Designing and Optimizing Systems for Compressor Rapid Cycling

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Abstract

The purpose of this project is to explore the system design implications of compressor rapid cycling. An earlier project yielded promising results for a typical 2-ton residential split system. It showed that cycle periods as long as 10-25 seconds should be able to produce COP’s comparable to those achievable with variable speed compressors. This project builds on those results and aims to develop ways of designing heat exchangers and other components to fully capitalize on this technology.

Even though the experimental program only includes residential systems, most findings all are also relevant for automotive applications where clutch-cycled open compressors are used. This project assumes that rapid cycling is possible by many means, some of them already in the market, and some still to be developed.

Earlier work identified the four most important factors that account for the difference between cycle efficiencies of systems with continuous (variable speed) and rapid-cycling compressors: 1) refrigerant-side temperature difference (heat transfer resistance); 2) refrigerant-side pressure drop; 3) heat exchangers thermal capacitance; and 4) refrigerant backflow through the compressor during off-cycle. Results suggested that microchannel heat exchangers may offer significant advantages for rapid cycling systems. This project was designed to test that hypothesis.

Several system configurations were conceived and tested, namely: 1) direct expansion (DX); 2) flash gas bypass (FGB) 3) high side receiver and 4) high side receiver with suction line heat exchanger (SLHX) all with a microchannel evaporator. Also different alternatives for minimizing each of the loss terms have been analyzed, analytically and experimentally. The results of those experiments and analyses are the subject of this report.
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## Nomenclature

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<tr>
<td>A</td>
<td>area, m&lt;sup&gt;2&lt;/sup&gt;</td>
</tr>
<tr>
<td>AMTD</td>
<td>arithmetic mean temperature difference</td>
</tr>
<tr>
<td>c</td>
<td>specific heat, kJ/kg-K</td>
</tr>
<tr>
<td>C</td>
<td>thermal capacitance (\bar{m} * c), kJ/K</td>
</tr>
<tr>
<td>CV</td>
<td>Closed valve</td>
</tr>
<tr>
<td>COP</td>
<td>coefficient of performance</td>
</tr>
<tr>
<td>? T</td>
<td>temperature difference, °C</td>
</tr>
<tr>
<td>? Tc</td>
<td>condenser side degradation term, °C</td>
</tr>
<tr>
<td>? Te</td>
<td>evaporator side degradation term, °C</td>
</tr>
<tr>
<td>? Tsat</td>
<td>temperature lift (cond-evap saturation temperature difference), °C</td>
</tr>
<tr>
<td>? P</td>
<td>pressure drop, kPa</td>
</tr>
<tr>
<td>? u</td>
<td>internal energy difference.</td>
</tr>
<tr>
<td>h</td>
<td>heat transfer coefficient, kW/m&lt;sup&gt;2&lt;/sup&gt;-K</td>
</tr>
<tr>
<td>hA</td>
<td>product of heat transfer coefficient and corresponding area (h*A)</td>
</tr>
<tr>
<td>OV</td>
<td>Open valve</td>
</tr>
<tr>
<td>(\bar{Q}_e)</td>
<td>evaporator capacity, kW</td>
</tr>
<tr>
<td>(\bar{Q}_{ref})</td>
<td>refrigerant side heat transfer rate, kW</td>
</tr>
<tr>
<td>SLHX</td>
<td>Suction line heat exchanger</td>
</tr>
<tr>
<td>t</td>
<td>time, s</td>
</tr>
<tr>
<td>u</td>
<td>internal energy, kJ/kg-K</td>
</tr>
<tr>
<td>(\dot{V})</td>
<td>volumetric air flow rate, cfm or m&lt;sup&gt;3&lt;/sup&gt;/s</td>
</tr>
<tr>
<td>W</td>
<td>power, kW</td>
</tr>
<tr>
<td>x</td>
<td>refrigerant quality</td>
</tr>
<tr>
<td>(\mu)</td>
<td>run time fraction</td>
</tr>
<tr>
<td>t</td>
<td>cycle period, s</td>
</tr>
<tr>
<td>(\rho)</td>
<td>density, kg/m&lt;sup&gt;3&lt;/sup&gt;</td>
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### Greek

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<td>(\mu)</td>
<td>run time fraction</td>
</tr>
<tr>
<td>t</td>
<td>cycle period, s</td>
</tr>
<tr>
<td>(\rho)</td>
<td>density, kg/m&lt;sup&gt;3&lt;/sup&gt;</td>
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### Subscripts

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<tr>
<td>air</td>
<td>air</td>
</tr>
<tr>
<td>avg</td>
<td>average (mean)</td>
</tr>
<tr>
<td>c</td>
<td>condenser</td>
</tr>
<tr>
<td>cp</td>
<td>compressor</td>
</tr>
<tr>
<td>e</td>
<td>evaporator</td>
</tr>
<tr>
<td>f</td>
<td>fin</td>
</tr>
<tr>
<td>in</td>
<td>in</td>
</tr>
<tr>
<td>init</td>
<td>initial</td>
</tr>
<tr>
<td>m</td>
<td>metal</td>
</tr>
<tr>
<td>max</td>
<td>maximum</td>
</tr>
<tr>
<td>mean</td>
<td>arithmetic mean with respect to surface</td>
</tr>
<tr>
<td>min</td>
<td>minimum</td>
</tr>
<tr>
<td>off</td>
<td>off-cycle</td>
</tr>
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</table>
on  on-cycle
out  out
ref  refrigerant
sat  saturation
sc  short-cycling
t  tube
vs  variable speed
Chapter 1: Introduction

1.1 Motivation
Previous ACRC projects investigated rapid cycling as a method of capacity control and produced promising results showing that on/off cycle periods as long as 30 seconds can produce COP’s comparable, if not higher to those achievable with variable-speed compressors (Ilic et al. 2001). This project builds on those results, and explores ways of designing systems and components for rapid cycling operation; however the implications of cycling in the design of compressors and electrical components of the system are beyond the scope of this project. Even though the experimental program only includes residential systems, most findings all are also relevant for automotive applications where the compressor is rapid-cycled using the clutch. This project assumes that cycling is possible by many means, some of them already in the market, and some still to be developed.

1.2 Background
Since air conditioning (A/C) and refrigeration systems are usually specified to provide enough cooling capacity at the maximum heat load during the year at a specific location; systems rarely operate at such condition because the loads are almost always substantially lower than the peak. That introduces the need to regulate the system’s capacity to provide adequate temperature and humidity control. The most commonly used methods for capacity regulation are conventional (long) cycling and variable speed or displacement compressors. The most common form of the long cycle on-off controller is the dead band thermostat, which cycles the system in periods on the order of 10 to 100 minutes. Variable speed systems are capable of providing the required cooling capacity maintaining a specified temperature using a variable frequency drive and compressor that operate continuously at a reduced speed. Variable speed systems achieve higher efficiencies than conventional (long) cycling systems since the temperature lift at which they operate is reduced by using the heat exchangers continuously, however the addition of the variable frequency drive increases considerably the cost of variable speed (VS) systems.

In order to achieve efficiencies comparable to those of variable speed at reduced cost, Ilic et al. 2002 studied the effects of a different capacity regulation method: rapid cycling. In rapid cycling the compressor is turned on and off in cycles whose lengths are shorter than the thermal time constant of the heat exchanger, such that the heat exchangers do not reach ambient temperature and are thus utilized passively while the compressor is off, reducing on-cycle temperature lift and increasing COP. Ilic et al. identified the four most important factors that account for the difference between cycle efficiencies of systems with continuous (variable speed) and rapid-cycling compressors: 1) refrigerant-side temperature difference (heat transfer resistance); 2) refrigerant-side pressure drop; 3) heat exchanger time constant; and 4) refrigerant backflow through the compressor during off-cycle. Based on the results from the previous project some configurations were tested, namely: 1) direct expansion (DX); 2) flash gas bypass (FGB) 3) high side receiver and 4) high side receiver with suction line heat exchanger (SLHX) all with a microchannel evaporator. The findings, results and conclusions of those experiments and some related analyses constitute the body of this report.
1.3 Contents of the report

Chapter 2 includes an explanation of the experimental setup, and the criteria used in its selection. Chapters 3, 4 and 5 describe the effects of several design variables on the performance of rapid cycling systems, with one chapter dedicated to each of the main loss terms. Results from simulation and experiments on several types of superheat controllers and control strategies studied are presented. Finally, Chapter 6 presents some guidelines and recommendations for the design and optimization of systems for rapid cycling.
Chapter 2: Experimental facility

The experimental facility used in this project is the essentially the same that was used for ACRC CR 43. A thorough description of the original facility can be found in the report by Ilic et. al. 2001. Several modifications have been made to accommodate MCHX’s and to determine whether the degradation processes are the same as those observed in conventional HX’s. The modifications were designed: 1) to make the rapid cycling data as comparable as possible to that obtained with conventional HX’s previously available; 2) to be able to feed nearly saturated liquid to the evaporator during the entire cycle period for all test conditions; 3) to maintain a saturated evaporator outlet throughout the cycles; 4) to quantify the effects of refrigerant backflow through the compressor during off cycle; and 5) to be able to perform comparisons on the same system when only the capacity modulation method is changed (VS vs. RC). A brief description of each requirement and its solution is presented below, for more detailed versions please consult the appendices.

2.1 Comparability of data

Due to the fact that refrigerant maldistribution in microchannel evaporators is known to cause significant variations in metal temperature across the heat exchangers it was decided that a single pass flooded evaporator would provide the most consistent data for comparison purposes. Since the wind tunnel was already in place with the copper tube aluminum fin heat exchanger installed and could not be modified since it sits on top of another wind tunnel in the environmental chamber, packaging constraints limited the dimensions of the HX. With so many restrictions it was not possible to find one that fulfilled all the requirements. The closest was a 1 ton single pass 3 slab CO2 gas cooler which from now on will be referred to as the microchannel heat exchanger (MCHX) or evaporator.

2.2 Managing evaporator inlet quality

The main reason for maintaining the evaporator inlet as liquid or at a very low quality is refrigerant maldistribution in the MCHX. In commercial applications refrigerant maldistribution is to be avoided because of the losses associated with it (up to 30%; see Kulkarni et al. 2003). In our case we not only wanted to eliminate maldistribution losses but also wanted to have an isothermal evaporator in order to facilitate accurate measurement of the nonlinearity of surface temperatures. Maldistribution causes significant metal temperature variations across the HX. Therefore adequate refrigerant distribution not only guarantees optimal system performance but also accurate loss estimation of cycle losses.
2.3 Maintaining 2-phase at the evaporator exit

Maintaining a 2 phase evaporator outlet is important for two reasons. First, it guarantees more efficient use of the evaporator, and second it provides uniform temperature across it. In order to prevent compressor flooding while maintaining a two phase outlet at the compressor suction line (internal) heat exchanger (SLHX) was installed. With a fixed valve opening the SLHX made the system unstable, so a steady state could not be maintained with a single speed compressor. Therefore a PC-controlled EEV was added.

2.4 Refrigerant backflow

Refrigerant leakage through the compressor during the off cycle is considered to be one of the main factors that could degrade system performance (up to 5% of COP). The compressor used by Ilic 2002 had an internal valve that prevented backflow from occurring. In order to experimentally determine the effect of leakage, the modified facility includes a solenoid valve on the compressor suction line that can be closed during off cycles to prevent leakage. It can be left open to quantify its effects on COP.

2.5 Variable speed compressor

One of the main concerns was that there was not enough data for variable speed systems that would enable us to perform direct comparisons at different load conditions. Since the only single pass MCHX available had a capacity of about 1 ton, it was necessary to use a variable speed compressor with a capacity of around 1 ton that could also be cycled. There were concerns on the effects of the high pressure differential startup, which may cause
excessive heating due to the high starting currents, mainly in reciprocating compressors. But due to the impossibility of obtaining 3 phase ~1 ton rotary or scroll compressors that could be cycled and used as variable speed the only option was to use a Copeland semi-hermetic KANA-0075 reciprocating compressor.

2.6 Automatic controller for the EEV

The SLHX produced instabilities that made it impossible to maintain steady state operation when the valve opening was fixed. Modifications were needed to implement different controlling strategies for suction superheat. Therefore a digital/analog output board (Agilent E-1320A) was installed into the E1300B VXI mainframe in order to implement closed loop controllers along with the Sporlan TCB valve controller board. The E-1320A running on the existing VXI mainframe does not provide real time control capabilities, but turned out to be cost-effective for our experimental purposes when compared to real time applications.

2.7 List of references

Chapter 3: Off-cycle dryout minimization and superheat control for RC operation

This section is devoted to the explanation of off cycle dryout and its effects on the performance of rapid cycling systems. Some design strategies and system configurations that may prove useful for reducing it are explored and explained. Also some advanced control strategies developed to reduce it are presented along with experimental results.

3.1 Description of off-cycle dryout and its relation to superheat

Figures 3.1 to 3.8 will be used to show evidence of off-cycle evaporator dryout and explain its relation to superheat; they were obtained using our microchannel evaporator in flash gas bypass mode with a SLHX in place for the purpose of ensuring good refrigerant distribution across the evaporator header. For more information on the setup please refer to Chapter 2 and Appendices A, B and C. The evaporator is being fed saturated liquid and the outlet may be 2 phase since a low effectiveness (~50%) 1.5 m long SLHX ensures that the compressor is not flooded.

Off-cycle dryout occurs when the refrigerant in the high and intermediate quality regions of the heat exchanger boils and the walls dry as warm passes on the outside of the heat exchanger while there is no refrigerant flow. Figure 3.1 presents a superheat profile of our microchannel evaporator and Figures 3.2 to 3.8 correspond to infrared images of the evaporator during the same cycle. The system is running at a cycle period of 70 seconds, 42 seconds off and 28 seconds on, at approximately 40% of the design capacity.

At the end of the off-cycle, the evaporator has warmed as air has been flowing over it for the last 42 seconds, while there is no refrigerant flow. When the compressor is turned back on again at t=0 and the saturation temperature decreases, the refrigerant outlet temperature (immersion thermocouple) follows it for a short time as the liquid sitting in the outlet header exits the evaporator until superheated refrigerant coming out of the dried portions of the evaporator reaches the outlet thermocouple. This causes the superheat spike that occurs between 5 and 12 seconds into the on-cycle.

Figures 3.2, 3 and 3.4 illustrate clearly how the superheated refrigerant is removed from the evaporator causing the superheat spike seen in Figure 3.1. Figure 3.2 is the initial state, Figure 3.3 represents an intermediate step in the rewetting of the evaporator in which the lower part of the heat exchanger has already been cooled and wetted. Figure 3.4, taken at t=15 shows the evaporator walls completely wet again and no superheat present in the evaporator.

