Application of a hybrid control of expansion valves to a 3-ton large room cooling system

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ABSTRACT

The hybrid control, as proposed by Kim et al. (2008), 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 adjust the refrigerant flow to provide equal circuit exit superheats. This paper shows the experimental results for the application of the hybrid control of expansion valves for a 3-ton R404A large room cooling system. Data with the inbuilt expansion device, a pressure compensated TXV, was taken for a limited number of tests. Baseline data with an electronic expansion valve was taken 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 indoor room humidity were conducted with all control schemes to determine the influence of the control scheme on frost build up and system performance. For repeatable results, additional tests with partially blocked evaporator coil were conducted with the hybrid and EXV control scheme.

It was found that for TXV and EXV, even with a clean coil, substantial maldistribution occurs. This maldistribution lead to different usage of the individual circuits in terms of area fraction used for evaporation of, and area fraction used for superheating of refrigerant. With the EXV control scheme, the evaporation temperature had to be decreased to obtain sufficient superheat on the circuits that did not feed liquid into the suction header in order to evaporate the liquid of the circuits that fed liquid into the suction header. In addition, this resulted in uneven ice build-up and very poor controllability, which was especially noticeable as severe hunting when using the TXV in the ice-up test. With the hybrid control scheme, the surface usage for evaporation on all circuits was larger than 75% for most of the time, which resulted in a higher evaporation temperature and by that in a greater COP and larger capacity.

1. INTRODUCTION

If refrigerant and/or airside maldistribution occurs, the system performance degrades, as described by Kim et al. (2008). He studied the hybrid control applied to a simulation model for a 3 ton (10.55 kW) residential heat pump in AC mode and found that capacity and COP degrade by about 4 and 6% respectively with applied airside maldistribution. He 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. He 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 previously, 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. 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 leads to a similar COP improvement than the superheat control.

Mader et al. (2010) showed a expansion and distribution device that is able to controll the individual circuit superheats with one common evaporator exit superheat measurement for a residential AC system. Experimental

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1 In their publication they use the nominal evaporator size, which was increased from 8.8 kW to 10.5 kW.
results on the possible performance improvement, however, could not be found in open literature, limited information is available from marketing documents, e.g. Danfoss A/S (2009) which claims a COP improvement of up to 25% for HP mode or 1 SEER point for AC systems. Some indication that promises that the previously 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 overall air flowrate is held constant and individual circuit superheat is controlled. The purpose of this paper is to show the performance improvement that is possible if individual refrigerant flow control is applied to a 3-ton large room cooling system.

2. Experimental results

2.1 Description of system and experiments
The 3-ton large room cooling system uses an evaporator with 8 circuits, which have the same number of return bends. The coil was initially equipped with a TXV. This was subsequently replaced by an EXV and then as a third configuration, balancing valves were added after the refrigerant distributor to complete the hybrid control scheme. Tests were conducted with all three configurations. For the EXV and Hybrid control scheme, the outdoor room temperature was varied between 12°C and 46°C while the indoor room temperature was held at 2°C at low humidity. Additionally tests were conducted with high indoor humidity and 35°C outdoor room temperature to determine the influence of frost build up. For the EXV and hybrid control scheme, additional tests with blocked evaporator coil were conducted. The evaporation temperature was controlled using a PI control in the EXV and Hybrid control scheme. The set point for the evaporation temperature was adjusted manually to ensure that no major amounts of liquid left the evaporator while getting maximum performance by always having the smallest possible superheat. This was done to simulate what an adaptive EXV controller would do after finishing its learning process.

2.2 Performance indices
Performance indices represent cooling capacity and efficiency. The following performance indices are used:
Refrigerant side cooling capacity, which is calculated as

\[
q_{ref} = \dot{m}_{ref} \cdot (h_{\text{evap,out}} - h_{\text{evap,in}}) \quad (1.1)
\]

where

\[
\dot{m}_{ref} = \text{average refrigerant mass flowrate [kg/s]}
\]
\[
h_{\text{evap,out}}, h_{\text{evap,in}} = \text{average enthalpy of refrigerant entering respectively leaving the evaporator unit [kJ/kg]}
\]

