Application of two hybrid control methods of expansion valves and vapor injected compression to heat pumps

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ABSTRACT

This paper presents the benefits in capacity and performance of a hybrid control of expansion valves for two 5-ton (17.6 kW) domestic heat pumps (HPs) based on experimental results. The hybrid control, as proposed by Kim et al. (2008a), employs a primary expansion valve that provides most of the pressure drop, while small secondary/balancing valves in the distributor lines to the circuits of the evaporator control the refrigerant flow to provide equal circuit exit superheats. Initial baseline data for Heat Pump 1 was taken with an electronic expansion valve. After that, secondary/balancing valves were inserted into the distributor lines and the same tests were repeated. Frost build-up tests at high outdoor humidity were conducted with Heat Pump 1 to determine the influence of the control scheme on the frost uniformity and system performance. For repeatable results, additional tests with partially blocked evaporator coils were done for both heat pumps. Heat Pump 2 was equipped with a vapor injected compressor and had two interchangeable evaporator coils of the same type. One of the evaporator coils was equipped with a more cost effective reduced hybrid control scheme and employed prototype 2-step balancing valves.

Key Words: heat pumps, evaporator, flow distribution, hybrid control, vapor injection

1 INTRODUCTION

Refrigerant and/or air-side maldistribution in evaporators leads to system performance degradation, as described by Kim et al. (2008b). The authors used a simulation model to study hybrid expansion valve control applied to a 3-ton (10.55 kW) residential heat pump in cooling mode and found that capacity and COP degraded by approximately 4 and 6%, respectively, with applied air-side maldistribution. They furthermore found that the application of upstream hybrid control recovered 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. The authors came to the conclusion that quality maldistribution in the distributor led to a COP reduction of up to 13% and non-uniform air flow 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 air-side maldistributions, can be recovered if the exit superheat is controlled to the same value. However, the maximum performance recovery was obtained with unequal exit superheats. They furthermore found that a 19% increase in nominal evaporator size led to a similar COP improvement as the superheat control.

Li (2001) numerically investigated several different types of refrigerant distributors. His numerical results showed a high sensitivity of the distributor to imperfections. For a 4 branch distributor, depending on the type of distributor and misalignment of internal components, the flowrate in each branch changed by as much as +22/-35%. Additionally Li (2001) found that gravity can influence the distribution if the distributor is not aligned perpendicular. He suggested an alternative design, which is less vulnerable to misalignment of the inlet.

This paper investigated the effects of refrigerant and air-side maldistribution on the performance of two heat pumps. The first heat pump is equipped with the full hybrid control scheme, implemented by inserting individual electronic expansion valves into each of the passes of the evaporator. The second heat pump is equipped with a more cost effective reduced hybrid control scheme and a vapor injected compressor. In the reduced hybrid control scheme, two circuits are paired and their mass flowrate is controlled by a single expansion valve, which is a prototype 2-step solenoid valve. The solenoid valve can change between two degrees of opening.

2 EXPERIMENTAL RESULTS OF HEAT PUMP 1 WITH FULL HYBRID CONTROL SCHEME

Heat Pump 1 (HP1) consists of an outdoor unit that includes the compressor and a separate indoor unit, both are commercially available. The nominal cooling capacity of the unit is 17.6 kW. The rated seasonal energy efficiency ratio (SEER) of the unit is 15 Btu/Wh (4.4 W/W) and the rated heating seasonal performance (HSPF) is 8.5 Btu/Wh (2.5 W/W). The outdoor fan and compressor operate at fixed speeds. The electronically commutated (EC) motor driven indoor fan is set to provide an air flow rate of ≈1680 cfm (≈0.8 m^3), which is equivalent to ≈336 cfm per nominal ton (≈0.045 m^3/s per nominal kW). The original orifice expansion device of the heat pump was removed and replaced by an electronic expansion valve (EXV). This was done in order to obtain EXV baseline data. After the EXV baseline tests, EXVs were inserted in each of the 9 feeder tubes of the outdoor heat exchanger to complete the hybrid control scheme and the tests were repeated.

