# Application of a hybrid control of expansion valves to a 5-ton domestic heat pump

Christian K. Bach*, Eckhard A. Groll, James E. Braun

Ray W. Herrick Laboratories, Purdue University, West Lafayette, IN, USA

*Corresponding Author: bachc@purdue.edu

## ABSTRACT

The proposed hybrid control employs a primary expansion valve that provides most of the pressure drop, while small secondary balancing valves in the distributor lines to the evaporator circuits adjust the refrigerant flow to provide equal circuit exit superheats. This paper presents experimental results for application of hybrid control of expansion valves to a 5-ton R410A domestic heat pump. Baseline performance data was taken with an electronic expansion valve (EXV) to determine the best possible performance without using individual circuit flow control. After that, secondary balancing valves were inserted into the distributor lines to complete the hybrid control scheme. The same tests as done with the EXV were repeated to determine the achievable performance improvement. Ice-up tests at high outdoor humidity were conducted to determine the influence of the control scheme on frost build up and system performance. For repeatable results, additional tests with a partially blocked evaporator coil were conducted.

It was found that during operation with the EXV, even with a clean coil, substantial maldistribution occurred leading to individual circuits with significantly different refrigerant exit conditions. With the EXV control scheme maintaining an overall exit superheat, some circuits had two-phase refrigerant at the exit while other circuits had a high degree of superheat. As a result, the evaporation temperature was lower than would occur for more even distribution leading to a performance penalty. In addition, the maldistribution resulted in uneven ice build-up during the ice-up test. The tests with the hybrid control scheme indicated that the maldistribution can be avoided and better overall performance achieved.

Similar tests as for the domestic heat pump have been conducted for a 3-ton large room cooling system; refer to companion paper Bach et al.(2012).

## 1. INTRODUCTION

If refrigerant and/or airside maldistribution occurs in evaporators, system performance degrades, as described by Kim et al. (2008). The authors used a simulation model to study the hybrid control applied to a 3-ton (10.55 kW) residential heat pump in AC mode and found that the capacity and COP degrade by approximately 4 and 6%, respectively, with applied airside maldistribution. They found that the application of upstream hybrid control recovers 99.9% of those losses. Kærn et al. (2011a) evaluated the influence of various sources of maldistribution on system COP and capacity using a 2-pass generic evaporator model, neglecting fan and control device power. The authors came to the conclusion that quality maldistribution in the distributor leads to a COP reduction of up to 13% and non-uniform airflow distribution to a COP reduction of up to 43.2%. Kærn et al. (2011b) found, using the same model as above, that most of the COP and capacity degradation, for a wide range of airside maldistributions, can be recovered if the exit superheat is controlled to the same value. However, equal superheat did not coincide with the maximum performance recovery, which was obtained at unequal exit superheats. They furthermore came to the conclusion that a 19% increase in evaporator size[1] leads to a similar COP improvement as the equal superheat control.

Li et al. (2005) came to the conclusion that, for a 4 branch distributor, depending on the type of distributor and considering a misalignment of 3.7° of the flow into the distributer, the flowrate in each branch can change by as much as +22/-35%. Additionally, it was found that gravity can influence the distribution if the distributor is not

[1] In their publication they use the nominal evaporator size, which was increased from 8.8 kW to 10.5 kW.

alligned perpendicular. Furthermore, an alternative design was suggested that is less vulnerable to misalignement of the inlet.

Mader et al. (2010) presented an expansion and distribution device that is able to control the individual circuit superheats with one common evaporator exit superheat measurement for a residential AC system. Experimental results of the possible performance improvement, however, could not be found in the open liturature, and only limited information is available from marketing documents, e.g. Danfoss A/S (2009) claims a COP improvement of up to 25% for HP mode or 1 SEER point for AC systems. Some indication that the previousely described performance recovery might be possible can be found in Payne et al. (2002). They conducted extensive experiments with finned tube evaporators. They found that the capacity losses due to uneven air flow distribution can be recovered to within 2% if the overall air flowrate is held constant and individual circuit superheat is controlled.

