Investigation of Control Strategies for Reducing Mobile Air Conditioning Power Consumption

M. H. Dane, N. R. Miller, A. G. Alleyne, C. W. Bullard, and P. S. Hrnjak

ACRC TR-199

August 2002
The Air Conditioning and Refrigeration Center was founded in 1988 with a grant from the estate of Richard W. Kritzer, the founder of Peerless of America Inc. A State of Illinois Technology Challenge Grant helped build the laboratory facilities. The ACRC receives continuing support from the Richard W. Kritzer Endowment and the National Science Foundation. The following organizations have also become sponsors of the Center.

Alcan Aluminum Corporation
Amana Refrigeration, Inc.
Arçelik A. S.
Brazeway, Inc.
Carrier Corporation
Copeland Corporation
Dacor
Daikin Industries, Ltd.
Delphi Harrison Thermal Systems
General Motors Corporation
Hill PHOENIX
Honeywell, Inc.
Hydro Aluminum Adrian, Inc.
Ingersoll-Rand Company
Kelon Electrical Holdings Co., Ltd.
Lennox International, Inc.
LG Electronics, Inc.
Modine Manufacturing Co.
Parker Hannifin Corporation
Peerless of America, Inc.
Samsung Electronics Co., Ltd.
Tecumseh Products Company
The Trane Company
Valeo, Inc.
Visteon Automotive Systems
Wolverine Tube, Inc.
York International, Inc.

For additional information:

Air Conditioning & Refrigeration Center
Mechanical & Industrial Engineering Dept.
University of Illinois
1206 West Green Street
Urbana, IL 61801

217 333 3115
# Table of Contents

List of Figures ............................................................................................................................. V  
List of Tables ............................................................................................................................... vii  

1. Introduction ............................................................................................................................. 1  
   1.1 Introduction ......................................................................................................................... 1  
   1.2 Objectives ......................................................................................................................... 1  
   1.3 Motivation ......................................................................................................................... 1  

2. Experimental Facilities ......................................................................................................... 3  
   2.1 Test Loop .......................................................................................................................... 3  
   2.2 Air Loops .......................................................................................................................... 3  
   2.3 Refrigeration Loops ......................................................................................................... 5  
   2.4 Hardware Modifications .................................................................................................. 5  

3. Controls Equipment .............................................................................................................. 7  
   3.1 Introduction ....................................................................................................................... 7  
   3.2 Hardware .......................................................................................................................... 7  
   3.3 Software ............................................................................................................................ 7  

4. Controller Design ................................................................................................................... 9  
   4.1 TXV Model ....................................................................................................................... 9  
   4.2 Proportional Controller .................................................................................................... 11  
   4.3 Proportional - Integral Controller .................................................................................... 11  
   4.4 Dual SISO Control .......................................................................................................... 12  
   4.5 Multi Input Multi Output Controller ............................................................................. 13  

5. Results .................................................................................................................................... 29  
   5.1 Experimental Procedure ................................................................................................. 29  
   5.2 Compressor Speed Steps ................................................................................................. 30  
      5.2.1 Idle Conditions ......................................................................................................... 31  
      5.2.2 City Conditions ....................................................................................................... 33  
      5.2.3 Highway Conditions .............................................................................................. 35  
   5.3 Evaporator Fan Steps ...................................................................................................... 37  
      5.3.1 Idle Conditions ....................................................................................................... 37  
      5.3.2 City Conditions ....................................................................................................... 40  
      5.3.3 Highway Conditions .............................................................................................. 43  
   5.4 Full Condition Steps ...................................................................................................... 46  
   5.5 Start Up Response .......................................................................................................... 50  
   5.6 Humidity Addition Response .......................................................................................... 53  
      5.6.2 City Conditions ....................................................................................................... 56  
      5.6.3 Highway Conditions .............................................................................................. 59  
   5.7 Power Consumption ...................................................................................................... 62
List of Figures

