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ABSTRACT

Building heating requirements increase with decreasing ambient temperature, while the coefficient of performance of air-source heat pumps (ASHPs) shows the opposite trend. Additionally, heat pump heating capacity decreases with ambient temperature, which leads to the utilization of inefficient electric auxiliary heat below the design point. Increasing the capacity and coefficient of performance (COP) at lower ambient temperatures is important for improving the market penetration of heat pumps in climates having significant operating time at low ambient temperature. Simulation studies previously showed that compressor vapor injection leads to an increase of COP under exactly those conditions. Furthermore, reduced capacity degradation towards smaller ambient temperatures was predicted.

The work presented in this paper shows experimental results obtained from a commercially available 5-ton heat pump that was retrofitted with a two-port vapor injected scroll compressor. The injection ports within the two compression pathways are located in the fixed scroll with different distance from the suction chamber. The vapor for the two injection pressure levels was generated using two vapor separators in a cascade configuration. This configuration made it necessary to not only control the superheat but also the liquid levels in the separators and subcooling of the refrigerant leaving the condenser.

Baseline performance data of the heat pump without vapor injection was obtained and compared with that for the vapor injection and other system configurations. For the baseline, the injection lines to the compression pockets were plugged within the fixed scroll to reduce dead volume and re-expansion losses. Also, the vapor-separator section was shut off and bypassed. In the second step, the plugs were removed and a staged expansion process was performed using the separator section. The generated vapor from each separator was injected into the respective compressor port causing an intercooling effect on the compression process.

With identical compressor speed, a 28% improvement in capacity was achieved at the 8.33°C design point, when compared to the baseline without vapor injection. When the baseline and vapor injected system capacity were matched by adjusting compressor speed, the COP increased by up to 6% at -8.33°C. Results of a bin-type analysis of the experimental results predicts an improvement in the heating seasonal performance factor (HSPF) of 6% for Minneapolis and nearly 7% for ANSI/AHRI climate region 5. Further details on this can be found in the companion paper (Bach et al. 2014b), published in this conference.

1. INTRODUCTION AND MOTIVATION

Bertsch and Groll (2005) summarized the main problems in applying heat pumps in northern cold climates: 1) The adverse trend of increasing building heating demand and decreasing heat pump heating capacity with decreasing ambient temperature. 2) Increasing compressor discharge temperatures with decreasing ambient temperatures due to increasing pressure ratio, which ultimately makes it necessary to shut down the heat pump to prevent lubrication oil degradation. 3) Signification reduction of system efficiency at high pressure ratios. 4) On/off cycling of the heat pump at moderate ambient temperatures when the unit is sized for low ambient temperatures, which reduces the lifespan of the compressor, the overall system efficiency, and the comfort level of inhabitants.

Some approaches to overcome low ambient temperature limitations include vapor, liquid, and oil injection into the suction or during the compression process and the usage of cascade systems. Several system configurations were investigated by Bertsch and Groll (2005) with the three most promising identified as 2-stage approaches using conventional compressors: the cascade cycle, the intercooler cycle, and the economizer cycle. The economizer cycle was chosen for the development of a prototype heat pump having a 17 kW heating capacity at -10°C ambient temperature. This breadboard system featured two single speed scroll type compressors, which are connected in series. Refrigerant vapor from the economizer[1] is mixed with the discharged refrigerant from the low stage compressor. At medium ambient temperatures, only the high stage cycle operates to achieve better part load efficiency. Caskey *et al.* (2012) continued the study of the economizer cycle and designed two prototype set-ups (18.34 kW heating capacity/-20°C ambient temperature), which were tested in a field demonstration. A variable speed high stage compressor was used to better match the part load requirements. Their simulation study predicted 30% primary energy savings if the existing natural gas furnace were replaced with the cold climate heat pump. In the actual field test, the system controls were continually improved which meant that the heat pump did not operate at its optimum efficiency. A seasonal COP of 2.3 was predicted based on analysis of the field test data. This corresponds to a 19% savings of primary energy compared to the natural gas furnace (Hutzel and Groll, 2013).