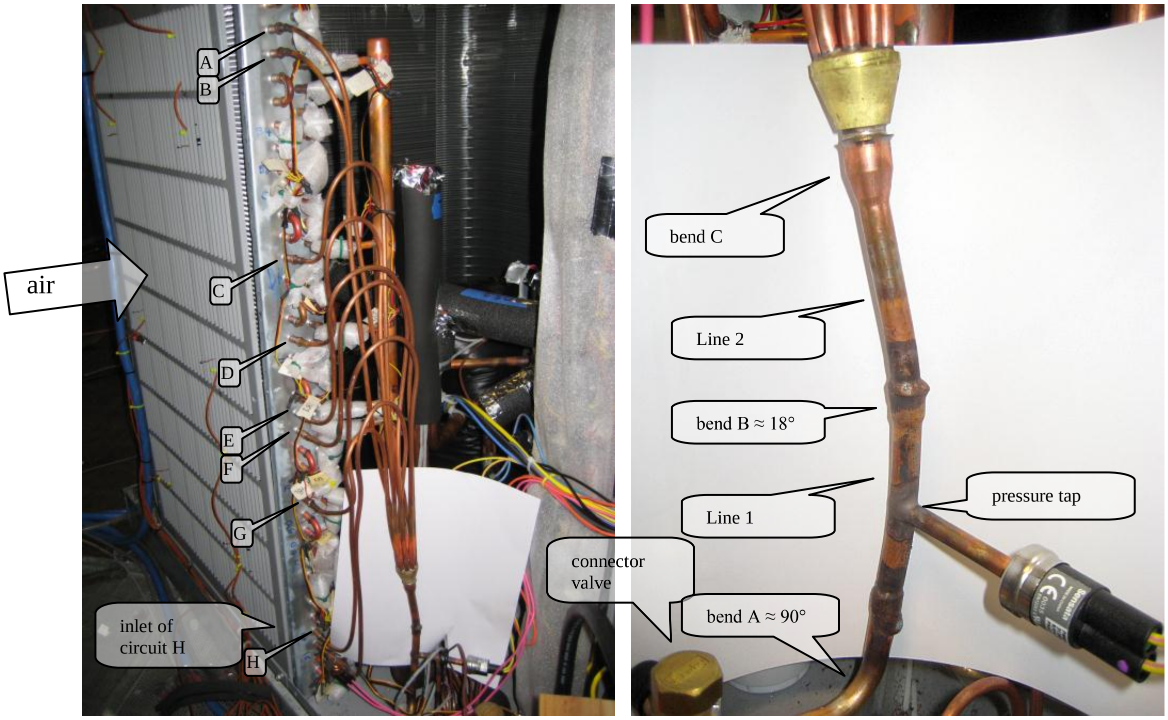
The purpose of this paper is to analyze the possible performance improvement if individual refrigerant flow control is applied to a 5-ton domestic heat pump. In addition to recovering most of the performance lost when air maldistribution was applied to the unit, the runtime under frosting conditions was increased by approximately 30% for the hybrid control as compared with an EXV control. In addition to the performance recovery that is possible by optimizing refrigerant flow distribution, the defrost control was found to have potential for improvement. In a conventional heat pump, the defrost cycle is initiated based on operating time and not based on actual frost built up. Furthermore, the defrost cycle is often ended based on a timer and not based on actual coil conditions. However, Meyer (2008) found that uneven ice built up for evaporators that are operated at sub freezing temperatures can be caused by incomplete defrosts which can be a result of broken defrost heaters or insufficient defrost end temperature and/or drip time. Since a timed defrost start and duration assumes a certain amount of ice build up, defrosts will either be too frequent or too unfrequent and a full defrost is not ensured under severe off design operating conditions. Bach (2009) found that intelligent defrost control with an adaptive defrost controller can lead to a 8-30% reduction in energy consumption for a -18°C freezing application. This reduction was achieved without a fully adapted controller.

## 2. EXPERIMENTAL RESULTS

### 2.1 Description of system and experiments

The 5-ton domestic heat pump that was used in this study has an outdoor heat exchanger (HX) with 9 circuits. The two top- and bottom-most circuits have 10 tubes while the 5 circuits in the middle have 5 tubes. The circuits are operated in cross-parallel flow if operated as an evaporator in heat pump mode as can be seen in Figure 1 a). Figure 1 b) shows a detailed picture of the distributor. A number of bends in the supply line to the distributer can be seen. Referring to Li et al. (2005) results, this is likely the main reason for the observed maldistribution in the EXV only tests.

The outdoor HX was initially equipped with a short tube orifice check valve. Since the drawbacks of fixed-opening expansion devices are well known, the system was not tested in that configuration. Baseline data was taken with an EXV, and after that balancing valves were added after the refrigerant distributor to complete the hybrid control scheme. The outdoor room temperature was varied between -20°C and 8.33°C while the indoor room temperature was held at 21.1°C. Tests were conducted at low outdoor humidity with and without coil blockage. Additional test were conducted with high outdoor humidity to determine the influence of frost build up. The evaporation temperature was controlled using a PI control applied to the primary expansion valve in both control schemes. The set point for the evaporation temperature was adjusted manually to achieve minimum overall superheat with no liquid refrigerant leaving the evaporator. This was done to simulate what an adaptive EXV controller would do after finishing its learning process.




a) Distributor and distribution lines b) Distributor, detail

**Figure 1:** Distributor and distribution lines for outdoor coil of heat pump

## 2.2 Performance indices

Performance indices for cooling capacity and efficiency are used as defined below.