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Figure 4.1-1</td>
<td>TXV Schematic James [1987]</td>
<td>9</td>
</tr>
<tr>
<td>Figure 4.1-2</td>
<td>TXV simulated sensor dynamics</td>
<td>10</td>
</tr>
<tr>
<td>Figure 4.1-3</td>
<td>TXV controller MatLab model</td>
<td>10</td>
</tr>
<tr>
<td>Figure 4.2-1</td>
<td>Proportional controller MatLab model</td>
<td>11</td>
</tr>
<tr>
<td>Figure 4.3-1</td>
<td>PI controller MatLab model</td>
<td>12</td>
</tr>
<tr>
<td>Figure 4.4-1</td>
<td>Dual SISO controller MatLab model</td>
<td>13</td>
</tr>
<tr>
<td>Figure 4.5-1</td>
<td>Pseudo Random Binary Sequence</td>
<td>14</td>
</tr>
<tr>
<td>Figure 4.5-2</td>
<td>Measured and model simulated superheat</td>
<td>15</td>
</tr>
<tr>
<td>Figure 4.5-3</td>
<td>Measured and model simulated evaporator refrigerant inlet temperature</td>
<td>15</td>
</tr>
<tr>
<td>Figure 4.5-4</td>
<td>Estimator Simulink model</td>
<td>17</td>
</tr>
<tr>
<td>Figure 4.5-5</td>
<td>Estimator results (note that the estimated and measured data plot on top of each other)</td>
<td>17</td>
</tr>
<tr>
<td>Figure 4.5-6</td>
<td>Zoomed in estimator results</td>
<td>18</td>
</tr>
<tr>
<td>Figure 4.5-7</td>
<td>Controller Simulink model</td>
<td>19</td>
</tr>
<tr>
<td>Figure 4.5-8</td>
<td>Error for superheat and evaporator refrigerant inlet temperature set point steps</td>
<td>20</td>
</tr>
<tr>
<td>Figure 4.5-9</td>
<td>Expansion valve and compressor command signal response to set point steps</td>
<td>21</td>
</tr>
<tr>
<td>Figure 4.5-10</td>
<td>Error for pulse input of 5°C superheat and 3°C evaporator refrigerator inlet temperature</td>
<td>21</td>
</tr>
<tr>
<td>Figure 4.5-11</td>
<td>Superheat and evaporator refrigerator inlet temperature error response after pulse inputs (see text)</td>
<td>22</td>
</tr>
<tr>
<td>Figure 4.5-12</td>
<td>Expansion valve and compressor command signal responses to pulse inputs (see text)</td>
<td>22</td>
</tr>
<tr>
<td>Figure 4.5-13</td>
<td>Error signals following superheat pulse</td>
<td>23</td>
</tr>
<tr>
<td>Figure 4.5-14</td>
<td>Control signals after superheat pulse</td>
<td>23</td>
</tr>
<tr>
<td>Figure 4.5-15</td>
<td>Error signals after evaporator refrigerator inlet temperature pulse</td>
<td>24</td>
</tr>
<tr>
<td>Figure 4.5-16</td>
<td>Control signal after evaporator refrigerator inlet temperature pulse</td>
<td>24</td>
</tr>
<tr>
<td>Figure 4.5-17</td>
<td>Error signal after pulse inputs (2nd order model)</td>
<td>25</td>
</tr>
<tr>
<td>Figure 4.5-18</td>
<td>Control signals after pulse inputs (2nd order model)</td>
<td>26</td>
</tr>
<tr>
<td>Figure 4.5-19</td>
<td>Error signals after superheat pulse (2nd order model)</td>
<td>26</td>
</tr>
<tr>
<td>Figure 4.5-20</td>
<td>Control signals after superheat pulse (2nd order model)</td>
<td>27</td>
</tr>
<tr>
<td>Figure 4.5-21</td>
<td>Errors after pulse of evaporator refrigerator inlet temperature (2nd order model)</td>
<td>27</td>
</tr>
<tr>
<td>Figure 4.5-22</td>
<td>Control signals after evaporator refrigerator inlet temperature pulse (2nd order model)</td>
<td>28</td>
</tr>
<tr>
<td>Figure 4.5-23</td>
<td>Compressor steps with TXV control at idle conditions</td>
<td>32</td>
</tr>
<tr>
<td>Figure 4.5-24</td>
<td>Compressor steps with proportional control at idle conditions</td>
<td>33</td>
</tr>
<tr>
<td>Figure 4.5-25</td>
<td>Compressor steps with PI control at idle conditions</td>
<td>33</td>
</tr>
<tr>
<td>Figure 4.5-26</td>
<td>Compressor steps with TXV control at city conditions</td>
<td>34</td>
</tr>
<tr>
<td>Figure 4.5-27</td>
<td>Compressor steps with proportional control at city conditions</td>
<td>34</td>
</tr>
<tr>
<td>Figure 4.5-28</td>
<td>Compressor steps with PI control at city conditions</td>
<td>35</td>
</tr>
<tr>
<td>Figure 4.5-29</td>
<td>Compressor steps with TXV control at highway conditions</td>
<td>36</td>
</tr>
<tr>
<td>Figure 4.5-30</td>
<td>Compressor steps with proportional control at highway conditions</td>
<td>36</td>
</tr>
<tr>
<td>Figure 4.5-31</td>
<td>Compressor steps with PI control at highway conditions</td>
<td>37</td>
</tr>
<tr>
<td>Figure 4.5-32</td>
<td>Evaporator fan steps with TXV control at idle conditions</td>
<td>38</td>
</tr>
<tr>
<td>Figure 4.5-33</td>
<td>Evaporator fan steps with proportional control at idle conditions</td>
<td>39</td>
</tr>
<tr>
<td>Figure 4.5-34</td>
<td>Evaporator fan steps with PI control at idle conditions (Teri)</td>
<td>39</td>
</tr>
<tr>
<td>Figure 4.5-35</td>
<td>Evaporator fan steps with dual SISO control at idle conditions</td>
<td>40</td>
</tr>
<tr>
<td>Figure 4.5-36</td>
<td>Evaporator fan steps with TXV control at city conditions</td>
<td>41</td>
</tr>
<tr>
<td>Figure 4.5-37</td>
<td>Evaporator fan steps with proportional control at city conditions</td>
<td>42</td>
</tr>
<tr>
<td>Figure 4.5-38</td>
<td>Evaporator fan steps with PI control at city conditions</td>
<td>42</td>
</tr>
<tr>
<td>Figure 4.5-39</td>
<td>Evaporator fan steps with TXV control at city conditions (Teri)</td>
<td>43</td>
</tr>
<tr>
<td>Figure 4.5-40</td>
<td>Evaporator fan steps with dual SISO control at city conditions</td>
<td>43</td>
</tr>
<tr>
<td>Figure 4.5-41</td>
<td>Evaporator fan steps with TXV control at highway conditions</td>
<td>44</td>
</tr>
<tr>
<td>Figure 4.5-42</td>
<td>Evaporator fan steps with proportional control at highway conditions</td>
<td>45</td>
</tr>
<tr>
<td>Figure 4.5-43</td>
<td>Evaporator fan steps with PI control at highway conditions</td>
<td>45</td>
</tr>
<tr>
<td>Figure 4.5-44</td>
<td>Evaporator fan steps with TXV control at highway conditions (Teri)</td>
<td>46</td>
</tr>
<tr>
<td>Figure 4.5-45</td>
<td>Evaporator fan steps with dual SISO control at highway conditions</td>
<td>46</td>
</tr>
</tbody>
</table>
List of Tables