2.1 Frosting Tests

The purpose of the frost build-up tests was to investigate the change of the performance of the heat pump as the evaporator frosting increases. In the manufacturer supplied control scheme, a defrost cycle is initiated for sensed heat exchanger coil temperatures of less than 0°C after a selectable time of 30, 60, 90 (factory setting) or 120 minutes of compressor operation. The frosting tests were conducted at -8.33°C dry bulb temperature and -11°C dew point temperature measured at the air inlet to the outdoor unit, starting with clean coil conditions. The air inlet temperature of the indoor unit was kept at 21.1°C for all tests. Figure 1 shows that the frost build-up for the EXV is very uneven with circuits B and H not showing any frost build-up. Circuit B frosted up after more than 1 hour of operation and circuit H did not show any frost until the termination of the experiment 3.1 hours after the start of the compressor. Figure 2 shows that this was caused by the uneven distribution of refrigerant between the different circuits, estimated based on the measured return bend temperatures. For circuit H, only approximately 30% of the available refrigerant side surface area was used for evaporation while circuit E led to liquid feed-over during the later parts of the experiment. Circuits C-D where better utilized towards the end of the experiment while the utilization of circuits G-I decreased. The remaining circuits have the same final utilization as they had at the beginning of the experiment. While the hybrid control (no figure shown) did not lead to a better surface area usage during the beginning of the experiment, it led to a much better usage of the individual circuits towards the end of the experiment. This led to a

30% longer runtime at which time the same performance degradation as for the EXV controlled HP was reached.

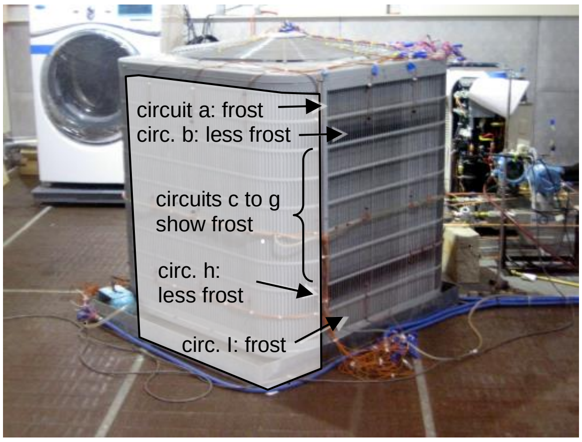


Figure 1: EXV Frosting Test, 0.47 hrs after Start

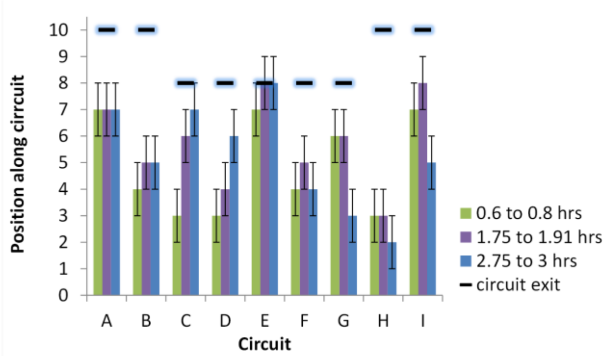


Figure 2: Circuit Two Phase Section End, for Different Time Periods, Frosting Test, EXV

2.2 Steady State Tests

Steady state tests with low outdoor room humidity were conducted to allow for comparison of the two different control schemes under steady state operating conditions. Tests were conducted at different ambient temperatures with and without uneven blockage of the air inlet grille of the outdoor unit. This uneven blockage was used to simulate uneven frost build-up occurring after several incomplete defrost processes. The performance degradation for the blocked coil tests was smaller than observed at the end of the frost build-up tests.

Figure 3 shows that COP improvement and capacity improvement for the hybrid control are on the order of 2% and 1.5%, respectively. The test at low ambient temperature shows a larger improvement, which is caused by slight frost build-up on the coil, leading to an advantage for the hybrid control system. The tests with primary expansion valve (PEXV) fully open were conducted to ensure equal inlet conditions to the balancing valves for virtual flowrate sensing purposes. These tests led to a reduction of subcooling since more charge was held between PEXV and balancing valves. This resulted in less subcooling, a lower condensing pressure and consequently in a slightly higher COP.