Refrigerant side heating capacity was calculated as

$$\dot{q}_{Ref} = \dot{m}_{ref} \cdot (h_{cond,in} - h_{cond,out}), \tag{1.1}$$

where

$\dot{m}_{ref}$ = average refrigerant mass flowrate [kg/s]

$h_{cond,in}, h_{cond,out}$ = average enthalpy of refrigerant entering or leaving, respectively, the indoor unit (condenser) in heating mode[kJ/kg].

The air-side heating capacity, $\dot{q}_{air}$, is calculated as

$$\dot{q}_{air} = \dot{q}_{ref} + W_{i-fan}, \tag{1.2}$$

where

$W_{i-fan}$ = average indoor unit fan power consumption [kW].

The applicable standard, AHRI 210/240 specifies that all power consumption of the unit has to be included in the calculation of the coefficient of performance (COP). For the HP, it is therefore defined as:

$$COP = \frac{\dot{q}_{air}}{(W_{comp} + W_{controls} + W_{ccht}) + W_{o-fan} + W_{i-fan}}, \tag{1.3}$$

where

$W_{o-fan}$ = average outdoor fan power consumption [kW]

$(W_{comp} + W_{controls} + W_{ccht})$ = average compressor, controls and crankcase heater power consumption [kW].

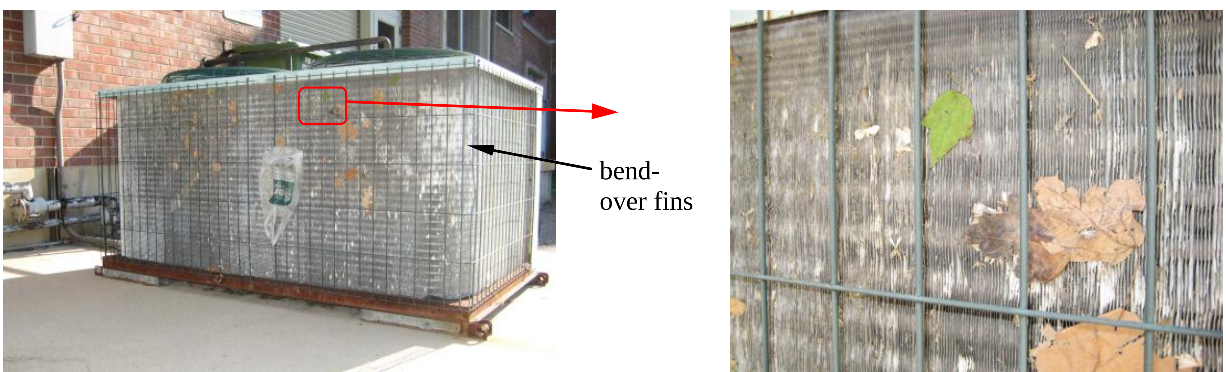
The crankcase heater consumption during the tests was 0 kW, since it was switched off as soon as the compressor was switched on. In part load operation, the crankcase heater will be on during compressor off-times, if the ambient temperature is lower than approximately 85 F.

The power consumptions of the primary EXV and balancing valves were not included in the power consumption of the system controls. This was done, since they will likely be negligible compared to the other power consumers of the control system once they are optimized.

For the transient ice up tests, instantaneous values were calculated for each measurement point according to the previously shown formulas.

## 2.3 Sources of maldistribution

The outdoor heat exchanger of heat pumps and AC systems that use air as a source/sink are subject to the ambient air. As compared to heat exchangers that are used to condition air for buildings, they do not have any type of filter in front of the heat exchanger. As a result, whatever particles can travel with air can end up on the outdoor coil. Figure 2 a) shows a university maintained AC unit to illustrate the resulting issues. The unit is subject to scheduled maintenance that includes coil cleaning. This leads to bent fins at the corners of the coil where it is difficult to keep the water jet used for cleaning aligned with the direction of the fins. Additionally, coil blockage is caused by waste and organic material. As Figure 2 b) shows, this includes insects, lawn clippings, and leaves with changing amounts depending on the time of the year. Figure 2 a) furthermore shows that this type of blockage preferably seems to accumulate at the top of the coil. Most likely, the reason for this is that the lower parts of the coil are less protected from rain. A simple way to reduce the amount of blockage would be to run the fans in reverse for a short period of time each day to blow the larger particles off the coil.




a) overview[2] b) detail

**Figure 2:** Coil blockage on a university maintained condensing unit, 08/29/2011 (Bach, 2011)

Additional sources of maldistribution are refrigerant side fouling, and the asymmetric and space-saving design of typical AC/HP condensing units which forces the air of the upper circuits to pass around the air shrouds of the fans. Furthermore, in case of heat pump operation, quality maldistribution in the refrigerant distributer affects the performance even without any fouling.