Table 5.1-1 Test Matrix ................................................................................................................................................................30
Table 5.2-1 Output settings and values .................................................................................................................................. 31
1. Introduction

1.1 Introduction
Most refrigeration systems use a very basic control strategy of single-input single-output controllers on the expansion valve or compressor. The expansion valve is normally controlled off of the difference in temperature between the inlet and outlet of the evaporator, as is the case of a thermal expansion valve (TXV). A compressor’s displacement is normally controlled only on the suction line pressure. While these types of control work reasonably well, there are situations where they are not ideal. Thermal expansion valve controlled systems often suffer from valve “hunting”. As will be shown, neither control strategy minimizes energy consumption. When systems actually try controlling the compressor and expansion valve (using a conventional TXV and an evaporator pressure based compressor displacement control), the two separate controllers can end up fighting one another. The goal of this work is the design of a controller capable of reducing air conditioning system energy consumption with improved transient performance.

1.2 Objectives
The objective of the following experiments is to fully understand the capabilities of conventional single-input single-output (SISO) controllers on refrigeration systems and the effects these controllers have on the power needed to operate the system. In addition to standard SISO controllers, a multi-input multi-output (MIMO) controller will be tested. This will serve as a base line for future testing done with the completed modeling of several types of refrigeration systems and the controllers developed from those models in other projects. The overall project goals are the development of MIMO controllers that can improve consumer comfort while simultaneously reducing the climate control system power consumption.