With the new concept of vapor injected scroll compressors, a less expensive approach to establish a cold climate residential heat pump is possible. Bell *et al.* (2013) performed a theoretical analysis of the vapor injected scroll compressor to be used in a cold climate heat pump. His simulation study predicted an efficiency improvement of 10% and 16% at -20°C evaporation temperature by using one or two injection lines, respectively.

One of the key components of a cold climate heat pump is the multi-circuit evaporator that is located in the outdoor unit. Even small deviations from its ideal operation can cause signification degradation of capacity and efficiency of the unit. Air side maldistribution, refrigerant side maldistribution, evaporator fouling, and frost build-up lead to those penalties. Bach (2014) summarizes various studies on the influence of refrigerant and air flow maldistribution on the heat pump's performance. The study by Payne and Domanski (2002), for example, is cited that demonstrates a capacity reduction of up to 41% for non-uniform air flow tests of evaporators. However, by controlling the refrigerant flow in each circuit to provide the original exit superheat, the capacity could be recovered to within 2% of the initial value. Kærn *et al.* (2011a) investigated air flow maldistribution and found a decrease in COP of up to 43%. Consistent with previous work, Kærn *et al.* (2011b) confirms that most of the penalty in capacity and COP can be recovered if the individual exit superheat is forced to a uniform value. Kim *et al.* (2008) introduced an approach to control the individual circuit superheat, consisting of a primary expansion valve which provides most of the pressure drop, and individual circuit flow balancing valves for superheat control. He found that this hybrid control method was more effective if the flow balancing valves were located upstream of the evaporator. This approach has been applied by Bach (2014) to four different vapor compression systems, showing small COP and capacity improvements for conditions without airside maldistribution. If airside maldistribution was applied in the form of evaporator frosting (heat pump (HP), walk in refrigeration system (WCRS)) or airside maldistribution (blockage for HP, cold climate HP, and WCRS and temperature and flow maldistribution for airside economized rooftop air conditioning unit), COP and capacity improvements in excess of 10% were observed for some operating conditions. In fact, in order to find a compromise between the benefit of the individual circuit control and its costs, a reduced hybrid control method was developed. This approach pairs two neighboring circuits using a secondary distributor with one balancing valve located between that distributor and the primary distributor. A short summary of its layout can be found in Bach *et al.* (2014a).

[1] The economizer in that setup was used to further subcool the refrigerant.

This paper presents findings from the experimental cold climate heat pump set-up shown in Bach (2014) that has the ability to run either in single stage, flash gas bypass, or two-stage vapor injection mode combined with the reduced hybrid control scheme.

2. EXPERIMENTAL SET-UP

2.1 Operations modes and key components

The heat pump is a split air-to-air system with an outdoor unit and indoor unit. The indoor unit contains the AC-mode expansion device, the heat exchanger, and the indoor blower. The outdoor unit contains the outdoor heat exchanger and blower motor, the compressor, vapor separators, and control valves necessary to facilitate the different operating modes of the system. Only heating mode operation was considered during the testing. The compressor is operated on a variable speed drive, allowing a closer match between heating requirement of the building and the capacity of the heat pump. For single stage operating mode, figure 1 (B0), the vapor separators are bypassed by an electronic expansion valve (EXV). The refrigerant is evaporated and superheated in the outdoor heat exchanger, then passes through the 4-way valve and accumulator (acc) to the compressor suction. The refrigerant is compressed by the compressor and passes through discharge muffler (M) and the 4-way valve to the indoor heat exchanger, where it is condensed and subcooled. The subcooled refrigerant travels through the bypass valve in the thermostatic expansion valve (TXV) and the filter drier back to the EXV. In flash gas bypass operating mode (B0 FGB), figure 2, the flash gas from the expansion process is taken off using the low pressure (LP) separator before the evaporator and bypassed through a control valve directly to the accumulator while the liquid refrigerant is drained to the evaporator.

