjak (2020b) does not agree for both systems. Two possible reasons for the lack of agreement, specifically for the Low-SEER system are: the wide range of pressure ratios, from 1.8 to 5.0; and the maldistribution in the evaporator introduced after installing the EXV for subcooling control upstream of its distributor. An implementation of subcooling control based on the pressure ratio provides better agreement, but its application is not cost effective for residential systems. Further investigation through modeling is required to define if using ΔT_{cra} as a control parameter is possible if the EXV does not affect any component's performance.

A system model was validated using the experimental data, and its results show that a decrease in condenser size is not strongly correlated to an increase in the potential improvement in performance from subcooling control.
Based on the experimental data from both systems evaluated in this study the use of subcooling control shows potential to provide significant improvement in HPF for most residential reversible systems because the refrigerant charge is generally optimized for cooling operation as opposed to the heat pump cycle.

NOMENCLATURE

A/C	Air conditioning		**Subscript**	
COP	Coefficient of performance	(–)	a	air (subscript)
Cp	Specific heat capacity	(kJ kg^{-1}K^{-1})	c	condenser (subscript)
DB	Dry bulb		e	evaporator
EXV	Electronic expansion valve		(subscript)	
h	Enthalpy	(kJ kg^{-1})	fg	vaporization
H/P	Heat pump		(subscript)	
HPF	Heating performance factor	(-)	i	inlet (subscript)
HSPF	Heating seasonal performance factor		l	liquid (subscript)
m	Mass flow rate	(kg s-1)	o	outlet (subscript)
P	Pressure	(kPa)	r	refrigerant
q	Specific cooling capacity	(kW kg^{-1})	(subscript)	
Q	Capacity	(kW)	sat	saturation (subscript)
SC	Subcooling	(°C or K)	SC	subcooling
T	Temperature	(°C or K)	(subscript)	
TXV	Thermostatic expansion valve		v	vapor (subscript)
w	Specific work	(kW kg^{-1})		
W	Work/Power	(kW)		

WB	Wet bulb

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