The evaporator walls will remain wet during the remaining portion of the on cycle (Figures 3.4 and 3.5). Excess refrigerant would be left in the evaporator to evaporate during the off-cycle as well, in an attempt to minimize dryout. Due to that effect, the evaporator remains relatively isothermal (hence wet) for about 20 seconds after the compressor is turned off. (Figures 3.6 and 3.7). However, after those 20 seconds, the evaporator walls warm up considerably and the refrigerant boils off the walls of the upstream slab of the 3-slab evaporator. Figure 3.8 illustrates an intermediate step of this process, opposite to that taking place in Figure 3, in this case the higher quality region is already superheated while the bottom of the evaporator still has some refrigerant to boil, finally all the refrigerant in this slab (1 of 3) has boiled and the metal is warmer by the time the on cycle starts again at t=70 when the compressor is tuned back on again (Figure 3.2)
From the previous data it should be clear that superheat can be used as a means to estimate, at least qualitatively, dryout in the evaporator at any time during the cycling operation taking into account the time lags associated with refrigerant transport.

Another way to interpret this data is to envision the counter flow evaporator tubes acting as heat pipes where refrigerant evaporates on the warm upwind end and condenses on the cold downwind end. Here an opportunity for improvement arises with the possibility of turning around the evaporator to a parallel flow configuration where the heat pipe effect can be delayed by the larger refrigerant inventory present in the upstream slab.

Figure 3.1 Evaporator and compressor superheat profiles (28 s on 42 s off)

Figure 3.2 Infrared image of microchannel evaporator, t=0 (28 s on 42 s off).
Figure 3.3 Infrared image of microchannel evaporator, t=8 (28 s on 42 s off)

Figure 3.4 Infrared image of microchannel evaporator, t=15 (28 s on 42 s off)
Figure 3.5 Infrared image of microchannel evaporator, t=28 (28 s on 42 s off)

Figure 3.6 Infrared image of microchannel evaporator, t=42 (28 s on 42 s off)
3.2 Effects of dryout and superheat on system performance

Since most of the heat transfer in rapid cycling takes place during the on-cycles, Illic et al. (2001) expected the temperature difference between metal and refrigerant to be larger than in variable speed by a factor of \(1/\mu\), although the higher heat transfer coefficients arising from higher refrigerant flow rates may decrease that effect. Moreover this temperature difference was expected to remain constant (independent of cycle period) for a given
runtime fraction. However that was not the case, and the average refrigerant-metal temperature difference was found to be dependent on the length of the off-cycles. Figure 3.9 shows that for off-cycle periods longer than 20 seconds, the refrigerant side heat transfer coefficient decreased significantly.

Figure 3.9 Refrigerant side temperature difference

Off-cycle dryout affects system performance in the same manner for conventional cycling systems, where the absence of liquid refrigerant to boil and absorb the incoming heat means that the heat exchanger metal gets warmer during the off-cycle. Therefore on-cycle evaporating temperature must be lowered (with the corresponding penalty in pressure lift) in order to maintain the same average metal temperature and capacity. Once the on cycle starts, the walls have to be cooled down and re-wetted causing additional temporary losses in heat transfer coefficient. It can be seen in Figure 3.10 that after about 20 seconds of operation with fixed valve opening, the refrigerant outlet temperature decreases by about 2 degrees to a new state. It is consistent with Figure 3.4 where at t=20 the evaporator is finally completely cooled and re-wetted.

Figure 3.10 Refrigerant side temperature difference during 28 s on-cycle
As with VS and conventional cycling, specific compressor work (kJ/kg) increases as the suction inlet state is pushed farther into the superheated region in rapid cycling operation. Simple calculations show that due to this effect specific compression work increases by about 0.5% per kg-K for most refrigerants. Due to this effect COP can decrease up to 0.5% per degree C.

Furthermore, in the absence of a SLHX, high superheat degrades evaporator performance as a consequence of lower heat transfer coefficients for superheated vapor versus those in the two-phase evaporating zone (Hrnjak et al, 2001). As a result of lower refrigerant side heat transfer coefficients lower refrigerant temperatures are required to keep the metal at a constant temperature in order to achieve the same capacity, increasing temperature lift and degrading performance. On the other side, liquid carry-over is also undesirable since it represents a loss of refrigeration effect where it is needed, in the evaporator (Hrnjak et al 2001), and also because excessive flooding may damage the compressor.

In rapid cycling superheat control is much more complex due to the transient behavior of the system at all times, the characteristic non-linear behavior of refrigeration systems and also their slow response. For a detailed discussion of the evaporator response to expansion device opening please refer to appendix I.

In rapid cycling, even if the evaporator walls are wet during, and at the end of the on cycle as in Figures 3.4 and 3.5 and 3.11a, during off-cycle as air is still being blown over the evaporator, the metal temperature increases causing the liquid refrigerant that is sitting in the tubes to evaporate. Some of the tube surfaces will dry out as the off-cycle progresses, (Figures 3.7, 3.8, and 3.11b) starting at the point where x=1 and subsequently drying upstream. During the off-cycle dryout increases the size of the evaporator superheated zone as the metal dries and more rapidly approaches the air temperature. Then during the next on-cycle the process needs to be reversed and the metal needs to be cooled again starting at the x=1 position and going downstream. In order to cool the metal it is necessary to boil excess refrigerant which also takes some time to arrive from the valve.

![Figure 3.11 Schematic of evaporator tubes](image)
3.3 System configurations and strategies for minimizing dryout and superheat

In this section the results of all the strategies and configurations used are presented, also some configurations not tested up to date are discussed. Some of the strategies are very simple and provide considerable benefits, while others are not. There are two main ways of controlling superheat: passively and actively. Passive control implies re-designing the systems such that it inherently meets the required criteria at most operating conditions, and no external control action is needed. The second method is active control, which requires the addition of a mechanical or electrical component that implements the control algorithm. At first, passive control methods are described followed by the active ones.

3.3.1 Passive control strategies

3.3.1.1 Addition of a high side receiver

In conventional systems the high side receiver fixes the outlet condition of the condenser as saturated liquid, and guarantees full utilization of the condenser surface at all operating conditions. This prevents excessive pressure drop and COP losses occurring in undercharged systems where refrigerant does not fully condense. This also prevents subcooling in overcharged systems which store excess refrigerant in the condenser, reducing its effectiveness. Finally in rapid cycling systems where no suction line heat exchanger is present, it guarantees liquid flow to the expansion device at all times. This prevents two-phase refrigerant from clogging the expansion valve and starving the evaporator during the transients of rapid cycling.

3.3.1.2 Addition of a low side receiver

For steady state conditions the addition of a low side receiver is a way to achieve proper control of evaporator exit state by maintaining the outlet conditions of the evaporator as saturated vapor. However this is not necessarily true, due to the transient nature of rapid cycling. The response of the system to rapid cycling is postulated as follows. Assume that at the end of the on-cycle the refrigerant at the exit of the evaporator is saturated vapor. During the off-cycle the refrigerant in the evaporator continues to absorb heat. Because of boiling and gravity effects, this results in regions of superheated vapor and pooling of liquid refrigerant. At the start of the on-cycle, vapor is pulled out of the receiver by the compressor, and liquid begins to flow through the evaporator. Eventually the regions of superheated vapor will disappear, but during the interim the control objective of maintaining the walls of the evaporator wet is not achieved. In general, only vapor will reach the compressor, thus meeting one of the two control objectives.

![Figure 3.12 Air conditioning cycle with high side receiver](image)

The key issue is that the addition of the low side receiver serves to protect the compressor from flooding but does not address any of the causes of dryout therefore it is not expected to improve the efficiency of rapid
cycling. Experimentally verifying the dynamic response of the system with a low side receiver will be the subject of future investigations.

Figure 3.13 Air conditioning cycle with low side receiver

3.3.1.3 Addition of a flash gas bypass tank
Using a flash gas bypass in rapid cycling systems is beneficial for microchannel evaporators in which the inlet quality/void fraction determines how well refrigerant is distributed among the numerous parallel ports and tubes. Good refrigerant distribution is obtained by ensuring that saturated liquid is always fed into the evaporator. However, operating with superheat at the outlet of the evaporator degrades performance as in other systems. Thus a suction line heat exchanger may be used to allow the evaporator to operate with a 2-phase exit state.

Figure 3.14 Air conditioning cycle with flash gas bypass

3.3.1.4 Addition of a suction line heat exchanger
The addition of a suction line heat exchanger is one of the simplest yet effective ways of improving performance of rapid cycling systems. A suction line heat exchanger allows two-phase refrigerant to flow out of the evaporator without flooding the compressor. That means that the average quality will be lower, the liquid inventory in the evaporator higher, and the walls would be wetter. For example, in a heat exchanger using R134a, evaporating at 8.5 C, for a given volume the mass inventory increases by 34% when the SLHX is installed. Since the average quality is lower the liquid inventory increases by about 80%. As in conventionally modulated systems the heat exchanger area is used more effectively during the on cycles. Additionally for microchannel heat exchangers, the use of a suction line heat exchanger helps to improve distribution as the liquid is subcooled and its void fraction reduced at the inlet of the evaporator. The penalty is an increase in pressure drop in both the suction and liquid lines. Additionally there are gains associated with changing the thermodynamic cycle when using a SLHX, but those benefits are refrigerant specific, and for some refrigerants there are losses.
However, using a suction line heat exchanger increases the control difficulties in rapid cycling by further distancing the actuator and sensor (two components between valve and superheat measurement), and further slowing the response due to the thermal capacitance of the internal heat exchanger.

3.3.1.5 Addition of a flood tank
The addition of a flood tank is the ideal theoretical solution to maximize heat exchanger performance in rapid cycling systems since the evaporator walls remain wet at all times, and also because the compressor is always fed with saturated vapor. Additionally the liquid stored in the tank and evaporator further increases the time constant of the low side of the system improving efficiency at longer cycles. However, this configuration presents many practical difficulties such as ensuring that oil returns to the compressor and does not get trapped in the tank. For that purpose “J” tubes are usually installed into flood tanks ensuring that liquid refrigerant carries some oil out of the tank. Ideally, the liquid refrigerant will evaporate in the suction line as the oil-refrigerant mixture returns to the compressor. Also, since this configuration depends on gravity to drive mass flow through the evaporator, proper installation of the components would be critical. This configuration will also be the subject of future investigations.
3.3.2 Active control strategies for dryout and superheat

The following pages deal with the development and testing of active control strategies for superheat in rapid cycling systems. Although originally outside of the original scope of this project, several sophisticated control alternatives were studied and implemented with the available EEV and its controller board: 1) Proportional, Integral and Differential controller (PID); 2) TXV Thermostatic expansion valve 3) Feed Forward controller (FF) and 4) Iterative Learning Controller (ILC). These sophisticated control strategies were developed in a joint effort of this project along with project # 123. Presented here are the main results related to rapid cycling. A brief and non-technical explanation of the control algorithms is also presented but for additional information on controller design please refer to Appendix J. The controllers were developed in Matlab with Quanser as a real time engine. The setup can be seen in Appendix E.

Some of the strategies previously discussed such as the addition of a suction line heat exchanger and high side receiver should be used along with control mechanisms that can provide the adequate amount of refrigerant at all times to guarantee that the evaporator does not dry out even at the end of long off cycles and also that no liquid is carried to the compressor.

The only way to reduce average superheat at a given condition is to decrease the amplitude of the oscillations as shown in Figures 3.18a and 3.18b where from a high superheat condition in Figure 3.18a, reducing the amplitude allows to reduce the average as in Figure 3.18b. The best controller should be able to cancel the oscillations and allow setting the system output at the minimum stable point.

![Figure 3.18 Effect of reducing the amplitude of the oscillations](image)
Before describing in detail the control algorithms implemented a brief description of the system dynamics associated with rapid cycling will be presented.

**3.4 Qualitative description of system dynamics**

The basic control problem associated with compressor rapid cycling can be best demonstrated with a thought experiment. After the initial start-up phase of operation, rapid cycling the compressor could modulate the capacity of the air conditioning system. The frequency of cycling would be determined in part by the thermal capacitance of the heat exchangers and typically would be between 20 and 80 seconds cycle time (Ilic et al. 2001). When the compressor cycles off, heat continues to be absorbed by the metal and refrigerant in the evaporator. But as mass continues to flow through the valve and liquid evaporates, the pressure in the low side increases reducing superheat. However, it is likely that liquid refrigerant would begin to collect in the lower sections of the evaporator.

When the compressor cycles on, the low side pressure is reduced and the refrigerant that evaporated during the off-cycle becomes superheated. Right after the superheated vapor has been evacuated from the evaporator the refrigerant that pooled in the lower sections of the heat exchanger reaches the outlet. Depending on the amount of refrigerant that pooled one of the following three scenarios are possible: 1) the liquid cools the walls and leaves as saturated, or slightly superheated vapor (the ideal case), 2) not enough liquid pooled and/or remained in the walls, therefore there is not enough liquid to cool the entire tube length and the superheated region remains present even during the on cycle (Figure 3.19), and 3) excessive liquid pools in the evaporator during the off-cycle does not evaporate completely as it flows through the evaporator, leaving as a two-phase mixture (Figure 3.20).

**Figure 3.19** High superheat case, effects of pooling are small

**Figure 3.20** Low superheat, notable effects of pooling
The task is to deliver the required amount of liquid at every point in time to guarantee that the walls remain wet during on and off-cycles. There are two factors that make this task complex. First, the response of superheat to valve opening is considerably slow (Figures 3.21 and 3.22). There is a delay of approximately 12 seconds and a first order time constant approximately 100 seconds, this is significantly longer than the 5 second time constant of the compressor (Figures 3.23 and 3.24) or the 40 second cycle period. Note that the change in superheat due to the valve opening is caused by decay of the outlet temperature while the saturation pressure remains approximately constant.

Second, the response of superheat to compressor cycling is instead almost immediate, and of different nature as can be seen in Figures 3.21 and 3.23. Note that refrigerant outlet temperature is almost constant whereas the pressure drop is what causes the superheat oscillations, exactly the opposite to the previous case.

Figure 3.21 Slow response of superheat response to valve opening

Figure 3.22 Slow response of mass flow rate response to valve opening

Figure 3.23 Fast response of superheat to compressor cycling
3.4.1 Active control methods
The following control algorithms described were implemented. Superheat was measured using an immersion thermocouple at the evaporator outlet, and a pressure transducer to determine saturation pressure/temperature. Mass flow rate was regulated by an EEV linked to a PC with the desired controller running. The control setup is presented in appendix E.