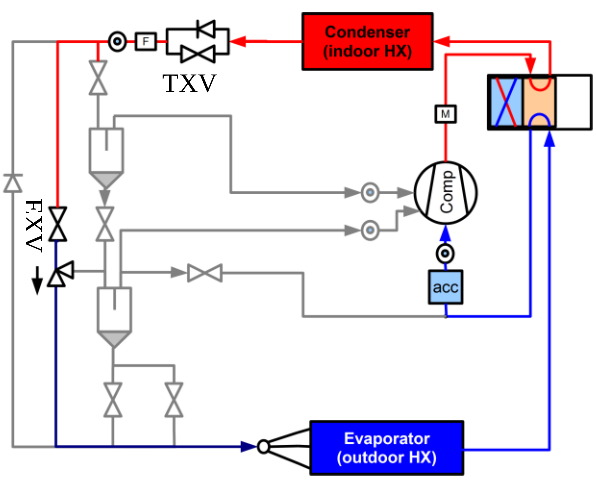


Figure 1: Single stage operating mode (B0)

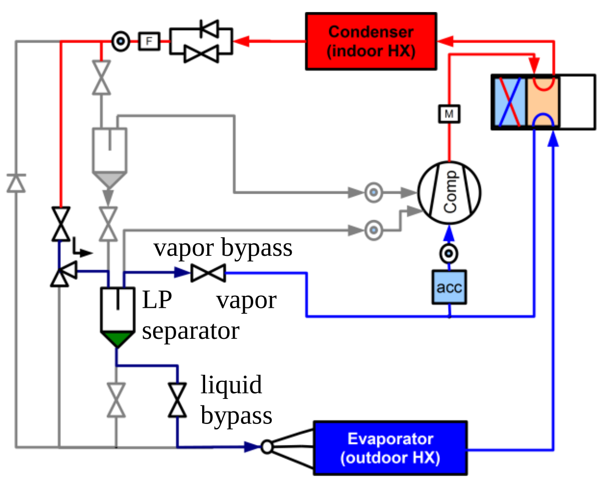


Figure 2: Flash gas bypass operating mode (B0 FGB)

The injection ports of the compressor are internally plugged for both the B0 and B0 FGB configurations. This was done to reduce re-expansion losses. These plugs are removed for the vapor injected configuration (B1), Figure 3. In that configuration, the expansion process is split up into three stages, where the flash gas from the high pressure and intermediate pressure expansion is injected into the injection ports of the compressor. For the vapor injected mode with hybrid control, as indicated in figure 4, the last expansion process is done using 5 balancing valves, where each valve controls the superheat of a neighboring circuit pair of the outdoor heat exchanger. This approach, named reduced hybrid control, reduces the number of balancing valves when compared to hybrid control as introduced by Kim *et al.* (2008) by 50% for an even number of circuits. The tested heat pump additionally used 2-step balancing valves, which are expected to be cheaper to produce than electronic expansion valves.

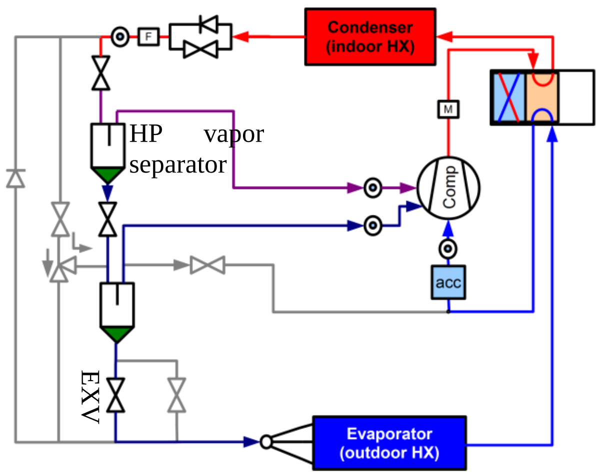


Figure 3: Vapor injection operating mode (B1)

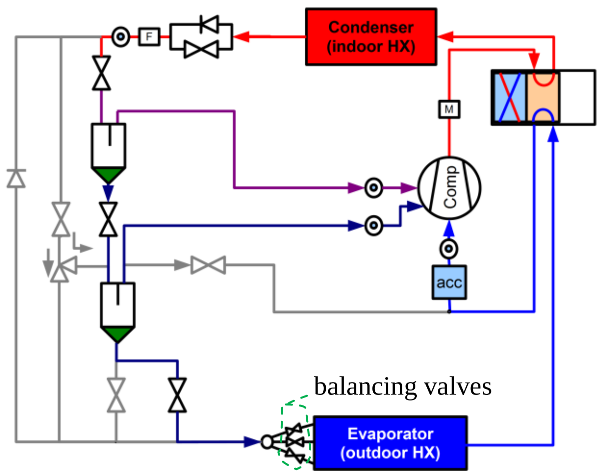


Figure 4: Vapor injection operating mode (B1H)