The cooling capacity \( q_{ss} \), which is used for the graphs and results and is calculated as

\[
q_{ss} = q_{ref} - W_{e-fans} - W_{\text{cond}}. \quad (1.2)
\]

where

\[
W_{e-fans} = \text{average evaporator fan power consumption [kW]}
\]
\[
W_{\text{cond}} = \text{liquid line solenoid power consumption [kW]}
\]

The applicable standard, AHRI 1251 specifies that all power consumption of the unit has to be included into the calculation of the coefficient of performance (COP). For the LRCS, it is therefore defined as:

\[
COP = \frac{q_{ss}}{W_{\text{comp}} + W_{e-fans} + W_{e-fans} + W_{\text{cond}} + W_{\text{chf}}}. \quad (1.3)
\]

where

\[
W_{e-fans} = \text{condenser fan power consumption [kW]}
\]
\[ W_{\text{ccht}} = \text{crankcase heater power consumption [kW].} \]

The power consumption of the EXV respectively balancing valves was not included. This was done, since these, once optimized for low power consumption, will likely be negligible compared to the other power consumers in the system. For the ice up tests, it is not appropriate to use average values for cooling capacity, compressor and fan power consumption due to largely changing system efficiency. In this case cooling capacity and COP are calculated for each time step. For all tests, a constant value was used for the condenser fan, the crankcase heater and the solenoid. These values were measured without the entire system running.

### 2.2 Clean coil test

Since it is believed that EXV’s will be the standard for cooling systems of the given size within a few years, all results were compared to the corresponding measured EXV base case.

Under clean coil conditions, as shown in Figure 1, COP and Capacity improve by about 4 and 6% respectively if the Hybrid control scheme is employed. There is a slight trend for the COP which might be caused by experimental accuracy. For the TXV only a slight degradation in COP is measurable; the degradation in capacity and COP increases with decreasing ambient temperature.

![Figure 1: Change of COP and Capacity for clean coil conditions compared to EXV base case](image)

The differences in performance between the different control approaches stem from different refrigerant distribution among the circuits. Since the coil has a nonmatching number of holes and tubes, the effective area for some of the circuits is larger than for others, Figure 2. This can also be seen in the return bend temperature of the coil for TXV and EXV control scheme, Figure 3 and Figure 4. In both cases, circuits 6, 7, 8 are the ones that start superheating first. The superheating start of the others does not fit to this, which indicates air- and/or refrigerant side maldistribution. From Figure 3 and Figure 4 it can be concluded, that the individual circuit exit superheat might not be a good indication of individual circuit refrigerant side area usage. In Figure 4, for example, the exit superheat of circuit 4 and circuit 6 are only 0.5°C different while only about 2/3s of circuit 6 and more then 3/4s of circuit 4 are used for evaporation.

For the hybrid control scheme, Figure 5, the temperatures of the different return bends are nearly identical. The difference between the different control schemes can also be seen in the evaporation temperature. For the TXV it is about -9°C. In the EXV case, a larger portion of the coil is used for evaporation, resulting in an increase of evaporation temperature to about -7.5°C. For the Hybrid control scheme, the used surface area is further increased resulting in an evaporation temperature of about -6°C.

One might be tempted to think that it would be possible to cheaply adjust the flowrates in the individual circuits in a test run by just pinching the distributor lines. Different ambient temperatures, however, lead to different profiles of the return bend temperatures in both, the EXV and TXV control scheme. This resulted in ice built up for the TXV starting at the bottom of the evaporator, Figure 6 a) rather than at the middle of the evaporator as one would expect if only given the return bend temperatures for 22°C outdoor room temperature of Figure 3. This suggests that just pinching the distributor lines does not work as an approach to fix maldistribution. Additionally we found that the opening positions of the individual valves in the hybrid approach were extremely different (<5% to 100% opening),
which suggests that adjusting mass flow rates at the distribution lines additionally increases quality maldistribution at the distributor.

Figure 2: Evaporator coil

Figure 3: Inlet and return bend temperatures, TXV, 22°C room temp.