![Figure 3.24 Fast response of mass flow rate to compressor cycling](image)

3.4.1.1 Feedback control strategies
The previous sections have discussed the principal control objective for this study, namely to regulate evaporator superheat to a constant value while the compressor is being rapid-cycled. In the following two sections four different control strategies are discussed; these include: thermostatic expansion valve emulation, proportional-integral control, feedforward control, and iterative learning control. The first two control strategies are feedback control algorithms used to regulate the average superheat response, but fail to eliminate the oscillations due to rapid cycling. The latter two control strategies are feedforward control algorithms designed to remove the undesirable oscillations and are discussed in Section 5.

![Figure 3.25 Feedback control block diagram](image)

3.4.1.1.1 Thermostatic expansion valve emulation
The thermostatic expansion valve (TXV) is a traditional superheat regulation mechanism. The dynamic performance of the TXV has been the subject of several investigations [3, 4, 9, 15]. This time we wanted to test the dynamic performance of the TXV in relation to rapid cycling. However, to facilitate the implementation of more advanced control algorithms, the experimental system used an electronic expansion valve (EEV) to control mass flow entering the evaporator. To compare the performance of alternative control algorithms against the more traditional control mechanisms of a thermostatic expansion valve (TXV), it was requisite to emulate the dynamic behavior of a TXV using an EEV. Dynamic models of TXV's have been developed previously. For the purposes of this study, only the linear first order sensor dynamic is included. The valve opening is related to superheat using the...
relationship in Equation 3.1, where the measured value of superheat is modeled as given in Equation 3.2 or in transfer function form as in Equation 3.3. This model assumes a first order response, where the time constant, $\tau$, is selected to be 5 seconds, and where $K_{TXV}$ is an adjustable parameter. This model obviously does not include the many nonlinear phenomena that would be present in an actual TXV, but since many of these would result in decrease in stability and performance, this TXV emulation can be considered the best possible case for this control mechanism.

$$A_v = K_{TXV} (T_{\text{desired}}^{SH} - T_{\text{measured}}^{SH})$$  \hspace{1cm} (3.1)

$$T_{\text{measured}}^{SH}(t) = \left(\frac{1}{\tau} e^{-t/\tau}\right) T_{\text{err}}(t) - T_{\text{ero}}(t)$$  \hspace{1cm} (3.2)

$$T_{\text{measured}}^{SH} = \left(\frac{1}{1 + \tau s}\right) T_{\text{err}} - T_{\text{ero}}$$  \hspace{1cm} (3.3)

### 3.4.1.1.2 Proportional integral control

Proportional-Integral-Derivative (PID) control is a widely known and implemented control algorithm. The continuous time version of this control algorithm is shown in Equation 3.4 where the error, $e(t)$, is given by the desired value of superheat minus the measured value of superheat (Equation 3.5). This algorithm varies the valve opening proportional to the error, the integral of the error, and the derivative of the error. The gains for each of these terms is selected or “tuned” by the user. This can be done experimentally, or using a model of the system response. The Laplace transform of this control algorithm is shown in Equation 3.6 and is equivalent to placing a pole at the origin, and two arbitrary zeros. Assuming a trapezoidal integration rule, the discrete time version is given in Equation 3.7.

$$A_v = K_p e(t) + K_i \int e(t) + K_d \dot{e}(t)$$  \hspace{1cm} (3.4)

$$e(t) = T_{\text{desired}}^{SH} - T_{\text{measured}}^{SH}$$  \hspace{1cm} (3.5)

$$G_p(s) = K_p + \frac{K_i}{s} + K_ds$$

$$= \frac{K_{PID}(s + a_1)(s + a_2)}{s}$$  \hspace{1cm} (3.6)

$$G_p(z) = K_p + K_i \frac{T_s}{2} \left[ \frac{z + 1}{z - 1} \right] + K_d \frac{2}{T_s} \left[ \frac{z - 1}{z + 1} \right]$$

$$= \frac{K_{PID}(z + a_1)(z + a_2)}{(z + 1)(z - 1)}$$  \hspace{1cm} (3.7)
A discrete time model for the effect of valve opening on superheat has been approximated in the previous section as a first order response with a time delay. This can be represented in discrete transfer function form as shown in Equation 3.8.

\[
G_{\text{valve}}(z) = z^{-d} \left( \frac{K_{\text{valve}}}{z - b_1} \right)
\]

(3.8)

Because of the multiple delays in Equation 3.8 the use of derivative control will result in instability. The integral term does require the use of smaller gains to ensure stability, but also guarantees zero steady state error in tracking the desired level of superheat. However, the reader should note that even with this feedback controller, the valve response is slower than the compressor, and will be unable to reject the disturbances caused by the compressor. This feedback control will simply drive the average superheat response to the desired value. The reader should also note the connection between the TXV control and the proportional control, namely that the TXV control is a proportional control with an additional sensor dynamic.

### 3.4.2 Feedforward control strategies

Because the superheat reacts much faster to compressor changes than to valve changes, a feedback controller will be unable to reject the disturbance to superheat caused by compressor cycling. However, because the nature of the compressor signal is known a priori, a feedforward controller can be used to reject this disturbance.

![Figure 3.26 Feedforward control block diagram](image)

#### 3.4.2.1 Feedforward control

Assuming that the effect of compressor on superheat is approximately a first order response (Equation 3.9) as identified in Section 3, the ideal feedforward controller would be a negative compressor model multiplied with the inverse valve model, as shown in Equation 3.10. Note that in order to implement this control algorithm, the compressor signal must be known \(d\) time steps in advance.

\[
G_{\text{comp}}(z) = \left( \frac{K_{\text{comp}}}{z - b_2} \right)
\]

(3.9)

\[
G_{\text{ff}}(z) = -\frac{G_{\text{comp}}(z)}{G_{\text{valve,approx}}(z)} = -z^{-d} \left( \frac{K_{\text{comp}}}{K_{\text{valve}}} \right) \left( \frac{z - b_1}{z - b_2} \right)
\]

(3.10)

This approach can be implemented two ways. First the delay can be estimated, and the combined gain and pole-zero locations can be tuned manually. Second, approximate models can be estimated offline as in Section 3. Using the
1st order compressor and valve models identified in Section 3, the resulting feedforward controller is given as in Equation 3.11.

\[
G_f(z) = z^{24} \begin{pmatrix} 651.3 & (z - 0.9950) \\ z - 0.8865 \end{pmatrix}
\]  \hspace{1cm} (3.11)

To demonstrate the effectiveness of this control algorithm, the identified models (Equations 3.9 and 3.10) are implemented in simulation, and the calculated feedforward control law (Equation 3.11) is applied. Because the valve model used in the calculation of the feedforward control law is approximate, the resulting control action does not perfectly eliminate the oscillations caused by rapid cycling, but their magnitude is decreased significantly, as seen in Figure 3.27.

The resulting valve action appeals to physical intuition. Because the compressor will increase superheat when it turns on, the valve attempts to anticipate this change by allowing a large amount of refrigerant mass to enter the evaporator. Ideally, this refrigerant mass would arrive at the compressor at the beginning of the on cycle, thus counteracting the rise in superheat. Likewise the valve attempts to anticipate the decrease in superheat during the off cycle by closing the valve and starving the evaporator.

![Figure 3.27 Fixed feedforward control](image)

This subsection demonstrates the effectiveness of feedforward control in decreasing the oscillations in superheat caused by rapid cycling. However, the parameters of this feedforward control law will be system dependent. An alternative method in determining the appropriate feedforward control law would be to estimate the compressor and valve models online and simultaneously implement the feedforward control. This adaptive approach offers the advantages that no tuning is required, and the ability of the controller to adapt to different operating conditions. However, the task of identifying a Multi-Input Single-Output (MISO) system is not trivial. The identification algorithm must simultaneously identify the dynamics associated with all inputs, and prevent the estimated parameters from drifting from their true values when the measured outputs are maintained at a constant value (the objective of the control algorithm). The latter difficulty is well known as “lack of excitation” and is an inherent difficulty when combining online identification and control.
One method to identify a MISO system is to parameterize the system as done when using the Boot-Strapping algorithm. To ensure that the parameters don’t drift when there is no new dynamic information, a variable forgetting factor could be augmented to the identification algorithm. When the identification error is large, the algorithm adapts quickly, and when the error is small, the parameters remain virtually unchanged. Then a classical recursive identification routine could be used. However, regardless of how the parameters are estimated, because this is a linear feedforward control law, we would not expect to compensate for the nonlinear effects such as refrigerant pooling. Additionally, the adaptive form of this control law requires “persistence of excitation” to converge to the correct parameters. Simple signals such as the on-off signal applied to the compressor may not excite the system sufficiently to correctly identify the dynamics. In this case, the controller will result in suboptimal performance. One possible means of avoiding these difficulties is by using a nonlinear control law such as iterative learning control.

3.4.2.2 Iterative learning control

Iterative learning control (ILC) is a relatively new control technique that can be used to improve the transient performance of systems that operate in a repetitive manner. The technique is motivated by the observation that if a system performs the same action repeatedly, a controller should be able to learn from the errors of the previous iterations to improve the performance during the current iteration. As explained in [13], the scheme is as follows. For the kth iteration, an input signal, $u_k(t)$, is applied to the system resulting in an output signal, $y_k(t)$. Based on the error between the desired output and the actual output, $e_k(t) = r_k(t) - y_k(t)$, the ILC algorithm computes a modified input signal for the next iteration, $u_{k+1}(t)$.

This algorithm is also implemented in simulation successfully. To demonstrate that this algorithm has the ability to compensate for nonlinear dynamics, a nonlinear dynamic is added to the identified models similar to the observed effect of refrigerant pooling as discussed in the previous section. As shown in Figure 3.28 during the on-cycle a sharp decrease in superheat is observed as the slug of liquid refrigerant that pooled at the bottom of the evaporator reaches the temperature sensor.

![Figure 3.28 Simulated superheat response to compressor cycling nonlinear effect](image)

When used in conjunction with the fixed feedforward controller, the ILC control perfectly compensates for the nonlinear effect within a few cycles, as seen in Figure 3.29. The fixed feedforward control compensates for most of the effect on superheat due to compressor cycling. After three cycles, the ILC controller is applied, and almost immediately removes the nonlinear effect, as well as slowly improving the effect of the feedforward control effort.
The ILC controller is also capable of removing the oscillations in superheat without the fixed feedforward controller, but requires several more iterations before converging to the desired result, as shown in Figure 30.
3.5 Experimental results and evaluation

3.5.1 Feedback control

The implementation of the TXV and PI control strategies are successful as a means of regulating the average superheat value to a desired setpoint. As expected the TXV controller results in a steady state error in superheat regulation, whereas the PI controller has zero steady state error. However, because the effect of the valve on superheat is delayed and slow with respect to the compressor, these control algorithms are ineffective in eliminating the oscillations due to rapid cycling. Figures 3.31 and 3.32 show steady state operation with the TXV emulation. Likewise Figures 3.33 and 3.34 show steady state operation with the PI controller.

Figure 3.31 TXV emulation, mass flow rate and valve position

Figure 3.32 TXV emulation, superheat at evaporator outlet

Figure 3.33 PI controller, mass flow rate and valve position
3.5.2 Feedforward control

The primary conclusion drawn from Section 3.2.2 is that feedforward control has the ability to eliminate the undesirable oscillations caused by rapid cycling, including any nonlinear phenomenon. However, the reader will note that the practicality of the results of these simulation studies depend on a few critical assumptions. First that the behavior of the system is the same over the full range of valve openings. Second that there are no limits on valve opening.

The violation of the second assumption can be seen from Figure 3.27, 3.29 and 3.30. Recall that the expansion valve is controlled by a stepper motor that has a range of -1596 to 1596 steps. When the feedforward algorithms are used, the control algorithm requires the valve to vary between -1000 and -1700 steps to completely eliminate the oscillations. This obviously exceeds the limits on the valve, but also indicates that to effectively eliminate the oscillations, the valve would be required to move mass backwards through the valve.

Figures 3.35 and 3.36 show the implementation of the feedforward control algorithm. To avoid violating the valve limitations, the feedforward signal is initially applied at only a fraction of the calculated value. From Figure 3.35 we see that the valve is behaving as in simulation. The effect on superheat, however, is virtually negligible. The general shape of the transient oscillation is changed, but the amplitude remains the same.
Subtler problems that are associated with the underlying physics of the system also prevent the control algorithm from eliminating the oscillations. First, the effect of the valve appears to degrade as the valve opening increases. This is evident by observing the mass flow rate in response to a large change in valve opening. As seen in Fig. 35 the mass flow rate never increases beyond 42 grams/sec, despite large increases in valve opening.

Second, the heat exchanger geometry prevents the removal of superheated fluid during the off-cycle. While the compressor is off, air continues to blow across the evaporator, which absorbs heat by boiling refrigerant. Because of the geometry of the heat exchanger, during this time of limited mass flow the liquid refrigerant pools in
the lower areas without wetting the higher, dryer areas, where regions of superheated vapor form. Despite increasing the valve opening, these regions of superheat remain present until the compressor is turned on, and mass flow through the evaporator resumes. These and other physical limitations render the use of feedforward control ineffective in this application.

3.6 Conclusion
This section presents a thorough discussion of the control issues surrounding rapid cycling of vapor compression cycles. Various system configurations are presented, and evaluated qualitatively. A common system configuration using plate-and-tube heat exchangers with a high side receiver is evaluated experimentally. The principle control objectives are identified as preventing liquid refrigerant from entering the compressor and maximizing the area of liquid refrigerant contact in the evaporator during rapid cycling. Simple feedback control of the expansion valve using a TXV or PI control algorithm is shown to regulate the average superheat temperature, but is unable to eliminate the oscillations in superheat due to rapid cycling. The principal difficulties are the slow response superheat from the valve relative to the compressor. Feedforward control algorithms are shown in simulation to be capable of eliminating the superheat oscillations, including the nonlinear dynamic effects using iterative learning control. However, the inherent dynamics of the system, and the physical limitations of the valve prevent these algorithms from working successfully in practice.

3.7 List of references


Chapter 4: Pressure drop

4.1 Introduction

The second factor that degrades the performance of rapid cycling systems when compared to variable speed systems is pressure drop. In rapid cycling operation, the compressor runs at full speed during the on-cycle, therefore the steady state frictional pressure drop is constant and equal to the pressure drop at design capacity. In variable speed, refrigerant flow rate decreases as capacity decreases, reducing pressure drop as capacity decreases. Hence the losses in rapid cycling when compared to variable speed.

As shown in Figure 4.1 Ilic et al. (2001) verified that pressure drop is constant for all capacities (run time fractions and cycle periods) with a 2 ton residential system using conventional aluminum fin and copper tube heat exchangers.