## 2.4 Clean coil test

For the clean coil tests, the outdoor humidity was kept as low as possible to keep ice build up at a minimum. Performance data was taken for both control schemes with ambient air temperatures of -20°C, -8.3°C and 8.3°C. To gain insight on how well the refrigerant is distributed, the temperature at the inlet to circuit A (top in Figure 1a) and the temperatures of each return bend from the $3^{rd}$ one to the exit were measured. It was found that the measured temperatures along each path do not always follow a clear trend, as shown in Figure 3. It is believed that this is caused mainly by cross fin conduction: the $4^{th}$ return bends of circuits B and C as well as G and H have neighboring tube pairs, leading to similar measured temperatures. The sensors used for the exit superheat measurements were placed as far as possible away from the fins and the suction header to reduce this type of effect. As a result of the

[2] The plastic bag was not placed on the coil for the purpose of this picture - its presence was by coincidence.

inaccurate temperature measurements, it is not possible to gain reliable information on the coil surface usage from Figure 3 alone.

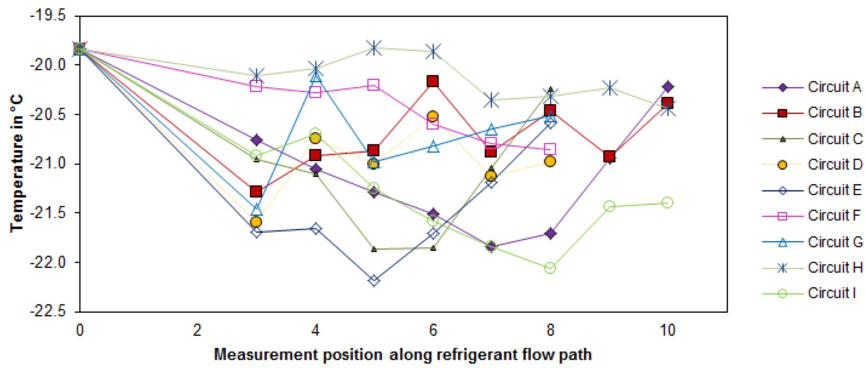
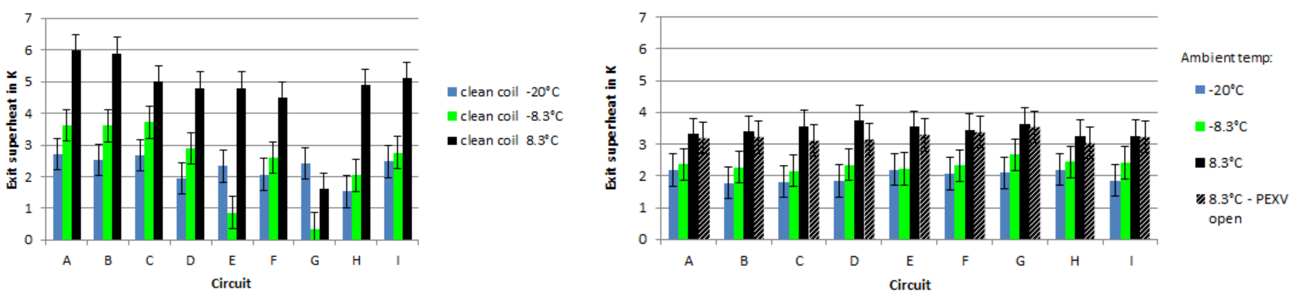



**Figure 3:** Inlet and return bend temperatures, -20°C ambient temperature, single EXV control scheme[3]

Figure 4 a) shows the individual circuit exit superheats for the different tests for the single EXV control scheme. The difference between the minimum and maximum superheat increases with ambient temperature from approximately 1 K at -20°C to more than 4 K at 8.3°C ambient temperature. This is a result of the increasing capacity of the system with ambient temperature that leads to a larger difference between air inlet temperature and evaporation temperature. This allows for larger superheats to occur if the coil surface usage of the individual circuits is uneven. For the test above -20°C some of the circuits with low exit superheat feed liquid through such that high superheat in the remaining circuits is necessary to maintain a sufficient overall superheat. Figure 4 b) shows the individual exit superheats for the experiments with the hybrid control scheme. The difference between minimum and maximum individual circuit exit superheat is less than 0.5 K for all tests.




a) single EXV control scheme

b) Hybrid control scheme[4]

**Figure 4:** Individual circuit exit superheat, clean coil tests[5]

Figure 5 shows the coil surface usage in terms of the end of the two-phase section. As noted for Figure 3, the average temperature value at the return bends is insufficient to gain reliable information on the coil surface usage. Therefore, the behavior of the return bend temperature over time was taken into account to determine the end of the two-phase section: if the end of the two-phase section occurs close to a return bend, the temperature changes significantly. This fact, in conjunction with the temperature measurements, was used to manually judge the end of the two-phase section. Figure 5 a) shows the result for the EXV control scheme. The end of the two-phase section ranges from before the 3rd return bend to full usage of the circuit with liquid fed over. For the test at -20°C the overall superheat was sufficiently low even with no circuit feeding liquid over. Figure 5 b) shows the individual circuit usage for the hybrid control. The majority of the circuits are very well used.