2.2 Instrumentation

Refrigerant inlet and outlet temperatures were measured for all major components. Air inlet and outlet temperatures were measured using thermocouple grids for both indoor and outdoor units. Inlet dew point was measured for the indoor and outdoor units with the outlet dew point measured at the outdoor unit. Chilled mirror sensors were used for all dew point temperature measurements. The relative humidities at the air inlet of the indoor and outdoor units were measured as a backup in case of sensor failure. The air-side flow rate of the indoor unit was measured using an ASHRAE nozzle box that follows ASHRAE 41.2 (ASHRAE, 1987). Table 1 and 2 list the measurement uncertainty of the employed measurement devices in terms of absolute uncertainty and relative uncertainty with respect to the measured value. Figure 5 shows the system level instrumentation.

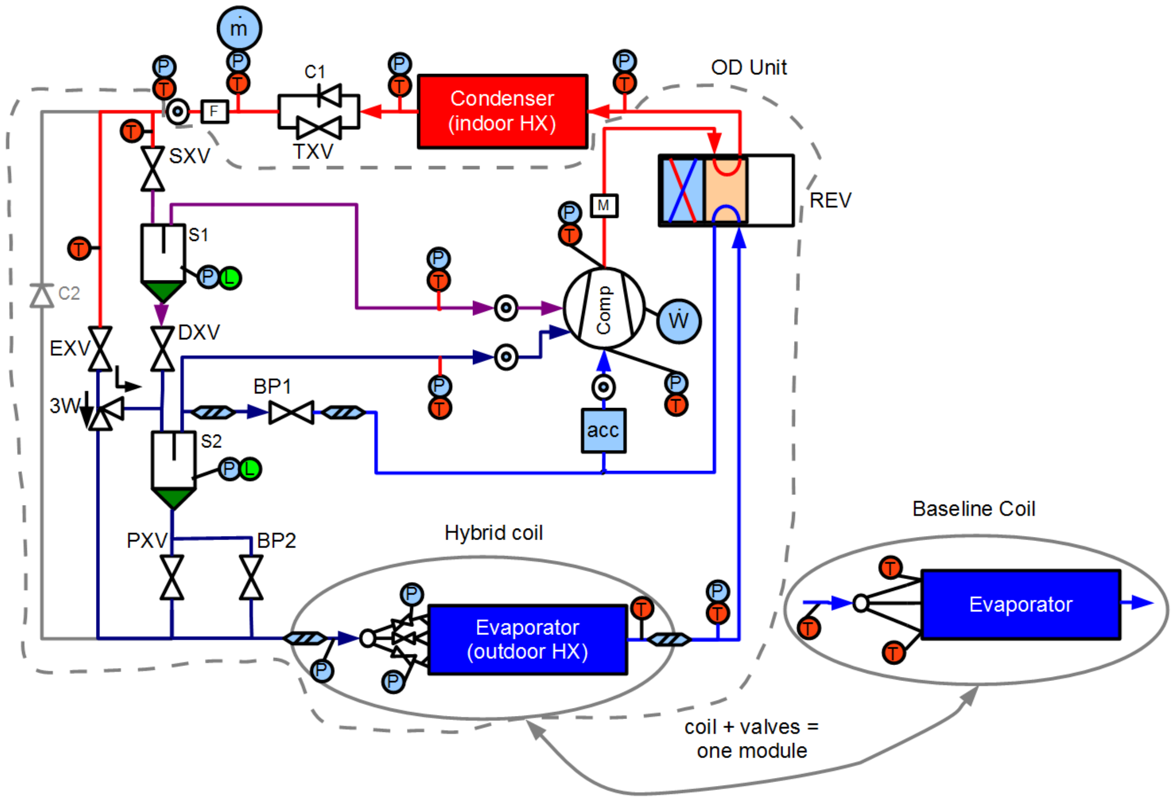
Table 1: Measurement device uncertainty for temperature and humidity measurements[2]