![Figure 4.1 Evaporator pressure drop. 6.9 kW system](image)

4.2 Results

The same was expected to occur in microchannels, since the same reasoning applies, however two differences were found: 1) although the steady state pressure drop is constant for all cycle periods and runtime fraction the on cycle average pressure drop is influenced by start up transients and is no longer considered to be independent of runtime fraction and cycle period as before and; 2) the magnitude of the pressure drop across the evaporator was significantly greater than what could be expected on a microchannel. The explanation for the latter is that pressure drop is higher because as mentioned in Appendix C the heat exchanger used as evaporator was originally designed as a CO\textsubscript{2} gas cooler. The explanation rest of this chapter is devoted to the explanation of the former. Figures 4.2 and 4.3 illustrate condenser and evaporator average pressure drop for several runtime fractions and cycle periods.
4.3 Analysis and discussion

In both cases the trends are caused by transient phenomena occurring at the beginning of the on-cycle. Transient phenomena are present in all runtime fractions and cycle periods and have a greater weight when averaged.
over the lower runtime fractions and cycle periods, and therefore those present different average pressure drops, although the steady state pressure drop in all cases is constant at about 40 kPa in the evaporator and 7 kPa in the condenser which correspond to the steady state pressure drop at full capacity.

![Evaporator and condenser pressure drop 30 sec. on-cycle 20 sec. off-cycle](image)

The spike in the compressor occurs because: 1) the mass of refrigerant in the condenser (600 gr.) has to be accelerated from rest (to 1.5 m/s) and; 2) since the receiver was located above, a column of liquid has to be pushed upwards to the receiver. In the evaporator there is also a transient phenomenon that delays the onset of a steady state pressure drop but its origins are have not been elucidated up to date.
Chapter 5: Heat exchanger’s thermal time constant

5.1 Introduction
Another factor reported by Ilic et al. to degrade the performance of rapid cycling systems relative to variable speed systems is the heat exchanger’s thermal time constant. In rapid cycling, the heat exchanger (time-area) average metal temperatures are the same as those in variable speed, however, the on cycle temperature is lower in the evaporator and higher in the condenser, due to the exponential nature of the process. Since the refrigerant temperature is closely coupled to the metal temperature then the temperature lift is increased. As the thermal time constant of the heat exchanger is increased this effect on temperature lift is diminished. For a more detailed discussion of the non-linearity losses please refer to Ilic et al. 2001. The thermal time constant is given by equation 5.1

\[ t(s) = \frac{mC}{hA} \] (5.1)

Where
- \( t(s) \) thermal time constant of the heat exchanger (\( s \))
- \( m \) the mass of the heat exchanger
- \( C \) specific heat of the heat exchanger material (kJ/kg-K)
- \( h \) air side heat transfer coefficient
- \( A \) air side area

The time constant can be determined theoretically just by substitution of the values using appropriate air side heat transfer correlation and the dimensions of the heat exchanger, or experimentally by a pull-down or warm-up test.

In general the design of heat exchangers tends towards low thermal time constant: reducing mass, while increasing air side areas and air side heat transfer coefficients. Since the limiting factor heat transfer resistance is usually on the air side that is where design efforts are found. As environmental and economical considerations impose higher efficiency requirements the air side heat transfer resistance would have to be minimized. However that could be achieved in two different ways, increasing the heat transfer area, by adding more fins, if fin thickness is unchanged the net increase in area will be roughly proportional to incremental increase of mass, and the heat exchanger thermal time constant will not been change, but its effectiveness will, for both variable speed and rapid cycling. To increase the thermal time constant when adding air side area, the fins must be made thicker. The second way to increase the effectiveness of the heat exchanger is to increase the heat transfer coefficient on the air side by modifying the fin surface. Since additional machining is required to generate the higher heat transfer enhancements, at some point it may become effective to add more mass.

The average price of copper traded for the next is about 0.75 USD/lb or 1.65 USD/kg whereas the price of Aluminum is about 0.66 USD/lb or 1.45 USD/kg (www.metalprices.com). With densities of 8933 kg/m³ and 2700 kg/m³ and specific heats of 0.385 kJ/kg-K and 0.9 kJ/kg-K for copper and aluminum respectively, the cost of adding
an additional kJ/K in a heat exchanger is about 1.6 USD if using AL, or 4.5 USD if copper is used. That makes aluminum clearly the choice if more material is to be added (e.g. in the form of fins).

5.2 Experimental results
Since dryout was considerably more extensive than originally expected, the metal temperature became uncoupled from that of the refrigerant. Compare Figures 5.1 for an experiment with a conventional evaporator and 5.2 with the CO$_2$ gas cooler as evaporator. Since the compressor capacity was less than that of the evaporator, the nonlinearity term’s effect on temperature lift is expected to be relatively small due to the “oversized” evaporator. However, since the metal temperature became decoupled from the refrigerant saturation temperature, the refrigerant does not follow the metal temperature path during the offcycle, nor the metal follows the refrigerant during the on-cycle resulting in a very small contribution of nonlinearity to the temperature lift term (undistinguishable from measurement uncertainties). Instead the COP degradation due to nonlinearity still appears in the overall temperature lift, but as part of the refrigerant side resistance, due to the dryout effect.

![Figure 5.1](image1.png)

**Figure 5.1** Evaporator dryout, and its effect on nonlinearity. Conventional evaporator.

![Figure 5.2](image2.png)

**Figure 5.2** Evaporator dryout, and its effect on nonlinearity. MC evaporator.
Chapter 6: Conclusions and recommendations

6.1 Offcycle dryout and superheat control

Several strategies to minimize offcycle dryout and improve refrigerant distribution in micro-channel evaporators were studied: 1) direct expansion; 2) flash gas bypass; 3) suction line heat exchanger; and 4) flooded evaporator. Experimentally the use of direct expansion, suction line heat exchanger and flash gas bypass were tested experimentally. The best distribution was achieved in the flash gas bypass configuration, followed by the high effectiveness suction line heat exchanger. Theoretically the best alternative is the flooded evaporator; however it could not be experimentally tested due to space limitations in the environmental chamber. In practice oil return may become a problem in such a system.

The use of a flash gas bypass tank along with a suction line heat exchanger yielded the best results in terms of achieving good distribution. However it requires some kind of device to control the condenser outlet state at low or now subcooling, in order to prevent in the system from simply using the suction line heat exchanger as an extension to the condenser. When that occurs, the heat exchange between condensing and evaporating refrigerant reduces system capacity without reducing compressor work, and spoils the COP.

Several complex controllers were also implemented with the conventional copper tube-aluminum fin evaporator in DX mode to evaluate the importance of rapid valve response. Since the superheat responded slowly to TXV opening while its response to compressor cycling was immediate, simple TXV’s and PI control algorithms do not work successfully in practice. Simple feedback control of the expansion valve using a TXV or an EEV with PI control algorithm were both shown to regulate the average superheat at the compressor suction inlet, but both were unable to eliminate the oscillations in superheat due to rapid cycling.

Since condenser walls are always wet during the off-cycle, replacing conventional condensers with MCHX’s in a rapid cycling system is expected to have little effect on performance. There is no gain from the lower inside heat transfer resistance benefits associated with MCHX’s, and their thermal time constants are approximately equal to conventional HX’s.

The results confirmed that the best way to design systems in order to reduce dryout is then to 1) minimize the low side internal volume-to-surface ratio; 2) minimize the low side mass and volume outside the evaporator and 3) design evaporator to keep tubes wet as long as possible during the off-cycle, by orienting them to minimize drainage or to accept slow liquid drainage from inlet header. The rapid dryout of the prototype microchannel evaporator was attributable to the fact that it was originally designed as a CO$_2$ gas cooler, having far less refrigerant-side surface than one designed for a fluorocarbon refrigerant. However the performance shortfall was consistent with this area difference. The suction line heat exchanger successfully proved that evaporator inlet qualities on the order of 1% are achievable, but the prototype was not optimized to minimize low-side volume and a performance penalty was apparent.

6.2 Pressure drop

The on-cycle-averaged pressure drop was expected to be constant for all run time fractions and cycle periods. Such results were not observed, since start up transients caused variations in the time averaged pressure
drop during the on-cycle. In all cases the expected, constant steady state pressure drop was attained after the startup transients, but time-averaged pressure drop was not constant for all run time fractions and cycle periods. Further experiments are recommended to identify and eliminate the causes of the startup transients; these experiments established that they are not negligible. The best way to optimize systems to minimize pressure drop losses in rapid cycling, is to 1) increase the number of tubes and ports; 2) reduce their length and 3) design headers (especially the outlet) for low pressure drop.

6.3 Thermal time constant

Our comparison of the thermal time constants of conventional and microchannel heat exchangers found it to be about 7% higher for conventional copper tube heat exchangers.

Since a larger time constant \((mc/h_{\text{air}} A_{\text{air}})\) leads to improved performance, the best strategy was sought and identified. For a given thermal capacitance, air-side heat transfer enhancements reduce the time constant. Conversely if air-side resistance is minimized by adding fin area instead of enhancing them, the thermal time constant of the heat exchanger remains constant if area increases at the same rate as thermal mass. Therefore the best approach would be add mass by making fins thicker, increasing the time constant by increasing the denominator while leaving the numerator unchanged. The economics of adding mass instead of geometrical enhancements are beyond the scope of this project, but it seems clear that a heat exchanger having thick (efficient) plain fins would have a larger thermal time constant and would require less fan power.
Appendix A: Low side facility setup

To carry out the proposed experiments the low pressure side was designed such that it was possible to: 1) maintain low quality inlet to the evaporator to provide good distribution 2) provide saturated or superheated vapor to the compressor and; 3) maintain a uniform temperature in the evaporator.

To fulfill the above requirements three solutions were proposed: 1) flash gas bypass; 2) flooded evaporator; and 3) DX with high side receiver and suction line heat exchanger. Since it was not known a priori which of the solutions would work, the installation had to be flexible enough to allow switching from one strategy to another in a short time. Also it had to be possible to run in direct expansion (DX) mode to facilitate comparison with the previously used copper tube aluminum fin evaporator.

Considerably low (<0.01) refrigerant inlet qualities are required to ensure good refrigerant distribution in the evaporator. For example assuming homogeneous void fraction, refrigerant entering the evaporator at x=.07 would yield a void fraction around 0.8. The usefulness of the strategies employed is evident. Saturated refrigerant is fed into the heat exchanger either by using FGB or flooded evaporator. Low quality refrigerant (<0.01) can be obtained from the high effectiveness suction line heat exchanger providing better distribution than conventional DX operation.

A.1 Flash gas bypass

Flash gas bypass has been used to improve refrigerant distribution in several ACRC projects with promising results. FGB tanks were available and many unknowns about flooded evaporator systems were still unresolved at the time the setup was being built, therefore the FGB configuration was ran first. Figure A.1 illustrates the complete low side setup. It can be used to run the system in FGB, DX and flooded evaporator modes by adjusting several valves and/or changing the tank. Figure A.2 illustrates refrigerant flow when the system is run in FGB mode and Figure A.3 in DX mode. The system can be run with either of the two tanks (flooding or FGB) in place or they can be bypassed for DX operation. Changing between flooded evaporator and FGB requires changing the tank, but since all connections to and from the tank are flexible it is not a difficult task and can be performed in a few hours. Most of the work consists of evacuating the system before removing the tank, and then recharging it after installation.

Figure A.1 Low side setup
The flexibility of the tank connections also permits adjustment of the tank height to control the refrigerant level in the HX. Initially the evaporator was placed in crossflow configuration, exposing the refrigerant exit (warmest slab) to the incoming (warmest) air. That caused increased superheat in RC operation but also allowed us to take infrared pictures of the MC evaporator using FGB and DX at steady state operation. As expected the use of FGB improved refrigerant distribution considerably as can be seen in Figures A.4 and A.5 (continuous operation). Distribution when using a flooded evaporator is expected to be as good as in the FGB case since only liquid refrigerant is fed into the evaporator. These results were obtained in a counterflow configuration as it appears in all Figures.
**Figure A.4** Infrared image of MC evaporator using FGB

**Figure A.5** Infrared image of MC evaporator using DX (Inlet quality 0.09)

**A.2 Flooded evaporator**

Flooded evaporator tests have not yet been run, however, Figure A.6 illustrates the corresponding configuration. The main difficulty with installing a flooded evaporator is ensuring that oil does not get trapped in the tank, and that returns to the compressor. For that purpose “J” tubes are usually installed into flood tanks. The J tube
ensures that liquid refrigerant carries some oil out of the tank. Ideally, the liquid refrigerant will evaporate in the suction line heat exchanger as the oil mixture returns to the compressor.

![Diagram](image)

Figure A.6 Low side in flooded evaporator configuration

However, this configuration still presents many practical inconveniences and therefore was not used. But theoretically it may be one of the most beneficial for rapid cycling systems since the evaporator walls remain wet at all times, and also because the thermal capacitance of the low side increases due to the low quality (high density) refrigerant contained in the evaporator.

### A.3 High effectiveness suction line heat exchanger

The third option for ensuring good refrigerant distribution in the microchannel evaporator consists of using a high effectiveness suction line heat exchanger that subcools the saturated liquid coming out of the receiver almost to evaporating temperature (within 1 or 2 C) achieving inlet qualities low enough (<0.03) that ensure good distribution of refrigerant in the Microchannel evaporator header. The added benefit of the suction line heat exchanger is that in order to cool the saturated liquid, lower quality vapor is extracted from the evaporator, which means that dryout decreases when compared to FGB and the compressor is still protected from being flooded. The down side is of course additional pressure drop and that is only viable for certain refrigerants. It is important to mention that for MCHX’s it is imperative to use some device to lower inlet quality and therefore a very high effectiveness SLHX was used, however for conventional round tube, aluminum fin the SLHX is still a good option since it provides better use of the evaporator area during cycling operation, may add performance benefits on some refrigerants and helps to protect the compressor. Figure A.7 is an IR picture of a MCHX being fed refrigerant coming out of a high effectiveness (>0.9) SLHX. Note that distribution does not differ from that obtained by using FGB. Compare with Figure A.4.
Figure A.7 Refrigerant distribution in MCHX (SLHX)
Appendix B: Selection of the compressor

B.1 Introduction
Since refrigerant maldistribution in microchannel evaporators is known to cause significant variations in metal temperature across the heat exchangers it was decided that a single pass flooded evaporator would provide the most consistent data for comparison purposes. The only single pass MCHX available was rated at 1 ton, therefore a compressor of an equivalent capacity was required.