In addition to the experiments run on both SISO and initial MIMO controllers, several experiments were run to validate the overall project models as well as serve as the data for building SISO controllers. The tests are run by switching individual control inputs and recording the resulting effects on the temperatures and pressures of the system. The switching pattern is done according to the pseudo random binary sequence generated in MatLab. In addition to switching individual inputs, an additional experiment was run switching the condenser air flow rate, evaporator air flow rate, electronic expansion valve, and compressor simultaneously and recording the same pressure and temperature changes.

1.3 Motivation
The ultimate goal of this study is to reduce the shaft power consumption of the compressor. Much work has been done improving the efficiency of individual components of air conditioning systems; however, there is still room for improvements in the strategy of controlling air conditioning systems. Reducing the power consumption benefits the consumer with increases in fuel economy and benefits the environment, indirectly, by reducing the emission which impact global warming.

Automotive air conditioning systems have traditionally been operated in a very inefficient manner. There are two large inefficiencies seen in automotive air conditioning systems that leave room for considerable improvement. The first area is that automobiles often operate with 100% fresh air. Many automobiles have options for turning recirculation on or off. Operating with 100% fresh air is extremely inefficient. Quoting Forrest and
Bhatti [2002] (p.58) “this is like air conditioning a home or building with screen windows!” One would never dream of doing this as it wastes tremendous amounts of energy. The second area of inefficiency comes from the fact that automobiles have oversized air conditioning systems for fast initial pull downs. After the initial pull down of the passenger compartment the chilled air must be reheated before being returned to the passenger compartment. Reheating the air after it has been cooled down, obviously throws away some of the work done by the air conditioning compressor. Some reheat is still useful as it also lowers the relative humidity of the air stream to a comfortable level.

Forrest and Bhatti [2002] offer two strategies to address these problems. Their first strategy is known as the air inlet mixture control strategy. With this strategy the key is to blend “the optimal amount of re-circulated air with outside air” (Forrest and Bhatti [2002] (p.58)). By re-circulating the passenger compartment air the system can decrease the cooling load on the air conditioner. This decrease of the cooling load decreases the power consumption of the air conditioner.

The second strategy Forrest and Bhatti [2002] offer is known as the series reheat reduction strategy. In principle, this strategy reduces compressor shaft power by elevating “the evaporator air outlet temperature or outlet refrigerant pressure instead of reheating the cooled and dehumidified air” (Forrest and Bhatti [2002] (p67)). Increasing the outlet refrigerant pressure is directly related to increasing the evaporator refrigerant inlet temperature as the evaporator is at saturation pressure and temperature. Therefore by increasing the evaporator refrigerant inlet temperature, power consumption can be decreased because compressor work is not wasted by reheating the chilled air.

It is also desirable to have the evaporator as full as possible to ensure that energy is not being wasted. Keeping the evaporator full maximizes the heat transfer from the refrigerant to the air. Additionally, for normal refrigeration systems COP is the standard measure of performance; however this isn’t true for this case. In automotive air conditioners as cooling load changes, for example with re-circulation, the COP stays almost constant because the power consumption varies directly with the cooling load. Since COP is no longer of concern, one has the freedom to seek a strategy which reduces power consumption as much as possible, with the only restriction being ensuring that the cooling load needed to cool the passenger compartment can still be met.

An overall strategy is desired that reduces power consumption while simultaneously insuring that passenger comfort is maintained. This leads to the idea of using the vapor compression system as a source of chilled air at a fixed, or slowly varying temperature. The control of the air conditioner would then be a regulator designed to hold the evaporator at a fixed temperature. This temperature should be picked just low enough to adequately cool the passenger compartment with just enough fresh air mixed to ensure good quality air. This temperature will be higher than traditionally used in automotive systems, as the evaporator air outlet temperature is usually set a few degrees above 0ºC. The temperature would be lowered initially to help with the initial cooling of the passenger compartment and raised again at normal operating conditions. This, tied to the idea of keeping the evaporator as full as possible, reduces the overall power consumption of the air conditioner. The air conditioner controller should then keep the evaporator refrigerant inlet temperature at as high a value as possible while ensuring that the superheat is at a minimum.
2. Experimental Facilities

2.1 Test Loop
The Mobile Air Conditioning Lab was originally built by Weston [1996]. The basic facility includes an air loop for the condenser, a second air loop for the evaporator, and a refrigeration loop as shown in Figure 2.2-1. The air loops are specifically constructed to insulate the air from the room, and to provide accurate measurements of the air speed and temperatures. The refrigeration loop is mainly constructed from 1994 Ford Crown Victoria parts, including the compressor, evaporator, and condenser. The expansion valve is an electronically controlled valve made by Sporlan. Several modifications have been made to the original system, both on the air and refrigerant side, including modifications for these experiments. The modifications made previous to this experiment include an evaporator humidity control system developed by Whitchurch et al [1997], an oil separator added by Chappell et al [2000], and an oil test sample section added by Drozdek et al [2000].