Measurement Device	Absolute Uncertainty
Thermocouples (TC) using internal cold junction	1.12 K
TC with external cold junction for increased accuracy	0.56 K
RTD	0.15 K
Dew point	0.2 K
Relative humidity	3%

Table 2: Measurement device uncertainty for pressure, flowrate and power measurements

Measurement Device	Absolute Uncertainty	Relative Unc.
Note: next 3 items are gage pressure transducers		
High pressure (S1, high pressure VI port, comp. discharge)	9.0 kPa	
Low pressure (distributor and circuit inlets, comp. suction)	2.2 kPa	
Medium pressure (remaining locations)	4.5 kPa	
Differential pressure (indoor unit static pressure)	2.5 Pa	
Differential pressure (flow measurement nozzle pressure drop)	6.2 Pa	
Atmospheric pressure	0.12 kPa	
Refrigerant mass flow rate		0.5 %
Power, fan		0.2 %
Power, compressor	26.3 W	
Power, VSD	5 W	0.2 %

[2] By applying external cold junction devices the thermocouple accuracy used for discharge temperature, as well as condenser inlet and outlet temperature has been improved.



Legend

Valves
- BP1 – Vapor bypass valve (active in baseline only)
- BP2 – Liquid bypass valve (active in baseline only)
- BP3 – Injection bypass valve (needed when 2^{nd} port open)
- EXV – Primary expansion valve, baseline
- DXV – Drain expansion valve
- PXV – Primary expansion valve, VI-system
- SXV – Supply expansion valve
- TXV – Thermostatic expansion valve
- 3W – 3-way valve
- REV – Reversing valve
- C1 – TXV bypass check valve
- C2 – S1/S2 bypass check valve

Other
- S1 – High pressure vapor separator
- S2 – Low pressure vapor separator

Measurement points and components
- T Temperature measurement point (immersion TC)
  125-type TC for in-unit vapor measurements
- P Pressure measurement point (capillary tube)
- L Liquid level measurement
- $\dot{W}$, $\dot{m}$ Power, mass flow
- Flared connection or rotolock
- F, M Filter/Drier, Muffler

Figure 5: System instrumentation

2.3 Controls and Observation

The different operating modes lead to different methods for controlling the system. For the B0 configuration, superheat was the only controlled variable; subcooling was between 4 and 5 K for clean coil operating conditions. For the B0 configuration with flash gas bypass, subcooling was additionally controlled. For the vapor injected configurations, liquid levels in the two separators were controlled to allow the charge in the system to balance. Superheat for the vapor injected system was either controlled by a single valve or, in case of the hybrid control scheme, by the balancing valves. The balancing valves were used to equalize the superheats and to move the overall superheat to the target value. The setpoint for both superheat and subcooling was 5 K for all system configurations. It was necessary to increase the superheat setpoint for some operating conditions to maintain stable operation of the system.

3. METHODOLOGY

3.1 Test plan

Testing was conducted at test conditions similar to AHRI 210/240 (AHRI, 2008). Table 3 shows the resulting test plan. The following modifications were made:

- Relative humidity in the outdoor room was reduced for ambient temperatures that can lead to frost build up. This step was taken to reduce the penalty caused by frost built up – which is larger for the vapor injected system due to non-optimal system control.
- Tests with coil blockage of two different levels were conducted to simulate the effects of frost build up and/or fouling.
- An additional test (HX) was added to investigate low temperature performance of the heat pump.
- Since vapor injection leads to an increase in capacity, additional clean coil tests H2, H3, and HX were added where the compressor speed was adjusted to match the baseline systems capacity.

The COP and capacity results for the blocked coil tests are not part of this paper but can be found in Bach *et al.*(2014 a).