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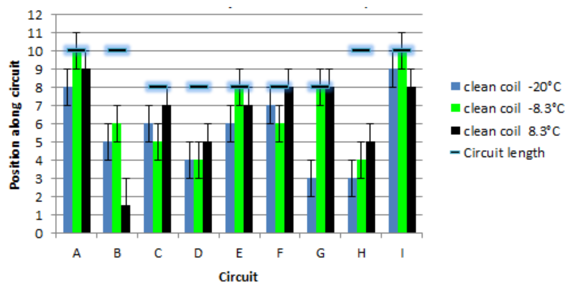
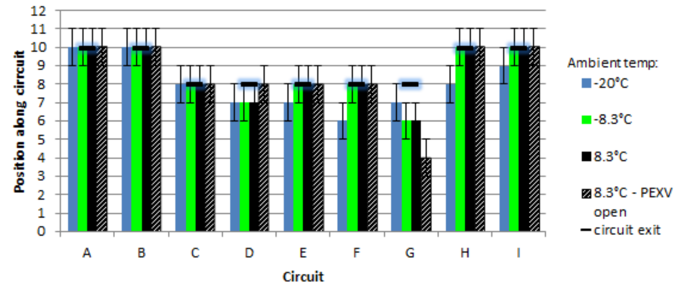
[3] Rated uncertainty of temperature sensors is 0.5 K.
[4] The "PEXV open" test was done with the primary expansion valve fully open to reduce inlet quality to the balancing valves, such that their opening degree could be used to determine the individual circuit mass flow rates.
[5] Error bars show rated uncertainty of temperature measurement only.




a) EXV control scheme




b) Hybrid control scheme

**Figure 5:** Circuit two phase section end, clean coil tests[6]

Figure 6 shows the improvement of COP and capacity that is achieved by employing the hybrid control scheme as a function of ambient temperature. For the tests at -8.3°C and +8.3°C, a COP improvement of approximately 2% and a capacity improvement of more than 1% were achieved. For the test at -20°C, a larger improvement was achieved, which is caused by frost build-up of the coil due to the limited dehumidification capacity of the psychrometric rooms at that temperature. Interestingly, in all tests, the COP improved more than the capacity improved. Figure 7 shows that the subcooling for the hybrid tests was approximately 1 K lower than that for the EXV controlled tests. The smaller subcooling led to a decrease in condensing temperature and by that to less required compression power. An additional test was conducted, where the primary expansion valve was kept fully open to reduce the inlet quality to the distributor in order to be able to use the balancing valve openings to determine the individual circuit flow rates. As a side effect, more charge was held in the connection lines between primary EXV, distributor and balancing valves, leading to a reduction in subcooling as shown in Figure 7. This led to an additional reduction in condensing pressure. Figure 6 shows that the COP improvement increased by approximately 0.5% compared to the hybrid control scheme while the capacity increase is within the measurement uncertainty. As a result, one can conclude that the effect of subcooling on system capacity is minor for the tested conditions while there is a noticeable effect on COP. Therefore, the shown capacity improvement can be assumed to be realistic while the COP improvement should be at least 0.5% lower if a comparative subcooling was chosen.