2.2 Air Loops
The two air loops are square sheet metal duct sections which house the heat exchangers and thermocouple grids to measure the air side temperatures both before and after the heat exchangers. Each duct feeds into a reducing nozzle to feed the air fans. Each fan is driven by an electric motor controlled by a Toshiba variable frequency drive so that the normal range of air flow rates can be obtained. The air is then passed through a section of round ducting, through a venturi to measure the air flow rate, and is then dumped back into the plenum.

The condenser loop has an inlet for fresh air from the room. The condenser loop can also mix re-circulated warm air back in with the fresh air to preheat the air going into the condenser. For these experiments warm return air was partially mixed in order to simulate more realistic conditions. The evaporator loop is completely closed and continually uses the same air. The evaporator air loop also contains a 7 kW duct heater which is controlled with a proportional-integral-derivative controller. The heater heats the air back up after it is cooled by the evaporator to insure a constant inlet air temperature.

The temperatures in the air loops are measured with Omega type T thermocouples. The thermocouples are placed in a grid formation both before and after the heat exchangers. Each grid is wired in such a way as to average the temperature reading across the entire grid as shown in Appendix B. The pressures are measured with pressure transducers, also from Omega which either output 0-5 volts or 4-20 mA, and are used to calculate the velocity through the venturi.

In addition the evaporator loop has humidity control. The humidity is measured before and after the evaporator with Vaisala HMP35A humidity sensors. It is controlled with an electrical PID controller and an electronically actuated valve off of the inlet humidity sensor.
Figure 2.2-1 Test Facility Schematic
2.3 Refrigeration Loops

The refrigeration loop employs a Ford FS 10 swash plate, fixed displacement compressor driven by a 7.5 HP electric motor. The compressor of course would normally be controlled by a belt driven directly from the engine with a pulley ratio of approximately one. For this reason the electric motor is sized to match speeds from idle conditions, approximately 800 RPM, to highway conditions, approximately 3000 RPM. The speed of the motor is controlled with a PoWrMaster® variable frequency drive by IDM Controls, Inc. and in combination with the electric motor the two can actually operate over a greater range than required by driving conditions.

The condenser, also from the 1994 Ford Crown Victoria, is an aluminum cross flow tube and fin heat exchanger. The refrigerant enters the top of the condenser as a compressed vapor and is cooled to the liquid state by heat transfer to the air being blown over the air side of the condenser. After making two passes across the face of the condenser the refrigerant then leaves from the bottom and goes to the high side receiver. The receiver insures that the refrigerant is a saturated liquid as it goes to the expansion valve. This is not an original part of the Ford air conditioning system and is a modification made to make the system similar to most automotive air conditioning systems that employ thermal or electronic expansion valves.

The refrigerant then travels through a mass flow meter on its way to the electronic expansion valve. The flow meter is a Micromotion™ Coriolis type mass flow meter. This gives an accurate measurement of the overall flow rate as the refrigerant is all in the saturated liquid state. The original Ford orifice tube throttling device has been replaced with a Sporlan electronic expansion valve (EEV). This electronic expansion device gives accurate control of the throttling area with a current input of 4-20mA. The EEV is oversized to insure that step changes in valve opening are quickly reached. In turn only the first 30 percent of the valve operating ranged are used during normal conditions.

After the refrigerant is expanded, and therefore dramatically cooled, it travels as a two phase liquid to the evaporator. The evaporator is an alloy cross flow plate and fin heat exchanger also from Ford. The refrigerant takes in heat from the air and leaves the evaporator a superheated vapor. In this experimental facility the refrigerant is valved around an oil separator used in previous experiments to test the effects of oil return on the compressor. When the oil separator was added the standard Ford low side accumulator was removed. The vapor refrigerant then makes its way back to the compressor.

The refrigeration loop is instrumented before and after the 4 major devices for pressure and temperature measurements. The pressure is measured with Omega pressure transducers similar to the air side pressure transducers, except for much higher pressure ranges. The temperatures are measured with type T thermocouples. There are also sight glasses and Schrader valves in several locations to observe the internal conditions in the refrigerant lines and to add refrigerant.