B.2 Requirements
The most important limitation in the compressor setup and selection came from the need to make the data comparable when the system capacity was modulated by means of VS and RC operation. Therefore in order to isolate the effects of compressor performance the same compressor is to be used for both capacity modulation strategies. That posed three problems:

- There is very limited availability of 3 ph 1-ton variable speed compressors in the US. All are the reciprocating type.
- Most of the newer VS drives and compressors use proprietary algorithms to improve performance (such as rotor position feedback), the compressor and drive are matched and in some cases the compressor can not be powered from conventional 60 Hz AC outlets or from conventional frequency controllers. In other words the proprietary controller for the specific compressor has to be used at all times.
- If a proprietary VS drive is to be used, then it is not possible to rapid cycle the compressor. All VS drives that were made available to us do not start motors at full voltage/full frequency (usually 210 V / 60 Hz) but instead ramp up the motor speed with a fixed V/Hz ratio to keep currents at appropriate levels during start up and acceleration. The acceleration rate in the best case was about 5 Hz/sec which means that in order to reach 60 Hz (full speed) it will take about 12 sec which is too long for RC operation.

At the beginning of the compressor selection process a scroll or rotary compressor was thought to be the best option in order to avoid high starting currents in RC mode. Those currents would occur since the system starts at a high pressure differential, therefore a high torque which requires a high starting power that can only come from increased currents since the voltage is fixed. However no such rotary or scroll compressor could be found. A 3 phase 0.75 ton single speed semi hermetic compressor, Copeland KANA-0075 was installed. It has run in VS mode with a Reliance Electric GP2000 VVVF drive, and in RC mode with the use of a relay, triggered by an electronic timer/switch (Eagle Signal Control SX110A5) according to pre-specified on and off cycle time periods. The mode of operation is chosen by means of a single pole double throw switch which energizes the relay corresponding to the desired mode, in the top in Fig. B.1, rapid cycling, and in the bottom variable speed.
**B.3 Efficiency of the compressor and variable frequency drive**

Since the VS drive and compressor used in this experiment will not match the efficiency of the newer VS compressors and drives, the data were generalized by estimating the power losses associated with VS operation in the drive and compressor. The following paragraphs describe the calculation of two separate energy balances on the compressor when operating at full capacity (60Hz/210V) 1) from the AC outlet and 2) from the VVVF drive.

The power input into the system is known and the isentropic work in the compressor can be calculated from the refrigerant inlet and outlet pressure condition. Losses in the VS system are of three kinds: 1) AC-DC-AC power conversion in the VVVF drive; 2) motor losses, and 3) compression losses. Losses in the conventional AC powered system are of only two kinds: 1) motor losses, and 2) compression losses.
It is not possible to distinguish between motor and compression losses. A fraction of the motor loss is dissipated as rejected heat through the shell and, the remaining fraction in the refrigerant. The same applies for the compression losses, but the total losses occurring in the compressor/motor are known.

Since the losses in the compressor/motor can be estimated in both cases and the total power input/ output are also known then it is possible to estimate the losses in the VVVF drive. Also it is possible to estimate the additional losses occurring in the motor/compressor subsystem when operated via the VVVF at steady state conditions.

The losses associated with running the system from the VVVF drive at full capacity were found to be 40 W or 6% of the overall power consumption. There was no significant change in the motor/compressor losses which accounted in both cases for about 160 W that correspond to about 25% of the total consumed power. The overall efficiency dropped from 75 to 70% when the compressor was run from the variable frequency drive.

Finally it is important to mention that a rotary compressor and two variable frequency drives were sent from Samsung for us to try, but both of the controllers were damaged in shipment despite of proper packaging and labeling. Several attempts to fix the controllers were made with no success. Also, after the above described setup was finalized and running, a 0.75 ton, 3 phase, variable speed, rotary compressor arrived from Sanyo. It has not been installed yet due to concerns of operating it with a conventional VVVF drive in VS mode, also it is rated at 140 V and a 3 ph step-up transformer is not yet available.

B.4 List of references
Sanyo Electric co. C-6RV73HOW Compressor specifications. Jul 1996
Appendix C: Selection of the microchannel evaporator

For this project it was necessary to install a new evaporator, a microchannel evaporator for the following reasons: 1) the geometry of MCHX’s is completely different than that of conventional HX’s therefore it would serve to further validate the model developed earlier (Ilic et al. 2001); and 2) because MCHX’s have their own advantages and limitations require systems specifically designed for them. Therefore the effects of rapid cycling in systems specifically designed for MCHX use could also be investigated.

Originally it was intended to install a 2-ton MCHX in the wind tunnel in series with the 2-ton conventional copper tube aluminum fin evaporator that was used in the previous project. However it was not possible to match capacities, so a smaller 1 ton MC evaporator was finally installed. The following paragraphs describe the selection process. The main factors affecting the selection of the MCHX were: 1) availability; 2) space limitations. 3) capacity 4) uniform refrigerant distribution and; 4) ability to measure a representative metal temperature.

C.1 Availability

It was intended to use of the already available MCHX’s in the ACRC to reduce the fabrication costs and time associated with getting a new MCHX. That considerably limited the amount of MCHX’s to choose from. Three MCHX’s were available from previous ACRC projects and were considered for use as evaporators in this project and are presented in table C.1.

<table>
<thead>
<tr>
<th>Evap</th>
<th>Nominal Capacity (ton)</th>
<th>Slabs</th>
<th>Tubes/slab</th>
<th>Ports/tube</th>
<th>Tube ID (mm)</th>
<th>Other</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3</td>
<td>3</td>
<td>41</td>
<td>11</td>
<td>0.79</td>
<td>CO₂ Evaporator</td>
</tr>
<tr>
<td>2</td>
<td>1.2</td>
<td>3</td>
<td>64</td>
<td>4</td>
<td>0.625</td>
<td>CO₂ Gas cooler</td>
</tr>
<tr>
<td>3</td>
<td>2</td>
<td>1</td>
<td>40</td>
<td>19</td>
<td>0.762</td>
<td>R410 Evaporator</td>
</tr>
</tbody>
</table>

C.2 Space limitations

The wind tunnel could not be moved, nor its test section changed because of its location above another wind tunnel setup. Therefore it was necessary to fit the available MCHX into the tunnel without major modifications.

Evaporators 1 and 3 would only fit into the wind tunnel if their headers were placed vertically and the ports horizontally as in Fig C.1a, or if their headers were placed horizontally and their tubes tilted as in Fig C.1b. For experimental purposes it is possible to use either horizontal or vertical tubes because earlier experiments showed that the offsets of rapid cycling on system performance are identical for systems having wet or dry coils. Therefore the experiments could be carried out using only dry coils, so evaporator orientation need not be constrained by condensate drainage considerations. Evaporator 2 being a 3-slab, single pass could be fitted into the wind tunnel without significant modifications as shown in Fig C.1c and was the one eventually selected.
Figure C.1 Placement options for evaporators 1, 2 and 3.

There are several problems with tilting the evaporators. Song et al. 2002 found that for angles smaller than 45 degrees from vertical it is still possible to extrapolate the results to a vertical evaporator, but when the angle is larger than 45 then pressure drop and heat transfer coefficients are degraded substantially. Evaporators 1 and 3 would have had to be tilted at 60 and 65 degrees off the vertical, which would yield data with considerably higher pressure drop and different heat transfer coefficients than what would be expected from a conventional MCHX.

C.3 Capacity
Evaporator 1 is a 3 slab, 3 ton evaporator, so two slabs would provide 2 ton capacity. However if two slabs were used (installed in series in the airflow direction) then due to unequal loading it would have been difficult if not impossible to determine how much refrigerant flowed through each. Evaporators 2 and 3 were single pass, single slab, nominally 1.2 and 2.5 tons.

C.4 Uniform refrigerant distribution
Uniform refrigerant distribution is essential for adequate performance of MCHX. In order to achieve good refrigerant distribution a flash gas bypass tank was installed, however for vertical headers the pressure drop across the ports would have to be considerably higher than the gravitational head pressure in the vertical header, otherwise a considerable amount of the liquid refrigerant would flow through the lower tubes (an order of magnitude is considered to be enough for good refrigerant distribution as a rule of thumb). For evaporators 1 and 3 the estimated pressure drop across the ports was calculated using the Souza and Pimenta correlation and compared to the gravitational pressure drop across the header. Since the two were almost equal, it would result in maldistribution. Due to its shape evaporator 2 could be placed in the wind tunnel with its header in horizontal position as shown in Figure C.1c thus avoiding the pressure drop limitations regarding gravitational versus frictional losses.
**C.5 Ability to measure a representative metal temperature**

It is important to get a good measurement of the HX metal temperature because quantifying the asymmetry (non-linearity) loss term requires accurate measurement of the time-dependence of the average metal temperature. Therefore if the heat exchanger is isothermal it is very easy to estimate an average temperature, but if considerable superheat occurs, then it is difficult to quantify the thermal storage terms and the changes in air side temperature difference. This underscores the importance of uniform refrigerant distribution in MCHX headers and precise superheat control to produce good experimental data.

**C.6 Conclusion**

Evaporators 1 and 3 can not be placed with their headers horizontal and tubes vertical, due to space limitations. Also when tilted at 60 and 65 degree angles respectively, the data are not comparable. Finally if placed with their headers vertical oriented, and tubes horizontal, refrigerant distribution is compromised. Unfortunately, evaporator 2 has a lower capacity than what initially desired (1.2 vs. 2 tons) but it fits into the wind tunnel easily. Since its header could be placed horizontally refrigerant distribution would be better, therefore it was decided to use evaporator 2.

**C.7 List of references**


Appendix D: Thermocouple placement in the new evaporator

In order to estimate the average metal temperature on the MC evaporator, 16 surface thermocouples were placed on the inlet and outlet headers as well as in the tube bends. Figure D.1 and D.2 illustrate the shape of the MCHX and the corresponding location of the thermocouples. Once installed the thermocouples were re-checked and calibrated. At the end only 12 thermocouples provided accurate readings and were used to calculate an area weighted average temperature for the evaporator.

That average temperature was later used to estimate one of the degradation terms in rapid cycling operation: nonlinearity. Most thermocouples were placed in the second bend in order to capture the effects of maldistribution at the high quality regions which were expected to be more pronounced near the outlet. The refrigerant and air are flowing in counterflow configuration; therefore it is more likely that localized superheat would occur since the warmest air meets the refrigerant with the highest quality.

In real applications the most appropriate configuration would be parallel flow, however, in order to be able to obtain infra red images, the evaporator was placed in counter flow configuration.

Figure D.1  Sketch of MC evaporator
Figure D.2 Developed view, location of thermocouples.

After the infrared pictures were taken the evaporator was turned around and the test matrix was evaluated in parallel flow configuration.
Appendix E: Selection of software and hardware for superheat control

E.1 Introduction
Since the currently installed Electronic expansion valve (EEV) along with its controller board appear to be fast enough to be able to regulate superheat if an adequate controller is programmed, the next task is the selection of the software in which the controller is to be developed. The criteria used in the selection of the control software/hardware were: 1) flexibility and ease of programming; and 2) ease of installation and cost.

Two options were considered to implement the controllers: 1) Agilent-Vee 2) Matlab-Wincon-RTX. In the following paragraphs a brief description of each will be presented and its main strengths and weaknesses evaluated in terms of the selection criteria. It was not possible to find an output PCI board that was supported simultaneously by Agilent-Vee and Wincon-RTX-Matlab. Therefore choosing a particular software package predetermined the board to be used or vice-versa.

E.2 Flexibility and ease of programming
Technical characteristics of both systems were studied and it became evident that the same type of controllers could have been implemented in any of the two solutions. However the Matlab-Wincon-RTX solution makes it easier to implement more complex controllers and also provides real time control which Agilent-Vee does not. However several tests were conducted and it became evident that real time control is probably not necessary since refrigeration systems are somehow slow responding. Since the scope of this project is not the development of controllers but the design of systems for compressor rapid cycling, our priority is the development of the simplest workable controller.

E.3 Ease of installation and cost
The main advantage of using Agilent-Vee is that it is already installed in the laboratory, as the data acquisition software with the required A/D converters and data acquisition boards in place and only the output board needed to be added. Figure E.1 illustrates the setup of the system with the controller implemented using Agilent-Vee. The requirements for additional hardware and software were:

Output board HP 1328A $1120
Command modules RAM (512K) $  486
Total Estimated cost $1606

On the other hand, the main advantage of implementing the control system using Matlab-Wincon-RTX is that it has already being used in other applications in the ACRC. However, extensive hardware and additional wiring are required to implement the controllers using Matlab-Wincon-RTX, therefore setup time would have been longer. The required hardware was:

PCI Multi Q  D/A-A/D converter board $1550
OM5-LTC Omega 5B module $  152
OM5-IVI Omega 5B module $  184
Quanser Wincon-RTX $  800
Also the control system running Matlab-Wincon-RTX needed to run in a separate PC, requiring additional wiring and hardware to provide the controller the state variables from the system. If an additional PC is be used, there is an advantage since the data logging process becomes independent of the control process, distributing the computing load. The corresponding setup is illustrated in Figure E.2.

![Control/ data logging system using Agilent Vee](image1)

E.4 Conclusion

Because of its equivalent flexibility for the required tasks, lower initial cost and ease of installation it was decided to implement the controller running Agilent-Vee using Agilent’s E1328A analog output board to link the PC to the EEV stepper motor controller board.
Appendix F: Vapor leakage through the compressor during off-cycle

F.1 Introduction

Ilic et al (2001) suggested that one of the factors that caused an important degradation of the COP in short cycling systems was off-cycle vapor leakage through the compressor. Vapor leakage is known to occur in some scroll and screw compressors. Vapor leaking from the condenser to the evaporator degrades COP for two reasons. First, the work done by the compressor is lost; and second, the operating conditions of the heat exchangers are changed to the point that the cycle could be reversed and evaporation may take place in the condenser as well as condensation in the evaporator.

In order to provide some basic understanding of the phenomenon and also to get a quantitative idea of its relative importance a computer simulation was performed. It is assumed in this model that there is a solenoid valve that prevents vapor flow through the expansion device during off-cycles, thus the only way for vapor to flow from the condenser to the evaporator during off-cycle is through the compressor. The amount of refrigerant leakage is estimated to be on the order of 3% of the steady state mass flow rate. The following pages provide a brief explanation of the assumptions made in the model, as well as the results it yielded.

F.2 Model

Initially a 2 ton residential split system was modeled. Although its scroll compressor has an extra valve that prevents leakage from the condenser to the evaporator through the compressor, its operating conditions were measured in detail and therefore it was used as a model for the computer simulation. The inputs required by the model are:

- Definition of average thermodynamic states on both heat exchangers at the end of the on cycle (assumed equal to steady state conditions).
- Thermal mass of the heat exchangers.
- Internal volume of the exchangers.
- Air and refrigerant side heat transfer coefficients for the two heat exchangers.
- Refrigerant and air side heat transfer areas.
- Steady state mass flow rate and leakage rate (as percentage of steady state).
- Indoor and outdoor temperatures.