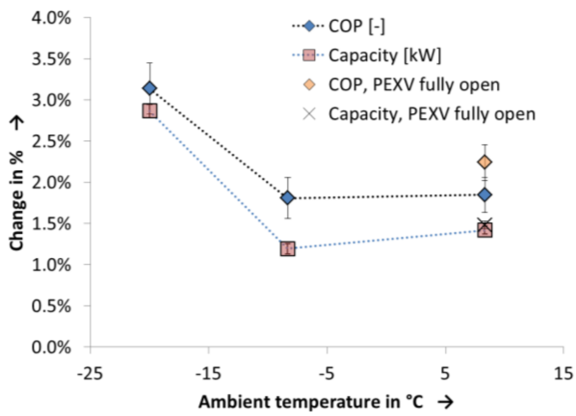


Figure 3: Performance Change after Implementing Hybrid Control, Clean Coil Cases

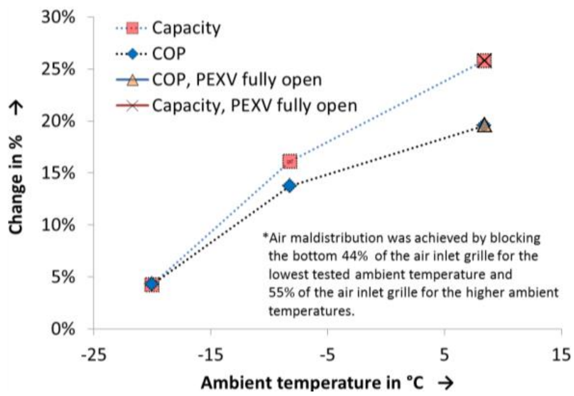


Figure 4: Performance Change after Implementing Hybrid Control, Blocked Coil Cases

Figure 4 shows that the performance benefit of the hybrid control increases with increasing ambient temperature if coil blockage is applied. COP and capacity improvement increase from approximately 4% at low ambient temperature to 20% (COP) and 26% (capacity) at high ambient temperature. Note that these numbers are a result of the recovery of lost performance: Application of severe blockage lead to a reduction of capacity (COP) of 23% (18%) for the high ambient temperature tests without hybrid control. For the same condition, the capacity (COP) with hybrid control is only 3% (2%) lower than the corresponding non-hybrid control clean coil case.

3 EXPERIMENTAL RESULTS OF HEAT PUMP 2 WITH REDUCED HYBRID CONTROL SCHEME

Heat Pump 2 (HP2) employs the same indoor unit as HP1 but with its outdoor heat exchanger divided into 10 circuits (rather than 9 circuits as used in HP1) leading to larger air and refrigerant surface areas since tube spacing is identical and circuit lengths are similar. As a result, the rated seasonal energy efficiency ratio (SEER) of the unit is 16 Btu/Wh (4.7 W/W) and the rated heating seasonal performance (HSPF) is 9.4 Btu/Wh (2.8 W/W). The original 2-stage scroll compressor of the unit was removed and replaced by a prototype dual-port vapor injected scroll compressor operated on a variable speed drive. The compressor speed was adjusted to obtain a better fit between building heating requirement and heating capacity of the heat pump. Figure 5 shows a simplified system schematic of the modified heat pump. The schematic shows the vapor injected HP operating mode, e.g. components that are not in operation are shown in gray. Starting from the evaporator, the refrigerant flows through the 4 way valve, through the suction side accumulator to the compressor main suction port. Flash gas generated by the expansion processes and separated using the high pressure (HP) and low pressure (LP) vapor separators is injected during the compression process. The discharge gas flows through muffler (M), 4-way valve, and vapor line to the indoor heat exchanger (condenser). From the indoor unit, the condensed and subcooled refrigerant passes through the check valve in the indoor TXV and returns through the liquid line to the outdoor unit where it enters the expansion and separation process. Note that, as a result of that process, the refrigerant flowrate at the evaporator is smaller than at the condenser. The subcooling of the refrigerant is controlled in the outdoor unit by the subcooling control valve (SXV), the level of the 2 separators is equalized by the drain valve (DXV), and the superheat of the evaporator is controlled by the primary expansion valve (PEXV) (standard evaporator coil, bypass BP2 closed) or the balancing valves (BP2 open) in case of the hybrid evaporator coil. Subcooling was controlled to 5 K at the outlet of the indoor unit. Superheat was controlled to 5 K at the evaporator exit manifold. In some cases, it was necessary to increase the target superheat to obtain stable operation. The filter-drier (F) removes any moisture and contaminants from the refrigerant flow. The EXV is used as sole expansion device for controlling the evaporator superheat to 5 K for single stage operating mode without vapor injection. BP1 is necessary for flash gas bypass operating mode, which is not part of this paper.