2.4 Hardware Modifications

Refrigerant side modifications were made to simplify and shorten the amount of tubing used in the system. The overall system had become quite complex due the large number and variety of experiments run in the MAC lab. The oil mixture sampling assembly was removed due to its large size and complexity. It contained eight different valves to separate sections of refrigerant and oil mixtures while the system was running. Several of the valves had
small leaks and since there was no need for this information the entire section was removed. The oil concentration sensor was removed for use in another lab and therefore the valves and the extra tubing used in the assembly were also removed. There was another oil separator placed after the compressor, in addition to the one before the compressor, which was completely removed as it was in a very inconvenient location and was also not needed for the current experiments.

Modifications were made to the condenser air loop at the reducing nozzle. This was done because the air was first converged to measure the temperature more accurately after the condenser and then diverged to the original duct size and converged again to the fan. This tends to cause a large pressure drop in the air ducting which makes for an increase in fan work. The reducing nozzle was reworked to just continue the convergence after the thermocouple grid.

On both the condenser and evaporator air side loop a section of ducting was added in between the heat exchanger section and the temperature measurement section to insure that there was proper mixing and flow evening. If the temperature is measured too close to the heat exchangers there is a good possibility that results will be inaccurate. With the new section added the air side temperature measurements could more accurately represent the overall flow. The three air loop modifications are shown in more detail in Appendix A.
3. Controls Equipment

3.1 Introduction
The controls equipment used in this work were selected on the basis of ease of modification of control strategies as well as the possibility of controlling a wide variety of components. The equipment needed to be able to handle control strategies ranging from simple single-input, single-output (SISO) to complex multi-input, multi-output (MIMO) controllers. With this in mind control over the following four components were required: compressor speed, condenser and evaporator air flow rates, and electronic expansion valve opening. With control over all four of these components, experiments can be run ranging from disturbances to a simple one component controller all the way to control of several components simultaneously using a MIMO controller.

3.2 Hardware
In order to implement the control software and hardware a new computer was purchased. This insured that the computer programs could keep up in virtual “real time” with the actual system. The computer was a Dell Optiplex GX240, with a Pentium 4 -1.4 GHz processor. Added to the computer were two PCI plug and play boards from Measurement Computing, the PCI-DDA 08/12 and the PCI-DAS 1200/JR.

The PCI-DDA 08/12 board produces the outputs from the computer. The board has 8 channels of analog output that operate from -10 to +10 volts, however, for these experiments a range of -5 to +5 volts was used. These voltages were then converted to a 4-20 mA range needed by the electronic expansion valve and the variable frequency drives by Omega 5B modules. For the compressor the range of -5 to +5 volts corresponded to 0 – 60 Hz. For the electronic expansion valve -5 to +5 volts corresponded to 0% - 100 % open. The 5B modules, part number OM5-IVI, take an input of -5 to +5 volts and convert it linearly to 4-20 mA while keeping the two sides electronically isolated.

The PCI-DAS 1200/JR board reads the inputs from the signal conditioner. The board had 8 analog input channels which can read voltages in the range of -10 to +10 volts. The thermocouple voltages read in by the PCI-DAS 1200/JR board are also processed by Omega 5B modules, part number OM5-LTC. The signal conditioners used are specified for type T thermocouples and output an isolated linearized voltage from 0-5 volts for a preset temperature range. During operation excessive noise was encountered and a RC circuit was added to the filter the signal from the signal conditioners to the PCI board. In addition, the temperatures seen in the system only comprised of a small range of the signal conditioners capabilities. This led to the voltage differences required to be read to be so small that the signal the PCI board showed a discrete signal with ½ºC steps. Therefore, an op-amp circuit was added before the RC filter to amplify the temperature range of -10ºC to 30ºC to be over the range of -10 to +10 volts. The wiring diagrams for the wiring from the PCI boards through the signal conditioners to the selected components is shown in Appendix B.

3.3 Software
The design software used to make the controllers was MatLab Simulink. Simulink is a visual programming platform, which allows the user to build models in block diagram form. With the creation of special functions for the inputs and outputs, one can connect the proper input signals through the desired controllers and output them directly to the actual system. The special functions used for the hardware inputs and outputs were originally created
by Zhang [2002] and modified for specific use. The blocks call specific C code drivers which link the communication between the Simulink files and the analog input and output boards. The modifications made to the C code drivers include the Bus setup for this specific computer as well as changes need to switch from the PCI-DAS 1200 board to the PCI-DAS 1200/JR board.