<table>
<thead>
<tr>
<th>Table F.1 Initial conditions</th>
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<tr>
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<tr>
<td>Quality</td>
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<tr>
<td>Pressure [KPa]</td>
</tr>
<tr>
<td>Refrigerant temperature[°C]</td>
</tr>
<tr>
<td>Metal temperature[°C]</td>
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### Table F.2 Geometrical specifications of the modeled system

<table>
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<tr>
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<th>Condenser</th>
<th>Evaporator</th>
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<td>Thermal Capacitance [KJ/K]</td>
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<td>Tube Material</td>
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<tr>
<td>Tube ID [mm]</td>
<td>7.9</td>
<td>7.9</td>
</tr>
<tr>
<td>External heat transfer area [m2]</td>
<td>19.5</td>
<td>21</td>
</tr>
<tr>
<td>Internal heat transfer area [m2]</td>
<td>1.21</td>
<td>1.34</td>
</tr>
<tr>
<td>Internal Volume [m3]</td>
<td>2.4 x 10-3</td>
<td>2.46 x 10-3</td>
</tr>
</tbody>
</table>

Tables F.1 and F.2 show the specifications and operating conditions for the system simulated1. With those values as inputs to the model, the simulations are run (the model was written using the Engineering Equation Solver). All modeling is based on Equations F.1 to F.8. The leak rate is given by Equation A1, where K is the initial rate at which vapor leaks through the compressor, expressed as fraction of the steady state mass flow rate. The solution to equation F.1 shows how the leak rate varies as pressures equalize.

\[
\dot{m}_l = m_{ss} \cdot K \cdot \sqrt{\frac{P_{C,1} - P_{E,1}}{P_{C,0} - P_{E,0}}} 
\]

(F.1)

\[
\frac{dT_{met}}{dt} = \frac{\dot{Q}_{ref,met} - \dot{Q}_{met,a}}{Cp_{met} \cdot met} 
\]

(F.2)

\[
\frac{d(m \cdot h_{ref})}{dt} = -m \dot{h}_v - \dot{Q}_{ref,met} 
\]

(F.3)

\[
\dot{Q}_{ref,met} = h_{ref,met} \cdot A_{ref,met} (T_{met} - T_{ref}) 
\]

(F.4)

\[
\dot{Q}_{met,a} = h_{met,a} \cdot A_{met,a} (T_{met} - T_a) 
\]

(F.5)

Several assumptions were made in order to simplify the model; the most important are presented below.

- In the ideal system there is no refrigerant flow through the exchangers during off cycles.
- At the beginning of the off-cycle valves are shut and the thermodynamic conditions for the refrigerant and metal throughout the exchangers are constant. Initial conditions are set according to steady state conditions at compressor shutdown.

---

1 Two simulations with different qualities and internal volumes in the condenser were used in order to simulate the system with and without the inclusion of a high side receiver. The data presented in all Tables corresponds to the case without receiver.
• Air side heat transfer coefficients during off-cycle are equal to on cycle coefficients at steady state conditions for the same capacity since the fan and blower are left running during off cycle.
• There is no refrigerant flow; Evaporation in the tubes is modeled as pool boiling and condensation as laminar film condensation on a vertical plate. See Equation F.6.
• The heat transfer areas are constant during the cycle, changes in available areas for heat transfer due to boiling or condensation are not considered.
• The inside and outside metal temperatures for both exchangers are assumed to be equal, $Bi << 0.1$ so we assume a lumped capacitance model.
• Energy can only be transferred between the air and the evaporator through the metal, or by incoming refrigerant if there is leakage. Energy can only be transferred to or from the condenser through the metal or taken away by leaking refrigerant.
• There is no mass flux from the compressor/high side receiver through the expansion valve (closed valve operation) into the low side volume. That operating mode is achieved in practice by placing a solenoid valve upstream of the expansion device.

Equation F.6, Nusselt’s correlation for laminar film condensation on a vertical plate is taken from Incropera (1996) and the pool boiling heat transfer coefficient is calculated from the Rohsenow (1952) correlation,

$$h_l = 0.943 \left[ \frac{g \cdot \rho \cdot (\rho_f - \rho_v) k_T^3 h_{fg}'}{\mu \cdot (T_{sat} - T_v) \cdot L} \right]^{1/4}$$

$$q''_v = \mu \cdot h_{fg} \left[ \frac{g \cdot (\rho_f - \rho_v)}{\sigma} \right]^{1/2} \left[ \frac{Cp \cdot \Delta T_e}{C_s \cdot h_{fg} \cdot Pr^a} \right]^3$$

where

$$q'' = C \Delta T_e^a h^b$$

and

$h_{fg}' = h_{fg}(1+0.68Ja)$

$C_{sf}$ Surface-fluid constant (0.068)

$n$ Correlation constant (1.0)

$a$ Correlation constant (1.2)

$b$ Correlation constant (1/3)

Now all inputs to the model are known.
F.3 Results

The thermal capacitance of a solid affects its response to changes in its environment and/or operating conditions. From the short-cycling point of view, it is better to have slow-responding exchangers so the temperature difference between the exchanger and the air is maintained nearly constant throughout the off cycle. If the thermal capacitance is very small, then the exchanger will reach ambient temperature in a short period of time and the amount of heat transfer during the off cycle will diminish. On the other hand, if the thermal capacitance is very large the exchanger will maintain a considerable temperature difference from its surroundings being able to exchange more heat. For more details, see Ilic et al (2001).

With that brief introduction to the dynamics associated with short cycling systems it is now time to relate it with leakage. Leakage through the compressor degrades COP for two reasons: 1) the work done by the compressor is lost since the vapor expands once it leaks to the low pressure side and has to be compressed again and; 2) the operating conditions of the heat exchangers are changed when leakage is present, and affects their performance.

![R22 T-v diagram](image)

Figure F.1 R22 T-v diagram

The first item does not require further explanation. The following pages are devoted to the explanation of the second issue. All the results presented in the following Figures were obtained by running the simulations with initial conditions as specified in Table F.1 and system parameters as in Table F.2.

It is important to note that the results shown here do not represent the actual operating off-cycle of the system since the effect of leakage will change the initial conditions used to run the simulation from the values used.
F.3.1 Evaporator

To get a better picture of the phenomenon it is a good idea to look at the T-v diagram presented in Figure A1 where the diagonals are isoquality lines. This figure will also be very helpful when dealing with the condenser.

The ideal (no leakage) process during off cycle in the evaporator will consist of an increase in pressure while the specific volume is held constant.

![Graph of evaporator capacity with leakage rates](image)

**Figure F.2 Evaporator capacity. 0% and 5% leakage rates.**

![Graph of evaporator capacity on refrigerant side](image)

**Figure F.3 Evaporator capacity. Refrigerant side.**
When leakage is present the addition of refrigerant mass into the evaporator will move the state point towards the left where qualities are lower; in both cases the pressure is increasing. Depending on the leakage rate and the thermal capacitance it is possible for the evaporator to become a condenser in an extreme case, as its saturation temperature rises above the air temperature when the indoor and outdoor pressures equalize rapidly.

Figure F.2 shows the changes in capacity in the evaporator. It can be shown that in this case with a 5% leak rate the capacity of the evaporator during the off cycle decreases about 7%. The cause of this degradation, a warmer evaporator surface temperature, is apparent from Figure F.3 where the heat transfer from metal to refrigerant not only decreases but becomes negative. The refrigerant is heating the metal from the inside due to the high enthalpy vapor that is receiving from the condenser via the compressor.

![Graph showing temperature changes](image)

**Figure F.4** Metal and refrigerant temperatures in the evaporator. No leakage.

Comparison of Figures F.4 and F.5 would also help to understand the phenomenon. As high enthalpy vapor enters the evaporator the pressure and saturation temperature of the refrigerant inside will increase and start to heat the metal from the inside which will cause its temperature to increase more rapidly than in the no leakage case where it is heated from the air side but cooled from the refrigerant side. As a consequence of being heated from both sides the metal temperature will increase faster than in the no leakage case. As a result the difference between the metal and air temperatures will be smaller, thus reducing the capacity of the system. Recall Figure F.2.
F.3.2 Condenser

If there is no leakage in the condenser as pressure decreases, it is clear that the quality will decrease as well; see Figure F.1. If there is leakage the amount of mass inside decreases. Therefore the specific volume increases, moving the system operating point towards the right hand side of Figure F.1. In those conditions, quality may increase or decrease but the condenser will not be as efficient as in the no leakage case. In the particular conditions of this simulation it turned out that the quality did not change during the off-cycle, remaining almost constant at around 0.2. Depending on the leakage rate and the thermal capacitance, evaporation may take place in the condenser. As vapor leaves the condenser pressure decreases and so does temperature. After a short time (20 s) the refrigerant will have reached the metal temperature and after a while it will be even colder (compare Figures F.6 and F.7). The metal is now being cooled from both sides, this causes its temperature to decrease faster and therefore the capacity of the system will be degraded. Figures F.8 and F.9 illustrate the effect.

It can be said that leakage affects the exchangers and alters their operating conditions by changing the net heat flux in or out of the metal during the off-cycle. In the case of leakage the refrigerant will warm the metal in the evaporator and cool it in the condenser during the off-cycles. It affects performance because it reduces the time constant of the system.

Figure F5 Metal and refrigerant temperatures in the evaporator. 5% leakage.
Figure F.6 Metal and refrigerant temperatures in the condenser. No leakage.

Figure F.7 Metal and refrigerant temperatures in the condenser. 5% Leakage
Figure F.8 Condenser capacity.

Figure F.9 Condenser capacity. Refrigerant side.
F.3.3 Effect of a receiver

Some systems include a receiver in order to store extra charge and maximize condenser effectiveness. A model of such a system was also developed by adding a volume of 3 liters to the system as receiver internal volume, then the amount of refrigerant inside the receiver was modified as shown on Table F.3. The initial quality inside the receiver was kept at 0.21 and not altered.

Table F.3 Effect of receiver liquid level

<table>
<thead>
<tr>
<th>Mass of liquid [Kg]</th>
<th>High side average quality</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.34</td>
</tr>
<tr>
<td>.75</td>
<td>0.16</td>
</tr>
<tr>
<td>1.5</td>
<td>0.11</td>
</tr>
<tr>
<td>2.25</td>
<td>0.075</td>
</tr>
<tr>
<td>3</td>
<td>0.04</td>
</tr>
</tbody>
</table>

The addition of refrigerant mass effectively increases the thermal capacitance of the system if all the refrigerant in the receiver, the condenser and the refrigerant inside it, are considered as a single system. As could be expected from an increase in thermal capacitance the response of the refrigerant to the cooling of the metal is slower and therefore the efficiency improves in both cases (leakage and non-leakage) even if the receiver is only filled with vapor. It is important to note that the 3 liter receiver volume is comparable to the condenser volume.

The results are presented in Figures F.10 to F.12 where a system in which the receiver is completely filled with liquid is compared to a system with no receiver. The receiver full of liquid provides the highest benefits; as the receiver is emptied the benefits decrease, even a vapor filled receiver provides some improvement over the case where no receiver is present. In the best case the COP will be increased about 2 % above the case where there is no receiver present. The gain on COP corresponds to the area under the curve of the metal temperatures as time progresses during the off-cycles.

Figure F.10 Refrigerant temperature comparison.
**F.4 Conclusions**

Leakage through the compressor degrades COP for two reasons: 1) the work done by the compressor is lost because any compressed vapor that leaks back to the low pressure side has to be compressed again and; 2) the operating conditions of the heat exchangers are changed when leakage is present, affecting their performance.

Refrigerant leakage degrades the performance of the heat exchangers in rapid cycling because the effective time constant of the heat exchangers is reduced. For example, vapor leaking into the evaporator causes the saturation and metal temperatures to raise faster than if the evaporator was isolated. In fact, a leak can eventually cause vapor
to begin condensing inside of a thermally massive evaporator. Similarly, leakage can turn a compressor into an evaporator as the saturation temperature falls below the metal temperature.

The addition of a high side receiver at the condenser outlet improves the COP of the system, since the addition of refrigerant mass increases the effective time constant of the condenser. Its effect is to retard the decrease in saturation temperature as the vapor leaks out of the condenser. The best case occurs when the receiver is filled with liquid. In that case an improvement of 2% on COP is achieved (in our system). As the amount of mass stored in the receiver decreases so do the benefits. In the extreme the addition of a vapor filled receiver should still provide marginal COP gains over the system with no receiver.

F.5 List of references


Appendix G: Performance degradation due to condenser thermal capacitance

G.1 Introduction

In the following pages the results of a simple modeling of the transient behavior of microchannel condensers during off-cycle will be explained and analyzed. All parameters and quantities used in the model were deduced or taken from well known systems and operating conditions but, by no means intend to represent any specific system. In previous experiments with conventional copper tube heat exchangers (Ilic et al. 2001) it was found that: 1) even during the offcycle, refrigerant side heat transfer resistance is small compared to the air side resistance and; 2) the thermal capacitance of the heat exchangers is a determining factor of efficiency in pulse-width modulated systems. Therefore it is expected that: 1) microchannel HX’s will not be able to increase heat rejection during off-cycle since the limiting resistance is on the air side and; 2) differences in thermal mass will play an important role in determining the performance of microchannel heat exchangers.

G.2 Initial internal heat transfer coefficients and metal temperatures

Based on a linear regression of the experimentally determined steady state heat transfer coefficients from Dobson et al (1993) at G=200 for R134a, it is possible to estimate the initial metal temperatures (at the end of the on cycle) as a function of quality by means of a thermal network approach. Since the metal’s Biot number in the radial direction is small (Bi~0.001 for the length scale in the radial direction) its temperature is assumed to be constant in that direction. However, considerable temperature differences are expected along the tube due to the variation of the heat transfer coefficient with quality. Figure 1 presents initial metal temperatures along the microchannel tube for Tref =45 °C, and Tair= 35 °C.

![Figure G.1 Initial tube metal temperatures.](image-url)
In order to determine off-cycle condensation coefficients, homogeneous void fraction and uniform initial refrigerant distribution (an annular film) were assumed to estimate the refrigerant film thickness as a function of quality. If the condensation coefficients are assumed to be inversely proportional to film thickness (as in Nusselt’s correlation for condensation on a flat plate) it would seem possible to solve for the heat transfer coefficient as a function of quality. Unfortunately the calculated values of the heat transfer coefficient did not agree with validation data particularly in the regions of high and low qualities, because according to Nusselt’s assumption the heat transfer coefficient is just a function of the thickness and the conduction of the vapor; therefore when quality tends to 0 so does the thickness and then the heat transfer coefficient increases as k/X. However the calculated thickness of the refrigerant layer was always less than 10% of the tube diameter which would discard the idea of bridging even in the event that initial refrigerant inventory had been underestimated by a factor of 2. Therefore, even if bridging or drainage do occur during off-cycle and all the liquid refrigerant accumulates (pools) in the bottom, it would result in a thinner refrigerant film layer in the rest of the tubes, increasing the already high condensation coefficients.