The hybrid evaporator coil of HP2 is equipped with the reduced hybrid control scheme. The secondary distributors for that coil are a round base design similar to the one which Li (2001) found to be less vulnerable to inlet flow misalignment. Figure 6 shows that the 10 circuits of the evaporator are paired up to 5 parallel circuits, each controlled by a single prototype 2-step expansion valve. Note that in Figure 5 only 3 balancing valves are shown to indicate that a hybrid control approach is used.

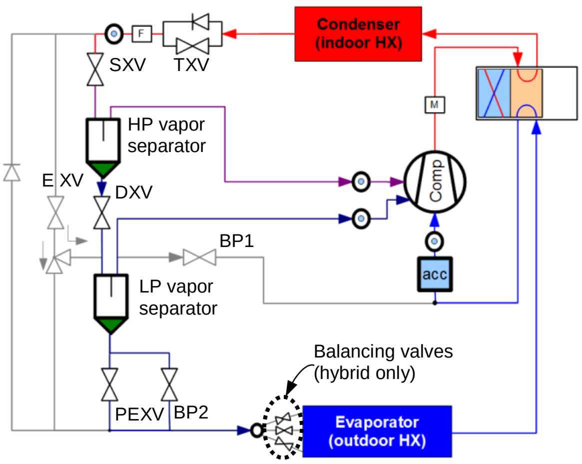


Figure 5: Simplified System Schematic of Vapor Injected HP

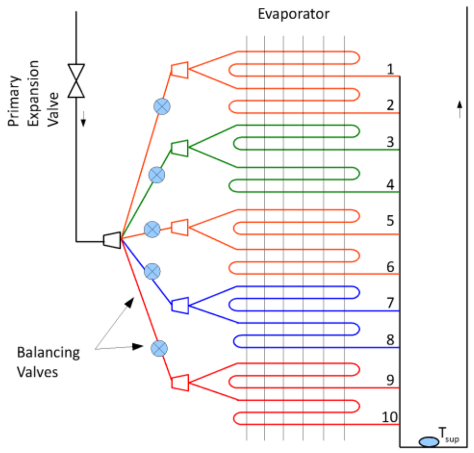
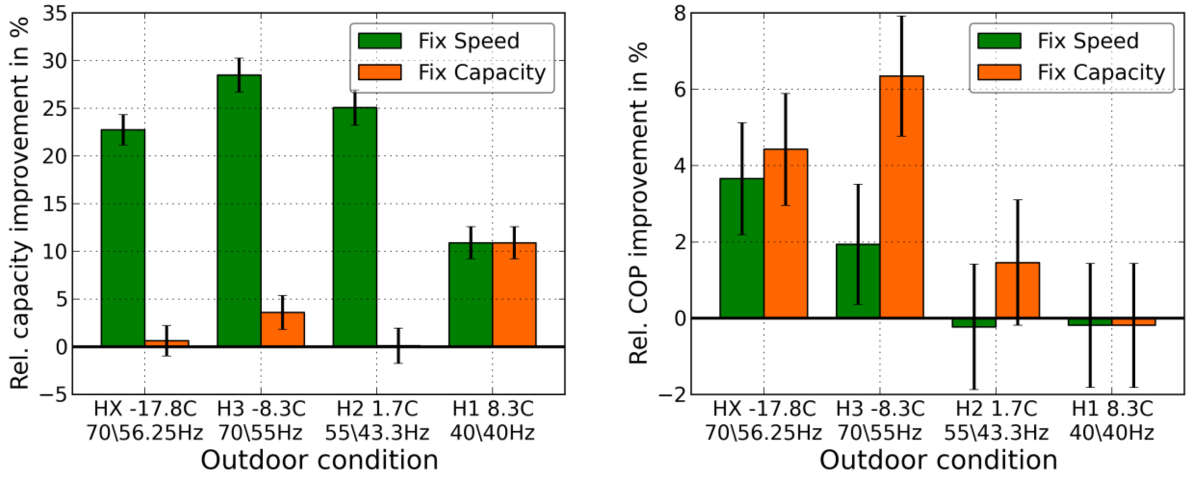