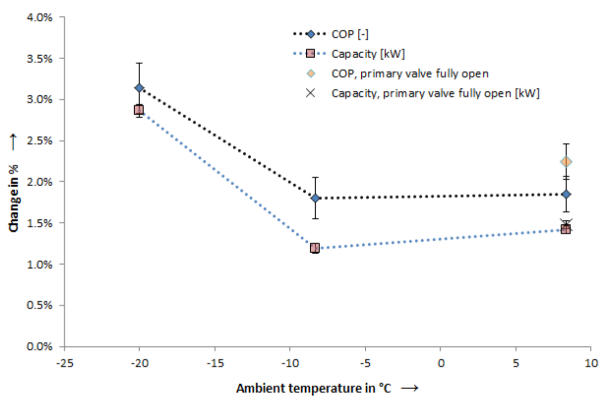



**Figure 6:** Hybrid COP and Capacity improvement, clean coil test

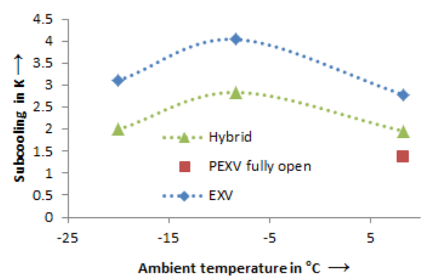



**Figure 7:** Subcooling, clean coil tests

## 2.5 Ice-up tests

For the ice up tests, the target room temperature and dewpoint were -8.3°C and -11°C respectively. Figure 8 shows the trend of the capacity for the two system configurations. The EXV control approach performed better at the start of the ice up tests than the hybrid control approach. The reason is that the superheat for the hybrid control scheme was not brought down aggressively enough. This in turn was partly caused by the changing quality maldistribution of the distributor, which depends strongly on the balancing valve positions. The hybrid and EXV control schemes performed similarly between 1 hour and 2 hours after the start of the experiments. At approximately 2 hours both system configurations had a

[6] Discrete measurements at in/outlet and return bends only, therefore limited resolution.

software crash, which led to missing data. Both systems were kept running during that time to avoid frost build-up reduction. After restart of the computer, the expansion valves had to be reset which resulted in a shut off time of approximately 5 minutes due to the start-up delay of the original heat pump controller. After the restart, the HP ramped up slowly to its initial performance. The delay in reaching the initial performance was caused by evaporating liquid refrigerant in the accumulator. The hybrid system started out better, and kept its capacity up to about 3 hours after the original start of the system, then followed the downward trend of the EXV. The trends for the COP are similar to the trends described for the capacity, as can be seen from Figure 9. For the hybrid control, the runtime increased by approximately 30%. It should be noted that the original defrost control was overridden to allow for the longer runtime. The original defrost control sums up the runtime where the coil temperature is below approximately 0°C, and then starts a defrost cycle after a pre-programmable time duration of 0.5 to 2 hours. The factory setting is 1.5 hours. The test duration was more than two times longer than that. In practice, however, incomplete defrost can occur, which leads to initial maldistribution when the system restarts after the defrost cycle. This was not the case for the test presented here, which was started with a clean coil.

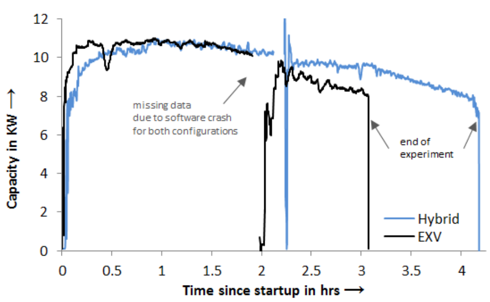



**Figure 8:** Capacity during ice-up test

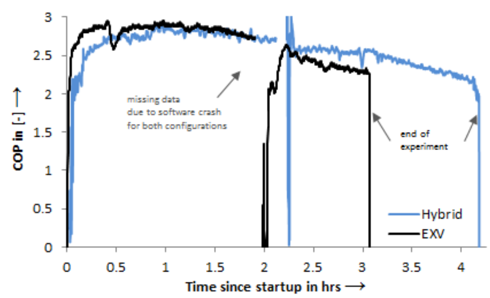



**Figure 9:** COP during ice-up test

Figure 10 shows the individual circuit exit superheats for the EXV control scheme based on the previously described surface measurements and the common evaporator exit saturation temperature. The surface measurements, which were necessary to avoid influencing the flow distribution, led to significant measurement uncertainties, such that the curve shape has to be taken into account to judge the outlet conditions. Circuit E fed liquid into the suction header for most of the test duration. The amount of liquid feeding over increased during the course of the experiment. Therefore, the evaporation temperature decreased, which led to increased superheat for the remaining circuits and reduced usage of their refrigerant side surface area for two-phase heat transfer. Additionally, the maldistribution on the air- and refrigerant side[7] changed over the course of experiment. This can be seen in the superheat signal of circuit G, which initially tended to feed liquid through but later on showed a large superheat. Figure 11 shows the exit superheats for the hybrid control scheme. There is no significant increase in the spread of the superheat signals towards the end of the experiment, since the system was based on individual circuit exit superheat control. Using the same approach as previously mentioned, the return bend temperatures were used to determine the end of the two-phase section. It was found that the coil surface usage at the end of the experiment was actually worse than for the EXV. To some extent, the better performance of the hybrid control scheme after 2 hours was caused by reduced liquid carryover, which does not require the evaporation temperature to be lowered unnecessarily. As a result, the evaporation temperature remained higher, leading to slower frost build-up. This furthermore means that if the hybrid control scheme had been run to achieve the superheat stability limit for individual circuits rather than to achieve the same superheat, further performance improvement would have been possible.