Figure 4: Inlet and return bend temperatures, EXV, 22°C room Temp

Figure 5: Inlet and return bend temperatures, Hybrid, 22°C room temperature

2.2 Ice-up tests
To determine the influence of ice build-up, ice-up tests at 35°C outdoor room temperature were conducted. Different ambient conditions lead to different outdoor room conditions due to infiltration/exfiltration of humidity into the outdoor room. Therefore the average relative humidity in the rooms was slightly different in the rooms for the different tests, between 75 and 79%. Additionally, defrosts of the coils of the psychrometric rooms had to be conducted during the experiment. Therefore the absolute results of those tests cannot be quantitatively compared to each other. Furthermore, no repeated frost build up/defrost cycles were conducted as they would occur in practice, which could potentially lead to an amplification of maldistribution due to incomplete defrosting of the coil. Figure 6
shows the coil frosting for the different control schemes. For the TXV scheme, ice built up occurred faster than for the other control schemes and preferably at the very bottom of the coil. For the EXV control scheme, ice built up occurred preferably at the top of the coil. For the Hybrid control, ice built up was first noticeable in the lower 2/3rds of the coil and was less concentrated. Note that Figure 6 a) was taken much earlier than Figure 6 b) and c). Figure 7 shows the relative COP of the different systems as a function of time with 100% base for each. For the TXV, degradation of COP can be noted at about 3 hours after the start of the experiment. This degradation comes along with severe hunting of the TXV, which eventually leads to (and is caused by) periodical occurrence of liquid at some of the evaporator circuits exits. For the EXV and Hybrid control scheme, the drop in COP starts about 6 hours after the start of the experiment. It is believed that the different shape of the COP curves is a result of different type of frost and different distribution of frost. While the frost was relatively evenly distributed in the Hybrid case, in the TXV and EXV case, uneven frost build up occurred. It is believed that the more linear trend of the EXV and TXV degradation is a result of the increasing influence of cross fin conduction between circuits with higher respectively lower air mass flowrates over individual circuits in conjunction with this frost build up. In the Hybrid case, there is no or little cross fin conduction, therefore the efficiency drops steeper once the coil frosts up severely.

Figure 6: Coil frosting for different control schemes

Figure 7: Change of capacity at 35°C ambient temperature and high indoor humidity

Figure 8 shows the start of the superheated section and average/overall evaporator exit superheat for the TXV. The start of the superheated section was based on a 1.5 K threshold: If the temperature difference to the preceding measurement along a circuit was larger than 1.5 K, then the refrigerant was considered to be superheated. Temperature measurements were placed on the return bends, therefore the results are discretized. Until about 4 hrs after the experiment, only small changes in the first superheated position can be noted. After 4 hrs, the first superheated position starts to switch for an increased number of circuits and within a greater range. At the same time, the difference between average circuit exit superheat and measured overall exit superheat reaches its maximum. This is due to more frequent liquid carryover from some circuits into the suction header. As a side effect, this liquid carryover causes severe hunting of the TXV – noticeable not only in the difference and fluctuation.
of the overall/average exit superheat, but also in a larger fluctuation of the average superheat position start (black). Note that Figure 8 has a shorter x-axis than Figure 9 to Figure 12. For the EXV, Figure 9, liquid carryover occurs from about 0.8 hrs after startup, as it can be seen from the difference between average and overall superheat and also from the individual circuit superheats, Figure 10. The reason that this happens earlier is, that the superheat was controlled to a lower overall value. Towards the end of the experiment, the difference between overall and average superheat increases due to increased liquid carryover. One noticeable difference between TXV and EXV is that the average start of the superheat does not show increased fluctuations towards the end of the experiment – it rather shifts closer to the beginning to the coil. This is caused by the manual control of the evaporation temperature, which prevents unstable behavior in case of liquid carryover. Figure 11 shows the start of the superheated section for the hybrid control. The position of the individual circuits is either at the exit of the coil or does not seem to exist. The reason for that is that a very small superheat was used, such that all circuits were operated at their maximum – which leads to periodic liquid carryover and/or an exit superheat of less than 1.5 K – which counts as not superheated for the algorithm. The average and overall superheat for the hybrid control case are close to each other, which is also the case for the individual exit superheats, Figure 12. This shows that liquid carryover is much smaller than in the EXV- or TXV case. The hybrid control therefore reduces liquid carryover significantly while maximizing the refrigerant side area usage of the coil.

\[\text{Figure 8: Start of superheated section and superheat, ice-up test, TXV}^2\]