G.3 Lumped capacitance model

Since the internal volume of the tube is known, and an average initial quality was assumed it is possible to solve for the refrigerant mass inside the tubes. With known masses, areas, condensation (3 and 2 kW/m²K) and boiling coefficients (3 kW/m²K) for refrigerant 75 W/m²K for air, it is possible to formulate a lumped capacitance model for a microchannel tube initially containing refrigerant at 0.5 average quality. However, since there are significant temperature variations across the length of the tube even in the 2 phase region (Figure G.3) the heat exchanger was modeled as two lumped systems of equal mass and area, which would have condensing coefficients and initial temperatures corresponding to points at 0.25 and 0.75 quality as can be seen in Figure G.2 where a sketch of the system model is presented. For these calculations, the superheated and subcooled regions are ignored.

Figure G.2 Model layout and off-cycle starting condition
Figure G.3 Metal and refrigerant temperatures in MC condenser during off-cycle.

Figure G.4 Heat fluxes in MC condenser during off-cycle

Figure G.3 presents the refrigerant and metal temperatures behavior during the initial 6.2 s. of the off-cycle for the initial conditions as presented in Figure G.2. Figure G.4 presents the heat fluxes from metal to air (ma) and from refrigerant to metal (rm) in the high and low quality regions, as well as conduction through the metal which is negligible. Note that in the zone where the metal is hot (high X) the refrigerant gets colder than the metal as the pressure equalizes in a short fraction of a second after shut-off. This accelerates the metal’s cooling process in the
high temperature region by boiling while condensation is still occurring in the low quality region until a uniform metal temperature is reached. Finally Figure G.5 presents the net energy fluxes between the metal, refrigerant and air during the first 6.2 s as well as the state after 6.2 s. Once this state is achieved uniform metal temperature can be assumed to be constant throughout the condenser. The system can be modeled as a single lump of metal cooling to room temperature where the only relevant parameters are thermal capacitance and air side heat transfer.

![Diagram of energy fluxes](image-url)

Figure G.5 Energy fluxes during initial 6.2 s.

### G.4 Time constant

Recall that the time constant of a heat exchanger determines the rate at which it cools or warms during the off-cycles. The time constant is one way to compare the rate of cooling for different heat exchangers. But there are others. In this case the ratio thermal capacitance/capacity was used to compare a microchannel and a conventional copper tube/aluminum fin heat exchanger.

First assume that the product of the finned area times the air side heat transfer coefficient in both cases is the same, because most of the resistance is on the air side. Furthermore, assume that the two heat exchangers will be operating at the same temperature difference on the air side. In this special case of equal capacities the thermal capacitance determines the difference.

The conventional 2 ton finned tube heat exchanger core had approximately 1400 g/kW of copper tubing and 1070 g/kW of aluminum fins. Microchannel heat exchangers based on the dimensions given by Kirkwood and Bullard for their residential a/c prototype had about 1550 g/kW, and were all aluminum. Since the specific heats of copper and aluminum are 385 kJ/kg and 890 kJ/kg respectively, these microchannel heat exchangers have a ratio thermal_capacitance/capacity about 7% smaller than conventional copper tube heat exchangers. The difference is not considerable and many assumptions have been made in order to reach this number.
Finally, and despite the considerable differences in refrigerant charge per unit capacity for conventional copper tube and microchannel heat exchangers it is important to mention that the effect of refrigerant capacitance is negligible when compared to metal capacitance. Note that for the copper tube heat exchanger, where the ratio refrigerant charge/capacity is higher than for microchannels the ratio of the refrigerant thermal capacitance/metal thermal capacitance is very small.

Assume an initial condition $X_{ave} = 0.6$, $T_1 = 45\, ^\circ C$ and a final condition $X_2 = 0$, $T_2 = 35\, ^\circ C$ on a 1 kW round copper tube heat exchanger, then

$$m_{metal} \cdot C_{metal} \cdot (45 - 35) = 1.52 \, [kJ]$$  \hspace{1cm} (G.1)

$$m_{ref} \cdot (u_2 - u_1) = 0.05 \, [kJ]$$  \hspace{1cm} (G.2)

The refrigerant thermal capacitance is less than 1/20th of the metal’s. Therefore we can consider microchannel HX’s to have approximately the same thermal capacitance as copper tube HX’s.

**G.5 Conclusions**

Refrigerant film thicknesses calculated for microchannels are a very small when compared to tube diameter, which suggests that bridging is not likely to occur and high condensation heat transfer coefficients are to be expected even in the absence of flow.

However if the refrigerant drains, bridges and/or pools during off-cycle, the remaining film on the tube walls would become even thinner because of the limited amount of liquid available and the condensation coefficients become larger, according to Nusselt’s assumptions for vapor condensing on a flat vertical plate.

The refrigerant-side resistances are expected to be very small when compared to the air side resistances for most qualities ($x>0.2$). Therefore the air side resistance can be regarded as the limiting factor in the heat rejection process during off-cycle.

It has been shown that the metal temperatures are equalized during off-cycle mainly by the different rejection rates to the surrounding air. Other processes such as boiling in the high temperature (quality) zone, condensation in the low temperature region and condensation play a minor role, accounting for a negligible fraction ($<0.01$).

Finally the thermal capacitances of conventional and microchannel heat exchangers were compared and the ratio Thermal capacitance/Capacity was found to be about 7% higher for conventional copper tube heat exchangers.

Therefore no improvement is expected from the use of microchannel condensers in pulse-width modulated systems in terms of heat transfer phenomena since there is no benefit from their enhanced performance when compared to conventional copper tubes.

The refrigerant contribution to the thermal capacitance of the systems for both conventional copper tube and microchannel heat exchangers is negligible when compared to the metal contribution and can be ignored in most cases.

For microchannel evaporators similar results are expected. Since the time constants of the heat exchangers do not differ considerably in a system, this analysis of the condenser should extend to the evaporator as long as the overall refrigerant side resistance does not differ considerably.
Therefore replacing conventional condensers with MCHX’s in a pulsed width modulated system is expected to have little effect on performance, because there is no gain from the inside heat transfer resistance benefits associated with MCHX’s and their time constants are nearly equal to conventional HX’s.

G.6 References
Appendix H: First law analysis of FGB in rapid cycling off-cycle

In order to understand the off-cycle behavior of the low side, a first law analysis was conducted. A summary of the approach taken and the main results are presented below. Figure H.1 is a diagram of the low side. The main purpose is to find the magnitudes and directions of the heat fluxes between the evaporator, refrigerant and FGB tank as illustrated in Figure H.2.

The following assumptions were made: 1) Quality increases linearly with evaporator length from 0 to .90. Average quality is therefore 0.45. 2) Inlet quality in the evaporator is 0, FGB is being used. 3) Outlet quality is 0.9. SLHX is being used. 4) Average quality in the suction line is 0.9. 5) The liquid level in the tank is about 20% of its height, therefore the homogeneous void fraction is 0.8 and the quality in the tank is 0.07. Based on the previous assumptions, and on the geometrical parameters of the system it is possible to estimate the amount of mass stored in each component and its state at the start of off-cycle. In order to simplify calculations, a uniform average refrigerant starting condition is calculated from data and from then on the refrigerant is assumed to be at uniform state.
Based on the results of two experimental runs at 70 second cycle period (~60% capacity 42 sec on, 28 sec off) conditions of the refrigerant, metal and FGB tank are used at 3 times during the off-cycle to quantify heat fluxes. Figures H.3 and H.4 illustrate the most relevant temperatures during a few cycles. The difference between the two is that for the runs in Figure H.3 a solenoid valve downstream of the EEV was closed (closed valve CV) during off cycle and the low side is completely isolated from the high side during the off-cycle, whereas in Figure H.4 the solenoid is left open (Open valve OV), and there is continuous mass flow into the low side. In both cases the line numbered 1 indicates the start of the off-cycle to be analyzed. Line No. 2 is located at the point at which the refrigerant saturation and evaporator outlet temperature reach the metal temperature at the start of the off-cycle, finally line No. 3 is located at the end of the off-cycle. Line 2 is located at an intermediate location where a transition in the operating condition starts.

In order to familiarize the reader with the system, a table and several charts with the most relevant parameters are presented below. Of particular importance are the volume ratios between evaporator and FGB tank,
as well as their areas. Note that most of the refrigerant is stored in the tank but most of the air side area is in the evaporator.

It is also important to realize that there is no direct path between the evaporator and FGB tank for heat transfer, therefore, all heat transfer to and from the tank has to be done by means of the refrigerant. This is an obvious but important observation.

Table H.1 Relevant geometrical and system parameters

<table>
<thead>
<tr>
<th>Initial condition</th>
<th>Evaporator</th>
<th>Tank</th>
<th>Suc. line</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volume [cc]</td>
<td>86</td>
<td>3000</td>
<td>250</td>
</tr>
<tr>
<td>Internal Area [m^2]</td>
<td>0.53</td>
<td>0.03</td>
<td>0.04</td>
</tr>
<tr>
<td>Cross sect area [m^2]</td>
<td>8.10E-05</td>
<td>0.012</td>
<td>7.20E-05</td>
</tr>
<tr>
<td>Ref Mass [gr]</td>
<td>9</td>
<td>1200</td>
<td>9</td>
</tr>
<tr>
<td>Liq mass [gr]</td>
<td>5.5</td>
<td>1080</td>
<td>0</td>
</tr>
</tbody>
</table>

Figure H.5 Refrigerant side areas

Figure H.6 Refrigerant mass in the low side

RC operation at ~60% capacity in the 0.75 ton system provides ~1.05 kW of power into the system during off cycle, Figure H.8 illustrates how the energy is distributed/stored in the system, between times 1 and 3, Figure H.9 serves the same purpose between times 2 and 3 and finally figure H.10 corresponds to the lapse between 1 and 2 as marked in Figure H.3. The initial average quality in the system is about 0.07, and since the low side is isolated in the suction line by the compressor and in the liquid line by the solenoid valve, the process consists of a temperature and pressure increase at a constant volume, with the corresponding increase in quality, as can be seen in the T-v diagram for R22. Table H.2 presents representative temperatures of the FGB tank, evaporator and refrigerant in each of the three points.
Table H.2 Evaporator, tank and refrigerant temperatures, CV run.

<table>
<thead>
<tr>
<th>Time</th>
<th>1</th>
<th>2</th>
<th>3</th>
</tr>
</thead>
<tbody>
<tr>
<td>T_metal_evap</td>
<td>14.8</td>
<td>17.1</td>
<td>21.6</td>
</tr>
<tr>
<td>T_tank</td>
<td>17.2</td>
<td>17</td>
<td>17.5</td>
</tr>
<tr>
<td>T_ref</td>
<td>12.35</td>
<td>16.15</td>
<td>16.48</td>
</tr>
</tbody>
</table>

In figure H.8 it is important to note that about 70% of the energy that enters the system is being stored in the evaporator metal, as temperature, also the refrigerant serves as a storage taking about 27% and the tank takes the remaining 3%.
A quick inspection of Figure H.9 reveals that there is no significant heat transfer between any of the system components during this time interval, the evaporator metal and the tank warm up, while the refrigerant remains at an almost constant state. A more detailed analysis reveals that the energy input into the system can not be found in any of the energy changes of its components, there is an energy excess in the system, the total heat input is 19 kJ and the energy stored in the three components only adds up to 15.1 kJ. The difference is 3.4 kJ and will be attributed to the non-uniform state of the refrigerant. That number will be kept as an excess energy, the analysis between times 1 and 2 will be performed, where it is expected to find an energy deficit between those two states.

As expected there is an energy deficit between states 1 and 2, the deficit is only 3.2 kJ when a 3.5 kJ deficit was expected, but small variations in the temperature measurements could be the cause of the differences but are in general small. As an example take the .3 kJ missing between states 1 and 3. That is about 1% of the 30kJ exchanged, which can be considered as a very good agreement. Also not all states for all components match exactly, but the cause is considered to be the same.

The main point is that there is no equilibrium in the state of the system around point 2. It is clear that the refrigerant stored in the tank will not reach the saturation temperature of the refrigerant in the evaporator immediately as pressure rises due to boiling. Therefore there is an energy deficit in point 2 since it was assumed that all refrigerant would be at the temperature that corresponds to the saturation pressure imposed by boiling vapor in the evaporator at the start of the off-cycle. As refrigerant boils in the evaporator pressure is increased and colder
refrigerant vapor in the tank will condense on the liquid surface as it warms it up to the new state. The final semi-steady condition is reached when the rate of boiling in the evaporator decreases due to dry-out. There is still some boiling present because the lowest part of the evaporator is kept flooded.
Appendix I: Evaporator superheat response to expansion valve opening

The evaporator superheat response time to expansion device opening represents the time elapsed from the expansion device opening until the change in the measured value of the superheat. Slow superheat response to the valve opening may cause problems during automatic expansion device operation, due to the long time lag between valve reaction and measured superheat response. The response time has been experimentally determined and presented below.

The a/c system was run at steady state 80/95 F dry condition with about 6 C evaporator exit superheat. The expansion device was set at about 800 steps opening, at the analyzed condition. The valve was suddenly opened an additional 100 steps, and the system response was observed. The refrigerant mass flow rate and evaporator exit superheat response are shown in Figure I.1.

![Figure I.1 Refrigerant mass flow rate (Mr) and evaporator superheat (SH)](image)

The refrigerant mass flow rate, measured in the liquid line, immediately upstream of the expansion valve increased instantaneously, as expected. This increase in refrigerant mass flow rate caused decrease in evaporator superheat which was not instantaneous but with time lag of about 10 seconds.

The superheat response time to the expansion device opening was experimentally determined to be about 10 seconds. A lower the superheat response time is needed to prevent degradation of evaporator performance during off-cycle. The superheat response time of 10 seconds may be regarded as slow, since it is almost equal to a half of the usual compressor short-cycling cycle period of about 20 seconds.

Figure 2 shows that the change in superheat is due to a decrease in the refrigerant temperature (measured by immersion t/c) at the exit of the evaporator while the saturation pressure remains fairly constant. One possibility is that this time lag exists, due to the time needed for the refrigerant to travel from the expansion device (evaporator inlet) at t=68 when the valve is opened to the evaporator exit about 10 seconds later.
Simple calculations were performed to estimate the time required for the refrigerant to flow through the entire 13 m length of the evaporator tubes. Assuming a linear distribution of quality along the evaporator and homogeneous void fraction, the speed for each differential length and required travel time is approximated. The integration of travel time yields a total travel time of about 10 seconds, which is consistent with experimental measurements.