Table 3: Test plan

Test description	Air Entering Indoor Unit Temperature [°C]			Air Entering Outdoor Unit Temperature [°C]			Compressor Speed	Airside blockage
	Dry Bulb	Wet Bulb	Dew Point	Dry Bulb	Wet Bulb	Dew Point	[Hz]	-/light/severe
H1 - low - clean	21.1	≤15.6	≤12.06	8.33	6.11	3.74	40	-
H1 - low - light block	21.1	≤15.6	≤12.06	8.33	6.11	3.74	40	light
H1 - low - severe block	21.1	≤15.6	≤12.06	8.33	6.11	3.74	40	severe
H2 - mid - clean	21.1	≤15.6	≤12.06	1.67	min	min	55	-
H2 - mid - light block	21.1	≤15.6	≤12.06	1.67	min	min	55	light
H2 - mid - severe block	21.1	≤15.6	≤12.06	1.67	min	min	55	severe
H3 - full - clean	21.1	≤15.6	≤12.06	-8.33	min	min	70	-
H3 - full - light block	21.1	≤15.6	≤12.06	-8.33	min	min	70	light
H3 - full - severe block	21.1	≤15.6	≤12.06	-8.33	min	min	70	severe
HX - full - clean	21.1	≤15.6	≤12.06	-17.78	min	min	70	-
HX - full - light block	21.1	≤15.6	≤12.06	-17.78	min	min	70	light
HX - full - severe block	21.1	≤15.6	≤12.06	-17.78	min	min	70	severe

Notes:
> Tests to be repeated with each different system configuration.
> H2, H3, and HX clean coil test additionaly with reduced compressor speed to match baseline capacity.
> H1 test condition adopted from AHRI 210/240; H2, and H3 modified humidity compared to AHRI 210/240 (2008)

3.2 Test Procedure

Test data was taken under steady state operating conditions, e.g. no or only small trend in discharge temperature and all other temperatures and pressures. The start of the steady state period was judged during system operation, after the start of that period, at least 30 minutes of steady state data was taken. This resulted in 30 minutes or more of steady state data after the final data selection. The average absolute mismatch between useful airside and refrigerant side capacity was 2.2 %, with the maximum occurring value being 3.7%. Reported values in this paper are based on the refrigerant side, due to the better accuracy of these measurements.

3.3 Uncertainty Analysis

Uncertainty analysis was based on the method outlined in Taylor and Kuyatt (1994). The following contributions to the uncertainty were considered:

- Distribution of fluctuations of the measurement values in terms of the one sided 95% confidence interval of the one sided t-distribution.
- Sensor accuracy and propagation of sensor accuracy through calculated properties.
- Uncertainty of the property calculation routines (REFPROP, Lemmon *et al.*, 2007) was not considered.

The resulting uncertainty is shown in the figures of this document.

4. EXPERIMENTAL RESULTS

4.1 Comparison to Baseline

Figure 6 and 7 show the improvement in COP and capacity relative to the baseline system (B0). Vapor injection leads to significant improvement in capacity, with about 11% at high ambient temperature and 28% at -8.3°C ambient temperature. For the “fix capacity” case, compressor speed was reduced to match the baseline B0 capacity – with exception of the H1 test, where no further reduction of compressor speed was possible. The COP improvements are smaller – with identical compressor speed than for the baseline, up to 3.7% improvement is possible. If capacity is matched, more than 6% COP improvement is possible. COP improvement tends to increase towards lower ambient temperatures. One of the reasons for this is that the performance improvement due to vapor injection becomes more important than the re-expansion losses at the injection ports since the cooling effect of the injected vapor becomes more significant. The flash gas bypass only leads to a COP (3%) and capacity (7%) improvement for the H2 condition, but did not lead to any benefits for the other operating conditions.