The benefits of using Simulink are that models can be made quite easily. When the models have been completed, the program WinCon is used to compile and run the models in real time. In addition, the program VenturCom RTX is used for the overall merging of WinCon, Simulink and the hardware. The benefit of using this combination of programs is that certain settings can be altered while the models are running in real time. For instance, the gains and settings of say the proportional controllers can be adjusted on the fly. This allows for easier tuning of the controllers. This also allows certain settings to be used for start up conditions and other controllers to be used during normal running conditions.
4. Controller Design

4.1 TXV Model

A very common method used to control superheat in air condition systems is the use of a thermostatic expansion valve, known as a TXV or TEV. A TXV senses the difference in temperature of the inlet and outlet of the evaporator using the pressures of the refrigerant entering the evaporator and what is commonly referred to as the bulb pressure (actually a measure of evaporator outlet temperature). The bulb pressure is the pressure inside of a bulb which is placed onto the tubing exiting the evaporator; see the schematic in Figure 4.1-1. As the temperature rises at the exit of the evaporator the heat is transferred through the walls of the refrigerant tubing and bulb to the fluid inside the bulb. As this temperature of the fluid in the bulb rises, so does the pressure in the bulb and through a small tube the pressure is transferred back to the TXV and drives the valve spindle.

![TXV Schematic](image)

Figure 4.1-1 TXV Schematic James [1987]

This control type is highly developed and inexpensive to make (because of its long development), however can lead to several problems. The problems stem from the first order lag which is associated with the heat transfer to the bulb. However, the relatively slow, first order response occurs because the temperature of the refrigerant is sensed after the heat is transferred from the refrigerant exiting the evaporator, through the tube walls, through the bulb walls, to the fluid inside the bulb which in turn stabilizes at its saturation pressure.

The first order response (a "lag") involved in the TXV is the only real difference between it and a proportional controller, which directly measures the temperature at the inlet and outlet of the evaporator. With this knowledge a simple model of the TXV was created by applying a first order response to the temperature measured at the outlet of the evaporator. Mathematically, this can be represented by

\[
\frac{T(t)}{T(0)} = \frac{1}{\tau} \cdot e^{-\frac{t}{\tau}}
\]

(1)

Transforming this to the Laplace domain gives
\[
\frac{T_{\text{bulb}}}{T_{\text{ero}}} = \frac{1}{1 + \tau s}
\]  

The time constant (\(t\)) can, of course, vary depending on the particular TXV that is to be modeled; however, a common time constant, which was used in the following experiments, is on the order of 5 seconds. This time constant was verified by James [1987]. This response is shown for a step of 1°C in Figure 4.1-2. For the MatLab controller the mathematical model of the sensor dynamics was converted from continuous to discrete and implemented as shown in Figure 4.1-3.

![Simulated TXV bulb sensor dynamics](image1)

**Figure 4.1-2 TXV simulated sensor dynamics**

![TXV controller MatLab model](image2)

**Figure 4.1-3 TXV controller MatLab model**

To calculate the superheat seen by the TXV, the refrigerant temperature at the inlet of the evaporator is subtracted from the refrigerant temperature at the outlet of the evaporator, which has been through the first order lag discussed above. The superheat is subtracted from the desired set point to find the error, which is consequently multiplied by the gain to achieve the control signal desired for the electronic expansion valve. The gain used in the TXV model was chosen by manually tuning the proportional gain value while running at city conditions. Tuning
attempted to provide good performance over the entire operating range. In order to improve the performance of the TXV some gain scheduling was done. The proportional gain was varied slightly depending on the compressor speed, having a different gain for idle, city, and highway. In addition, the valve preset used to shift the voltage output to the correct range was varied according to the evaporator fan speed. With this type of model the valve can be controlled equivalently to a real TXV.

### 4.2 Proportional Controller

The next step up in controller type used in these experiments was a proportional controller. This type of controller actually varies very little in general form from the TXV. The proportional controller would be slightly more expensive to implement directly on the system, however, the proportional controller has a large advantage in that it directly senses the temperature at the outlet and inlet of the evaporator. The proportional controller was completely set up in MatLab as seen in Figure 4.2-1.