Figure 6: Reduced Hybrid Control with Simplified Evaporator Layout

3.1 Steady State Tests

Initial tests of the unit were conducted under standard AHRI 210/240 (AHRI, 2008) heating test conditions in single stage mode. However, this led to frost build-up on the evaporator. To obtain an estimate on the achievable performance improvement, the part of the transient data with the best performance was selected manually. However, the vapor injected configuration led to a longer time until steady state operation of the unit was achieved since the employed control algorithm was not optimized. Furthermore, vapor injection leads to a larger capacity, which additionally increases frost build-up. Therefore the evaluation of that data underestimates the benefits of the vapor injected system. Figure 7a shows that the vapor injected system has an approximately 25% higher capacity for the tests at and below an ambient temperature of 1.7°C. To match the capacity, the compressor speed[1] was reduced, e.g. from 70 Hz to 56.3 Hz for the additional low temperature HX test with -17.8°C ambient temperature which is not specified in AHRI 210/240. For the test at 8.33°C, a further reduction in compressor frequency was not possible. Figure 7b shows that, a COP improvement of up to 6% was achieved if capacities were matched. If frequency was matched, the achievable COP improvement was smaller – partly due to the larger differences between the air inlet and phase change temperatures in the condenser and evaporator.

[1] Compressor speed is used within this paper to abbreviate the frequency of the three-phase power supplied to the compressor motor. Since an asynchronous motor is used, the actual motor speed is slower; based on test-data with 60 Hz, the rotor speed is about 97.7% of the input power frequency.



a) Heating Capacity b) Coefficient of Performance (COP)

Notes:

> Low temperature HX test was added in addition to the standard H1, H2, and H3 tests.

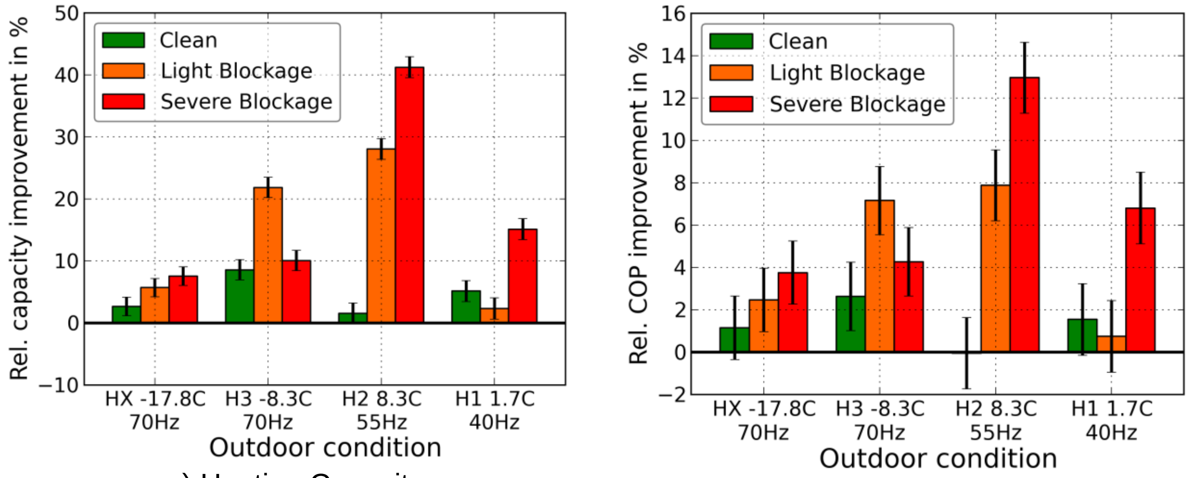
> The outdoor condition axis gives test name, outdoor temperature, and compressor frequency for baseline and vapor injected configuration with matched speed\vapor injected configuration with capacity match.