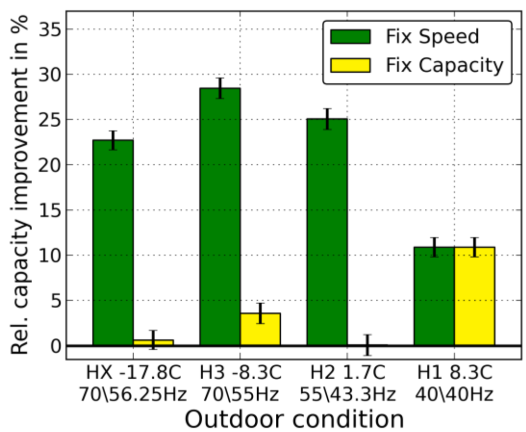


Figure 6: Relative capacity improvement, B1 vs. B0

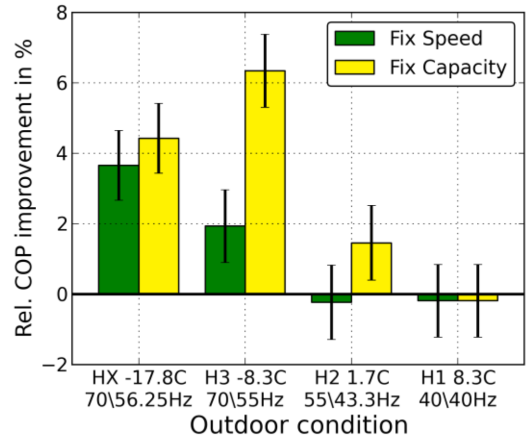


Figure 7: Relative COP improvement B1 vs. B0

One benefit of the vapor injection is that the capacity degrades less towards lower ambient temperature if the same compressor speed is used. Figure 8 shows that the capacity for the vapor injected system increases by nearly 7% as ambient temperature decreases from the H1 to the H3 test while compressor speed is increased from 40 to 70 Hz. For the same conditions, the B0 system capacity decreases by 2%. The differences in COP are less pronounced. The COP for all tested system configurations – even at the lowest ambient temperature – exceeds 2. COP decreases from the highest ambient temperature to the lowest ambient temperature. For the B0 system, the COP for HX conditions is 70% of the COP for H1 conditions. For the vapor injected system, a relative COP of 72% of H1 conditions is maintained under HX conditions. Application of the hybrid control lead to additional improvement of COP and capacity over the vapor injected system with standard distributor.

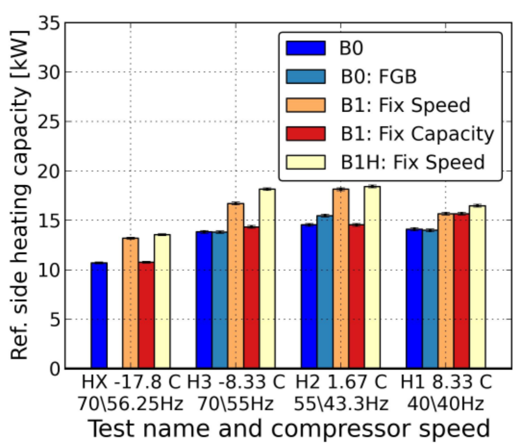


Figure 8: Refrigerant side heating capacity

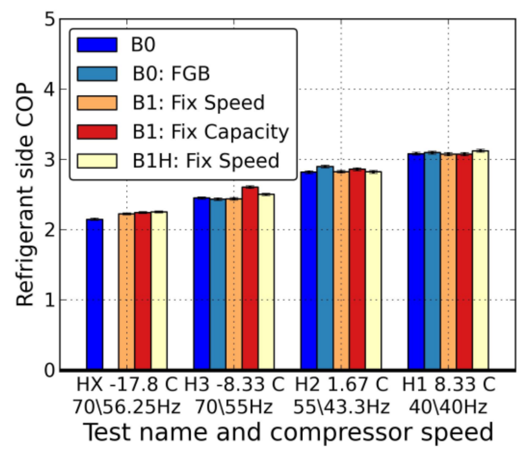


Figure 9: Refrigerant side COP

4.2 Issues Observed During Testing

An initially conducted test series used the original ANSI/AHRI 210/240 (AHRI, 2008) test conditions. This led to frost build up, which was – due to the higher capacity - more pronounced for the vapor injected system than for the B0 baseline system. The target outdoor room humidity was reduced to mitigate this issue. However, it was found that for the test with the larger humidity, the performance indices (COP, capacity) during the period of their maxima were very similar to the ones observed for the steady state tests with lower humidity.