[7] It is not possible to distinguish between air- and refrigerant side maldistribution from the available data; the combined effect of both shows up in the resulting frost build up and exit superheats.

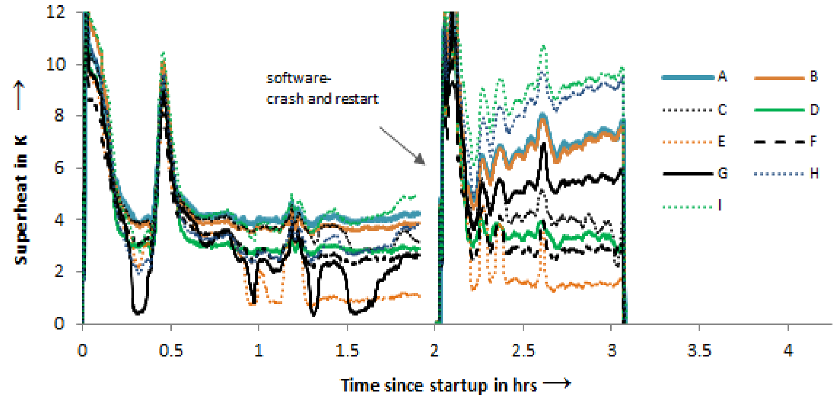



Figure 10: **Individual circuit exit superheats, EXV**

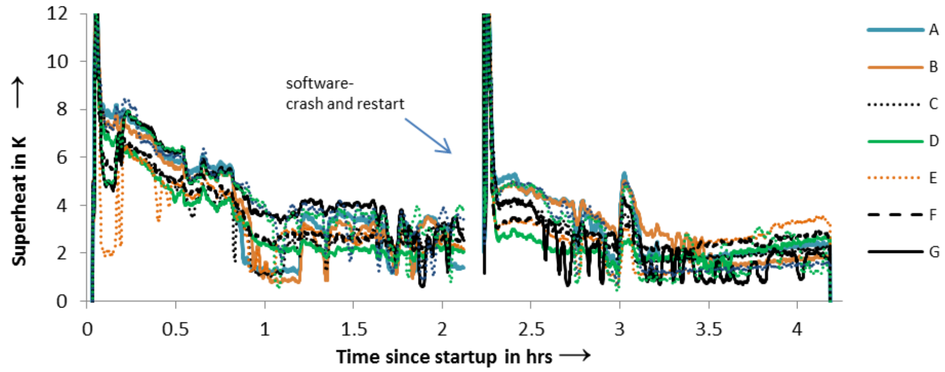



**Figure 11:** Individual circuit exit superheats, Hybrid

## 2.6 Blocked coil tests

For the blocked coil tests, the air inlet grille of the outdoor unit was blocked from the bottom up to approximately 50% of the coil. This did not result in a complete blockage but rather in a reduction of the air flowrate, since there is a gap of approximately 1.3 cm between air inlet grille and coil. This allowed for airflow to the lower circuits, where the air flowrate decreased from the middle to the bottom due to frictional losses in the gap. This is similar to what is expected if uneven defrost or severe buildup of leaves around the coil[8] occurs.

Figure 12 shows the refrigerant side coil surface area usage for the EXV control scheme. The uppermost 4 circuits had fully evaporated refrigerant before reaching the 3rd return bend due to the increased airflowrate of that part of the coil. The circuits of the lower part of the coil evaporated later with some of them feeding liquid over. The corresponding graph for the hybrid control scheme is not shown, since it is unspectacular: most of the circuits were used very well, with exception of the two bottommost circuits. These however, had a reduced capacity and therefore, did not contribute significantly to the capacity of the evaporator: only a small temperature difference between upper part of the coil and lower part of the coil existed and therefore, cross fin conduction was negligable (in contrast to the EXV case). Figure 13 shows the individual circuit exit and overall exit temperature measured at the outlet for the high ambient temperature EXV and hybrid tests. Despite having mostly much higher individual circuit exit superheats, the overall superheat for the EXV was slightly lower than for the hybrid control scheme. The reason for that is circuit G was feeding liquid through, which had to be evaporated by the excess superheat of the remaining circuits. For the hybrid control scheme, this problem did not exist, since the exit superheats were actively controlled to be within a span of less than 0.4 K.

[8] This is an issue of maintenance, which is highly variable. Residential units are typically not well maintained and outdoor heat exchangers are rarely cleaned.