\(^2\) A defrost of the psychrometric rooms outdoor room cooling coil had to be conducted 3 hours after startup which lead to fluctuations of the room temperature of less than 1.7 K.
A defrost of the cooling coils of the psychrometric rooms cooling coil had to be conducted 5 hours after startup which lead to fluctuations of the room temperature of less than 1.6 K.
2.3 Blocked coil tests
To overcome the disadvantages of the ice-up tests, tests with partial blockage of the evaporator coil were conducted. The coil was blocked from the top to reduce the air flowrate to specific circuits, Figure 13 a). To achieve the blockage, plastic foil and porous material as shown in Figure 13 b) was used. To determine an appropriate level of

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4 Defrost cycles of the outdoor room coils of the psychrometric room had to be conducted at 4.5 hrs and 9 hrs, which lead to a temperature increase of less than 2.5 K and a short spike when the fan was switched back on of less than 4.2 K.
blockage, the performance of the EXV-controlled system with different types of blockage was evaluated, Figure 14 and Figure 15. The material that had less porosity was always at the top of the coil. All blockages lead to less performance reduction than the ice up test after 8 hrs. Two types of blockage were selected as test cases. For case A, the top 1/3\textsuperscript{rd} of the coil face surface was blocked entirely, while the middle 1/3\textsuperscript{rd} of the coil face surface was partially blocked with porous material. For case B, 2/3\textsuperscript{rd}s of the coil face were blocked entirely. This blockage seems severe – since the coil is 5 ¼\" deep, this does not actually result in a complete blockage of the airflow to the upper parts of the coil but in a reduced velocity – and by that to the performance reduction shown in Figure 14. In contrast to ice build up, the blockage and performance degradation is more repeatable.

![a) blockage of coil, side view](image1.png) ![b) porous material (backlit)](image2.png)

**Figure 13:** Coil blockage and porous material

![Figure 14: Influence of coil blockage on COP, 35°C ambient temperature](chart1.png)

![Figure 15: Influence of coil blockage on capacity, 35°C ambient temperature](chart2.png)

Figure 16 shows the influence of coil blockage and ambient temperature compared to the unblocked EXV base case. With the EXV control scheme, the performance dropped with decreasing ambient temperature and increasing blockage. The degradation in COP was close to 15% with the light blockage and close to 29% with the case B blockage for low ambient temperature. The EXV B* test is a repetition of the EXV B 35°C ambient test, since minor frost build up occurred during the latter one. This lead to an increase in surface area which, in turn, lead to reduced performance degradation.

With the hybrid control scheme, the performance dropped by a maximum of less than 8% for low ambient temperature at case B blockage. For the lighter blockage, case A, the COP even exceeded the EXV base case COP at very high ambient temperature.

The results for the cooling capacity, Figure 17, look similar in trend but with a larger magnitude: For the EXV control scheme it is questionable whether the system would have enough capacity to maintain the room temperature under peak load with the given maldistribution. For the highest ambient temperature, the occurring capacity degradation was 16% and 23% for case A and B, respectively. For the same ambient temperatures, the hybrid control scheme had a capacity reduction of less than 1% and 4% for cases A and B respectively.
3. Conclusions

Capacity and COP of a 3-ton large room cooling system with different types of evaporator flow control were compared to the EXV baseline for the system. It was found that, for clean coil conditions, the Hybrid control scheme increases COP by about 4% and capacity by about 6% with a slight trend dependent on the ambient temperature. Capacity and COP were reduced for the TXV control scheme; this reduction was more than 1% in COP and more than 4% in capacity for 22°C ambient temperature. In the ice-up tests, it was found that the performance of the TXV system decreases quicker than for EXV and Hybrid. While for EXV and TXV, the frost build up starts more concentrated in about 20 to 30% of the inlet face area, it is more evenly distributed for the hybrid control scheme. Due to slightly different operating conditions, different frost built up may have occurred. Therefore additional tests with coil blockage were conducted to get more repeatable results for EXV and Hybrid control case. It was found that for low ambient temperature, COP decreases by 15 to 29% for the EXV, while for the hybrid control scheme, the COP reduction is less than 8%. For the cooling capacity, the results were similar but had a larger magnitude. For the same temperature, the capacity decreased by 19 to more than 30% for the EXV, while in case of the hybrid control scheme the reduction was less than 8%.

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