While the prototype 2-step valves for the hybrid control worked well for H1 and H2 conditions, they found their limit at H3 conditions. Figure 10 shows that it was no longer possible to equalize exit superheats if severe coil blockage was applied. Figure 11 shows that this was caused by saturation of the valves for the circuit pairs 1&2, and 3&4 in fully open high flowrate position. Note that the valve for circuits 9&10 was saturated in the closed, low flowrate position – even for the case without coil blockage. This suggests the existence of refrigerant maldistribution at the distributor. For HX conditions, it was necessary to close the refrigerant liquid bypass (BP2) from the low pressure separator and use the primary expansion valve (PXV) after that separator to allow for superheat control.

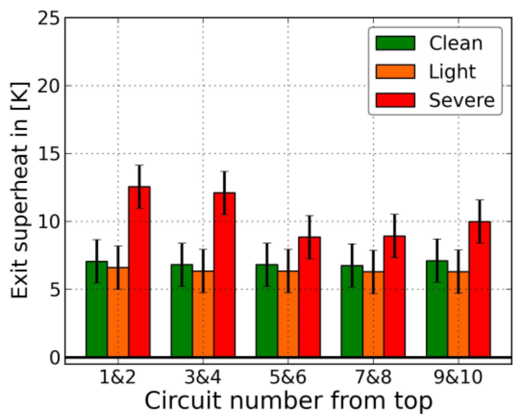


Figure 10: Individual circuit exit superheat (H3, B1H)

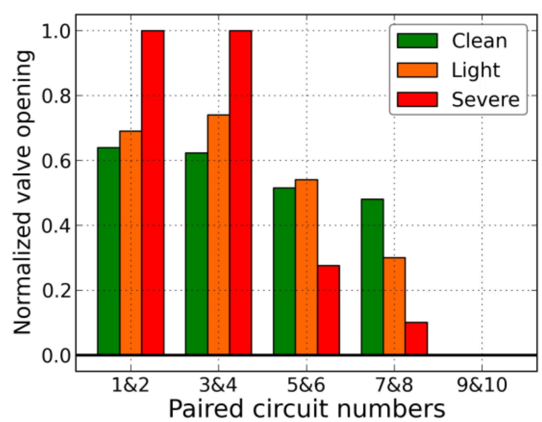


Figure 11: Balancing valve opening degree (H3, B1H)

4.4 Heating Seasonal Performance

Heating seasonal performance was calculated using a modified version of the ANSI/AHRI 210/240 HSPF (AHRI, 2008) calculation method. The results of this can be found in the companion paper, Bach *et al.* (2014b).

5. CONCLUSIONS

In this paper, an experimental test-set up of a cold climate heat pump that is able to operate in single stage, flash gas bypass, and two-stage vapor injection mode was introduced. Due to an interchangeable evaporator coil equipped with additional valves, the system can also run with the reduced hybrid evaporator flow control scheme.

- Compared to the baseline single stage configuration, the flash gas bypass mode increased COP and capacity by up to 3% and 7%, respectively.
- Running the vapor injected system at compressor speeds identical with the corresponding baseline tests resulted in 11% at 8.3°C (H1 test) to 28% at -8.3°C (H3 test) higher heating capacities.
- When the compressor speed of the vapor injected system was reduced to match the heating capacity provided in the baseline tests the COP improvement was about 6% at -8.3°C.
- The absolute value of the COP was higher than 2.0 for all system configurations and tested ambient air temperatures. On average, the COP decreases from approx. 3.1 at 8.3°C to approx. 2.2 at -17.8°C.
- Applying the reduced hybrid control scheme showed limitations of the employed 2-step balancing valves, For some operating conditions, saturation in fully open or fully closed position was reached.

Future work should include improving the vapor injected compressor: In the current configuration, the injection ports did not include check valves. This might be the cause for the smaller performance improvement than predicted by the simulations. On the hybrid control side, the opening area in the two positions of the two-step flow control valves should be modified to prevent saturation in open and closed position. On the evaporator side, a larger fin pitch and tube diameter should be used for cold climate heat pumps to reduce the effects of frost-build up and refrigerant side pressure drop.

NOMENCLATURE

Δ	Difference	(kW) or (-)
X	Value placeholder	(kW) or (-)

Subscript

baseline	baseline single stage configuration
i	system configuration index
j	climate zone index
norm	normalized

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