By opening the valve, refrigerant mass flow rate at the inlet of evaporator immediately increased from 43 to 47 gr/s. Experimental measurements and calculations have shown that it took approximately 10 seconds before the wave front of the new mass flow rate (47 gr/s) reached the evaporator exit. At that moment the dryout point started moving towards the exit of the evaporator, since the same amount of heat was transferred from the air to the refrigerant having a higher mass flow rate. As the dryout point moved steadily downstream refrigerant exit temperature decreased by 6 °C, starting the cooling of metal mass in the previously superheated region (~10% of evaporator metal mass). Due to difference in resistances, surface temperature in 2 phase zone was initially about 2°C warmer than refrigerant. However it is about 5°C warmer than refrigerant in superheated zone. Therefore metal in
superheated zone must cool, on average more, approximately 4.5 °C ((4+5)/2 °C. See Figure 3). The cooling of metal will occur first near x=1, and then proceed downstream as walls are wetted by increasing refrigerant inventory. This process may explain why superheat does not decrease immediately from 6 to 0 °C but will take time as the metal is cooled down.
Appendix J: Control algorithms

In this appendix five different control algorithms are discussed. These algorithms include: fixed valve opening, thermostatic expansion valve emulation, proportional control, feedforward control, and iterative learning control. The development and control theory behind the controllers discussed in Chapter 3 is presented in the following paragraphs.

J.1 Fixed valve opening
This is not truly a control algorithm, but really the absence of a control algorithm. This is implemented as a baseline for comparison of the other control strategies.

J.2 Thermostatic expansion valve emulation
To facilitate the implementation of more advanced control algorithms, the experimental system used an electronic expansion valve (EEV) to control mass flow entering the evaporator. To compare the performance of alternative control algorithms against the more traditional control mechanisms of a thermostatic expansion valve (TXV), it was requisite to emulate the dynamic behavior of a TXV using an EEV. This is the most likely non-EEV based control solution. Dynamic models of TXV’s have been developed previously in Broerson and James. Broerson modeled the TXV with higher order ODE’s while James modeled the TXV with a linear first order sensor dynamic combined with various static nonlinearities. For the purposes of this study, the simplified model proposed in Dane is used, where only the linear first order sensor dynamic from James is included. The valve opening is related to superheat using the relationship in Equation J.1, where the measured value of superheat is modeled as given in Equation J.2 or in transfer function form as in Equation J.3. This model assumes a first order response, where the time constant, \( \tau \), is selected as in James to be 5, and where \( K_{TXV} \) is an adjustable parameter. This model obviously does not include the many nonlinear phenomena that would be present in a actual TXV, but since many of these would result in decrease in stability and performance, this TXV emulation can be considered the best possible case for this control mechanism.

\[
A_v = K_{TXV} \left( T_{desired}^{SH} - T_{measured}^{SH} \right) \quad (J.1)
\]

\[
T_{measured}^{SH}(t) = \left( \frac{1}{\tau} e^{-t/\tau} \right) T_{err}(t) - T_{ero}(t) \quad (J.2)
\]

\[
T_{measured}^{SH} = \left( \frac{1}{1 + \tau s} \right) T_{err} - T_{ero} \quad (J.3)
\]

J.3 Proportional control
Proportional-Integral-Derivative (PID) control is a widely known and implemented control algorithm. The continuous time version of this control algorithm is shown in Equation J.4 where the error, \( e(t) \), is given by the desired value of superheat minus the measured value of superheat (Equation J.5). This algorithm varies the valve opening proportional to the error, the integral of the error, and the derivative of the error. The gains for each of these terms is selected or “tuned” by the user. This can be done experimentally, or using a model of the system.
response. The Laplace transform of this control algorithm is shown in Equation J.6 and is equivalent to placing a pole at the origin, and two arbitrary zeros. Assuming a trapezoidal integration rule, the discrete time version is given in Equation J.7.

\[ A_v = K_p e(t) + K_i \int e(t) + K_d \dot{e}(t) \quad (J.4) \]

\[ e(t) = T_{\text{desired}}^{SH} - T_{\text{measured}}^{SH} \quad (J.5) \]

\[ G_s(s) = \frac{K_p + \frac{K_i}{s} + K_ds}{s} = \frac{K_{PID}(s + a_1)(s + a_2)}{s} \quad (J.6) \]

\[ G_s(z) = K_p + K_i \frac{T_s}{2} \left[ \frac{z + 1}{z - 1} \right] + K_d \frac{2}{T_s} \left[ \frac{z - 1}{z + 1} \right] \]

\[ = \frac{K_{PID}(z + a_1)(z + a_2)}{(z + 1)(z - 1)} \quad (J.7) \]

A discrete time model for the effect of valve opening on superheat has been approximated in the previous section as a first order response with a time delay. This can be represented in discrete transfer function form as shown in Equation J.8.

\[ G_{\text{valve}}(z) = z^{-d} \left( \frac{K_{\text{valve}}}{z - b_1} \right) \quad (J.8) \]

When a root locus of this transfer function is plotted, the control algorithm and gains should be selected such that the roots are within the unit circle. The derivative term results in a pole at \(-1\) which drives the system unstable independent of the location of the adjustable zero. The integral term results in a pole at \(1\). A logical decision is to place the adjustable zero so as to cancel the effect of the first order pole. The resulting response can achieve zero steady-state error, but is slower than simply using the proportional term only. Thus a proportional control algorithm is selected and tuned to achieve the fastest response time, while avoiding large oscillations. However, the reader should note that even with this feedback controller, the valve response is slower than the compressor, and will be unable to reject the disturbances caused by the compressor. This feedback control will simply drive the average superheat response to the desired value. The reader should also note the connection between the TXV control and the proportional control, namely that the TXV control is a proportional control with an additional sensor dynamic.

**J.4 Feedforward control**

Because the superheat reacts faster to compressor changes than to valve changes, a feedback controller will be unable to reject the disturbance to superheat caused by compressor cycling. However, because the nature of the compressor signal is known a priori, a feedforward controller can be used to reject this disturbance.
Assuming that the effect of compressor on superheat is also a first order response (Equation J.9) the ideal feedforward controller would be a negative compressor model multiplied with the inverse valve model, as shown in Equation J.10. Note that to implement this control algorithm, the compressor signal must be known d time steps in advance.

\[
G_{\text{comp}}(z) = \left( \frac{K_{\text{comp}}}{z - b_2} \right)
\]

(J.9)

\[
G_{ff}(z) = -z^d \left( \frac{K_{\text{comp}}}{K_{\text{valve}}} \right) \left( \frac{z - b_1}{z - b_2} \right)
\]

(J.10)

This approach can be implemented two ways. First the delay can be estimated, and the combined gain and pole-zero locations can be tuned manually. An alternative method would be to estimate the compressor and valve models online, and simultaneously implement the feedforward control. This adaptive feedforward approach offers the advantages that no tuning is required, and the controller should adapt to different operating conditions. However, the task of identifying a Multi-Input Single-Output (MISO) system is not trivial.

The identification algorithm assumes a second order state space model, parameterized in a canonical form. A Boot-Strapping identification algorithm is used (Ahmed & Sait) with the addition of a variable forgetting factor (Fortescue)(Cordero). The assumed model is of form in Equation J.11. The matrices

\[
A \in \mathbb{R}^{n \times n}, B \in \mathbb{R}^{n \times d}, F \in \mathbb{R}^{n \times d}, L \in \mathbb{R}^{n \times d}, C \in \mathbb{R}^{m \times n}
\]

are canonically parameterized as shown in Equation J.11

\[
A = \begin{bmatrix}
0 & 1 & 0 & \cdots & 0 \\
\vdots & \ddots & \ddots & \ddots & \vdots \\
0 & \cdots & 0 & 1 \\
\end{bmatrix}
\]

\[
B = [b_1, \cdots, b_n]^T
\]

\[
F = [f_1, \cdots, f_n]^T
\]

\[
L = [l_1, \cdots, l_n]^T
\]

\[
C = [1 \ 0 \ \cdots \ 0]
\]

(J.11)

\[
x(k + 1) = Ax(k) + Bu(k) + Fw(k) + L\varepsilon(k)
\]

\[
y(k) = Cx(k) + \varepsilon(k)
\]

(J.12)

For this MISO system with n=2, we define the parameter vector \(\theta\), and the regressor \(\Phi\) as in Equation J.13 and J.14, such that the output can be written as in Equation J.15. Then the classical recursive identification routine given in Equations J.16-J.19 and discussed in Astrom can be used. The parameter \(\lambda\) is a forgetting factor to avoid
problems with lack of excitation conditions. This parameter is allowed to vary based on the amount of excitation information.

\[ \theta = \begin{bmatrix} a_1 & a_2 & b_1 & b_2 & f_1 & f_2 & l_1 & l_2 \end{bmatrix}^T \] (J.13)

\[ \phi(k) = \begin{bmatrix} x_1(k-2) & x_2(k-2) & u(k-1) & u(k-2) & \ldots \end{bmatrix}^T \] (J.14)

\[ y(k) = \phi^T(k) \theta(k) + \varepsilon(k) \] (J.15)

\[ \theta(k+1) = \theta(k) + K(k)[y(k) - \phi^T(k) \theta(k)] \] (J.16)

\[ K(k) = P(k-1)\phi(k)[I + \phi^T(k)P(k-1)\phi(k)]^{-1} \] (J.17)

\[ P(k) = [I - K(k)\phi^T(k)]P(k-1) \lambda(k) \] (J.18)

\[ \lambda(k) = 1 - \frac{\varepsilon(k)^2}{\sigma} [I + \phi^T(k)P(k-1)\phi(k)]^{-1} \] (J.19)

Using the identified model, a certainty equivalence control law is employed in feedforward. The derivation of the certainty equivalence control law is given as follows (with \( q \) as the forward shift operator). The output is defined in Equation J.20. For \( n=2 \) we find \( (qI-A)^{-1} \) as shown in Equation J.12. Assuming certainty equivalence \( \varepsilon(k) = y(k) - r(k) = 0 \), and since \( C = [1 \ 0] \), we write the control law as in Equation J.22. Then by shifting the time indices, we solve for \( u(k) \) as in Equation J.23.

\[ y(k) = C(qI - A)^{-1}[Bu(k) + Fw(k) + L\varepsilon(k)] \] (J.20)

\[ (qI - A)^{-1} = \frac{1}{q^2 - a_2q - a_1} \begin{bmatrix} q - a_2 & 1 \\ a_1 & q \end{bmatrix} \] (J.21)

\[ \begin{bmatrix} q^2 - a_2q - a_1 \end{bmatrix} \phi(k) = \\
[\begin{bmatrix} b_1(q - a_2) + b_2 \end{bmatrix}] u(k) + \\
[\begin{bmatrix} f_1(q - a_2) + f_2 \end{bmatrix}] w(k) + \\
[\begin{bmatrix} l_1(q - a_2) + l_2 \end{bmatrix}] \varepsilon(k) \] (J.22)

\[ b_1u(k) = -(b_2 - b_1a_2)u(k-1) \]
\[ -f_1w(k) - (f_2 - f_1a_2)w(k-1) \]
\[ -l_1\varepsilon(k) - (l_2 - l_1a_2)\varepsilon(k-1) \]
\[ +r(k+1) - a_2r(k) - a_1r(k-1) \] (J.23)

Because this is a linear feedforward control law, we would not expect to compensate for the nonlinear effects such as refrigerant pooling. Additionally, the adaptive form of this control law requires that “persistence of excitation” to converge to the correct parameters. Simple signals such as the on-off signal applied to the compressor
may not excite the system sufficiently to correctly identify the dynamics. In this case, the controller will result in suboptimal performance. This problem might be avoided by providing a turning the compressor on and off at random intervals (pseudo random binary sequence) during the startup phase.

J.5 Iterative learning control

Iterative learning control (ILC) is a relatively new control technique that can be used to improve the transient performance of systems that operate on a cycle. The technique is motivated by the observation that if a system performs the same action repeatedly, a controller should be able to learn from the errors of the previous iterations to improve the performance during the current iteration. As explained in Moore, the scheme is as follows. For the $k^{th}$ iteration, an input signal, $u_i(t)$, is applied to the system resulting in an output signal, $y_i(t)$. Based on the error between the desired output and the actual output, $e_i(t)=r_i(t)-y_i(t)$, the ILC algorithm computes a modified input signal for the next iteration, $u_{i+1}(t)$.

The ILC controller operates as a feedforward controller, resulting in the control law given in Equation J.24, where $G_{fb}(z)$ is the feedback controller used. $u_k^{ILC}(t)$ is the contribution from the ILC algorithm and is calculated as shown in Equation J.25 where $G_{Learn}(z)$ is the learning transfer function and $Q(z)$ is a stabilizing filter. Because the ILC algorithm will attempt to “learn” random noise, which leads to instability, a stabilizing filter is used. Messner suggests the zero-phase, non-causal Gaussian filter in Equation J.26, where $t_s$ is the sample time, and $\sigma$ is a tunable parameter. Small values of $\sigma$ will result in high bandwidth of the ILC, but can also cause the ILC to not converge.

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$$u_k(t) = G_{fb}(z)e_k(t) + u_k^{ILC}(t)$$  \hfill (J.24)

$$u_k^{ILC}(t) = Q(z)\left[ u_{k-1}^{ILC}(t) + G_{Learn}(z)e_{k-1}(t) \right]$$  \hfill (J.25)

$$Q(z) = \frac{1}{\sigma \sqrt{2\pi}} \sum_{i=-N}^{N} e^{-\frac{(-itz_s)^2}{2\sigma^2}} z^i$$  \hfill (J.26)

Ideally, the ILC algorithm will converge in determining the input signal that will minimize the error during a cycle. The ILC approach is explained in detail in Moore where conditions are given regarding convergence of the ILC algorithm. These conditions for convergence need to be considered for this application. First, the cycle time must be fixed. For this application this is true; the cycle time would be determined as part of the system design.

Second, the dynamics of the system should be the same for each iteration, and the initial conditions are assumed to the same for each iteration. For this study this is not strictly true. As the capacity demand changes over time, presumably the ‘on’ time of the cycle would be changed to match the demand. For this study it is assumed that the changes in demand would occur slowly over time, allowing the ILC sufficient time to converge. However, the dynamics of the system would also change with operating condition. In this study it is assumed that a conventional feedback controller would be able to regulate the average response to a desired setpoint, and the ILC would simply be used to reject the disturbance caused by the rapid cycling.

Third, as cited in Moore guaranteeing convergence of the ILC with actuator constraints is still an open problem. For this study it is possible that rejecting the disturbance of rapid cycling would require the valve to
exceed its physical limitations. Should this occur, the logical conclusion is that the underlying physics of the problem will prevent any controller from working properly.

J.6 List of references