Figure 7: Relative Performance Improvement for Vapor Injected Configuration

Testing is currently in progress to evaluate the performance benefits of the vapor injection without frost build-up to allow for more reliable and fairer comparisons of results. Steady-state tests with reduced outdoor room humidity (LRH) for the H2 and H3 test conditions have already been conducted at this point for the vapor injected system with standard and hybrid evaporator coil.

Figure 8 shows the relative performance improvement achievable with the reduced hybrid control scheme. Capacity (COP) improvement for the clean coil case is the largest for the H3-LRH test conditions at 8.5% (2.6%). The outdoor unit air inlet grille was blocked by 24% (light) and 50% (severe), with the blockage starting at the bottom of the air inlet grille. In general, COP and capacity improvements increase with increasing blockage level, with the largest capacity improvement (41%) and COP improvement (13%) occurring at the H2-LRH test conditions. One obvious exception of this is the H3-LRH test, where the capacity improvement for the severe blocked test is smaller than for the light blocked test. This is caused by the limited actuating capability of the employed 2-step valves, resulting in saturation of the controls. In this specific case, the valves of the 2 bottom circuit pairs saturate in closed position while the valve of the top circuit pair saturates in open position.

Note that the larger performance improvement for the blocked cases shown in Figure 8 is a result of performance recovery. For the severe blocked coil cases this does not lead to an improvement over the clean coil control cases. To give an example: Application of severe blockage lead to a reduction of capacity (COP) of 40% (19%) for the H2-LRH tests without hybrid control. For the same condition, the capacity (COP) with hybrid control is only 1.5% (1%) lower than the corresponding non-hybrid control clean coil case.



a) Heating Capacity

b) Coefficient of Performance (COP)

Notes:

> Low temperature HX test was added in addition to the standard H1, H2, and H3 tests.
> H2-LRH and H3-LRH tests conducted at lower outdoor room humidity to minimize frost build-up.
> The outdoor condition axis gives test name, outdoor temperature, and compressor frequency for baseline and vapor injected configuration with matched speed\vapor injected configuration with capacity match.

Figure 8: Relative Performance Improvement for Hybrid Control, Vapor Injected Configuration.

4 CONCLUSIONS AND FUTURE WORK

Hybrid and reduced hybrid control can lead to a significant improvement of COP and capacity for heat pumps if air-side or refrigerant-side flow maldistribution exists. Without additional air-side maldistribution, both control schemes lead to relatively small performance improvements. However, if additional air-side maldistribution is applied to the system, then the achievable COP and capacity improvement increases significantly. In the most extreme case for HP1 a capacity (COP) improvement of 26% (20%) was observed. The most extreme case for HP2 led to an observed capacity and COP improvement of 41% and 13%, respectively. The level of blockage of HP1 led to a performance degradation that was smaller than the one observed at the end of the frosting test.

NOTE: Test results for both HP1 and HP2 with blocked coils indicated significantly degraded performance (capacity and COP) compared to the corresponding clean coil tests. However, application of hybrid expansion valve control lead to recovery of most of the resulting performance losses. The COP improvement values stated above therefore are not be mistaken as an improvement over clean coil performance!

Initial results show an increase of COP and capacity for the vapor injected (vi) compression. When the compressor speed was held constant, an up to 25% increase in heating capacity was observed. When the compressor speed was varied to match the non-vi baseline's capacity, an up to 6% improvement in COP was observed. These results are interim and most likely underestimate the benefits of the vapor injected compression.

Future work should include projects to evaluate actual air flow maldistribution (e.g. caused by organic material and frost build-up) in the field. This step is necessary for judging the correctness of the chosen blockage levels. For HP2, a repetition of the baseline tests without vapor injection under low outdoor humidity conditions is necessary to accurately evaluate the benefits of the vapor injected compression.

5 REFERENCES

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