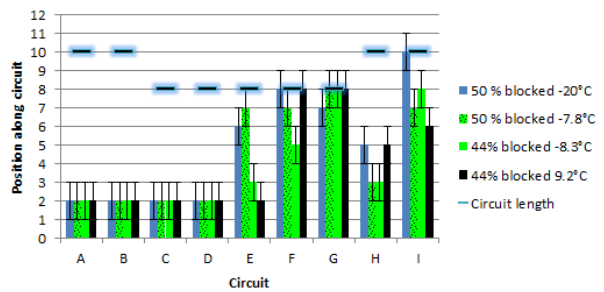



**Figure 12:** End of two phase section, blocked coil, EXV

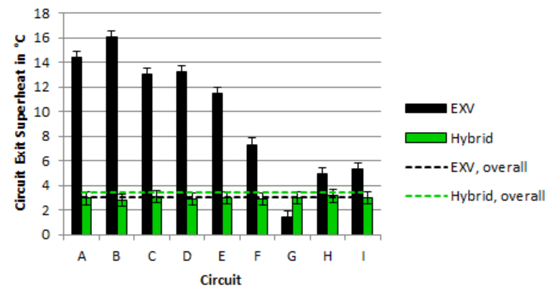



**Figure 13:** Circuit and overall exit superheat, Hybrid 8.4°C test and EXV 9.2°C test[9]

Figure 14 shows the capacity and COP improvement that was achieved after the EXV control scheme was replaced with the hybrid control scheme. The achievable performance improvement increases with ambient temperature due to the increasing capacity that leads to a larger difference between air inlet and evaporation temperature. The COP (capacity) improvement ranged from more than 4% (4%) at -20°C ambient temperature to nearly 20% (26%) at 8.3°C ambient temperature. It should be pointed out, that even though the capacity and COP improvements at lower ambient temperatures were smaller, it is only half the truth: the capacity improvement reduces the need for auxiliary heat, which is often electric, which then provides a greater improvement in COP.

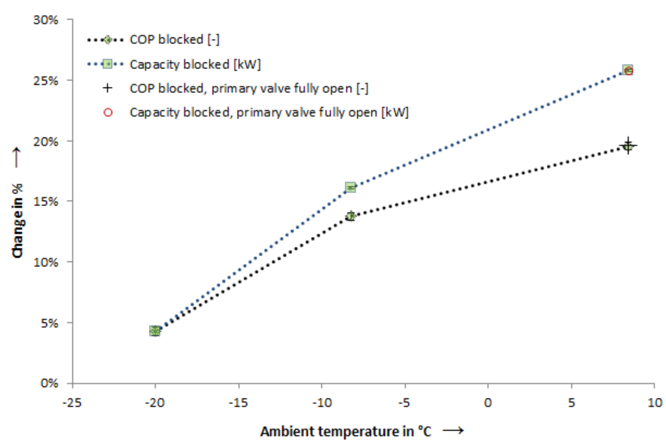



**Figure 14:** Capacity and COP improvement for blocked coil tests, Hybrid versus EXV[10,11]

## 3. CONCLUSIONS AND FUTURE WORK

The capacity and COP increase of a 5-ton domestic heat pump (DHP) with clean coil using the hybrid control scheme compared to the EXV control scheme were on the order of 1%. In practice, however, maldistribution due to refrigerant and airside fouling will occur over the lifetime of the system. A 4 to 26% improvement in capacity along with a 4 to 20% improvement of COP was observed with approximately half the air inlet grille blocked.

For the hybrid control scheme, the performance during the first part of a frost-up test was slightly less than for the EXV control scheme. The reason for that was changing quality maldistribution at the distributor during the process of reducing the exit superheat. Later on, the hybrid control system reached and exceeded the performance of the EXV control system. The reason in that case was that one of the circuits in the EXV test started feeding an increasing amount of liquid into the suction line, which required a lower evaporation temperature to gain sufficient overall superheat. In the hybrid control system, the flow rate of the circuits that started feeding liquid into the

[9] Uncertainty of temperature sensors shown. This uncertainty does not include effects of conduction along the tubes.
[10] Linear interpolation between adjacent measurements was used for the data of the EXV to account for different ambient temperatures.
[11] The point at -20°C used 50% air inlet grille blockage while the remaining points used 44% blockage.

suction line were reduced, and the evaporation temperature did not need to be lowered as much. This resulted in better COP and larger capacity. The hybrid control system extended the runtime of the frost-up test by approximately 30%. With better control of the balancing valves, performance and runtime extension would have been even greater.

To obtain better performance values for individual circuit flow control, future work should follow a different control approach. Achieving equal exit superheats does not provide the maximum performance due to measurement error and different behavior of the individual circuits due to maldistribution. The goal should therefore rather be to lower the superheat in each circuit individually until the minimum stable signal is found – similar to what is done for the overall superheat control for evaporators (e.g. Bachmann, 2008).

Additionally, improved defrost detection and control of heat pumps should be reconsidered in future research, since a more intelligent defrost detection and control will likely lead to an energy consumption reduction that is comparable to what can be reached using a hybrid refrigerant flow control scheme.

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