![Figure 4.2-1 Proportional controller MatLab model](image)

For this controller, the evaporator refrigerant inlet temperature is again subtracted from the evaporator refrigerant outlet temperature to calculate superheat. This value of superheat is then compared to the set point to give the error. The error is then multiplied by a proportional gain to supply the desired control signal for the electronic expansion valve. As with the TXV, the proportional gain was tuned while running the system at city conditions, hence giving the best performance at city conditions. In order to improve results, the valve presets for the proportional controller were scheduled to vary slightly according to evaporator air flow rate setting, from 13.5% open at the lowest air flow rate to 15.5% open at the highest air flow rate. No adjustment was needed for the proportional gain, as this arrangement demonstrated good performance under all operating conditions (see results in Chapter 5).

### 4.3 Proportional - Integral Controller

Proportional controllers inherently suffer from steady state error. The steady error varies directly with the offset of the set point from zero. The standard correction for this problem is the addition of an integrator to the controller. This control strategy will be explored here and the controller is known as a PI controller.
The integrator is added in parallel with the proportional gain, and continuously adds to the proportional control value a term which increases the longer the error is away from zero. This tends to drive the steady state error to zero. The error used by the controller is calculated exactly the same way as with the proportional controller. The error is multiplied not only by a proportional gain but also by an integral gain with the result of the latter operation being integrated. These two separate values are then added back together to get the control signal for the expansion valve (seen in Figure 4.3-1).

\[ \text{Error} = \text{Proportional Gain} \times (\text{Actual Superheat} - \text{Set Point}) + \text{Integral Gain} \]

Figure 4.3-1 PI controller MatLab model

The proportional gain and integral gain were tuned while running at city conditions. For a starting point the proportional gain was set to that of the proportional controller and the integral gain was set to a value one-tenth of that. These were changed, however, to achieve the best performance, once again giving the best performance at city conditions, with very good performance at all other operating conditions. It must also be noted that there seemed to be no noticeable problems of integrator wind up, which is a problem normally seen during start ups for which the integral term climbs to an extremely high value and can’t correct itself. As none of this was seen, no anti-wind up strategy has been implemented in this controller.

4.4 Dual SISO Control

One of the common problems encountered by mobile air conditioners is the frosting of the evaporator. This is obviously a very bad condition as frost can bridge the narrow space between automotive evaporator fins so that air can no longer pass through the evaporator. Normally the compressor clutch is just cycled off if the temperature of the evaporator gets to cold. It is much more desirable to be able to control the evaporator refrigerant inlet temperature to a temperature above freezing. This ensures that the air conditioner can continue to work properly.

A simple, yet effective way to accomplish this is to simultaneously control the expansion valve and compressor shaft speed or stroke (as in a variable displacement compressor). In the test system used in this work, this is most easily accomplished by varying the compressor speed, as it is directly driven by a variable speed electric motor. Both variable shaft speed and variable compressor displacement can achieve the similar results. This controller employs two single input, single output (SISO) controllers. The dynamics of the two variables are
inherently coupled, which can lead to the two controllers fighting each other. However, with proper tuning good results can be achieved. The implemented version of this controller can be seen in Figure 4.4-1.

![Figure 4.4-1 Dual SISO controller MatLab model](image)

The superheat is controlled by the PI controller discussed in section 4.3, as it obviously gave the best results of the expansion valve controllers. The selection of the compressor speed control was slightly different. First a proportional controller was tried, which gave very good results. In an attempt to improve results a PI controller was tried for the compressor control; however serious integral wind up problems were encountered. With the knowledge of how well the proportional controller worked for the compressor, attempts at trying to stop the integral wind up were abandoned. The proportional controller was then implemented to control the evaporator refrigerant inlet temperature with the compressor speed, and lead to the finalized dual SISO controller.

### 4.5 Multi Input Multi Output Controller

The basic foundation behind a multi input multi output (MIMO) controller is a state space model that captures the coupled effects of the system. Knowing the relationship between the multiple inputs and multiple outputs should allow the controller to improve all aspects of its control, such as overshoot, settling time, etc. Therefore, the first step in building a MIMO controller is coming up with the state space model. A model can be built from first principles. Alternatively, if a system already exists, the model can be found empirically. The latter course was followed for this investigation. Experiments were run by making known steps of the inputs, in our case the electronic expansion valve and compressor. The step order was generated in the form of a pseudo random binary sequence with the MatLab random number generator. The sequence is shown in Figure 4.5-1. The plant was then fed these strings of inputs. The outputs of superheat and evaporator refrigerant inlet temperature were recorded at a sample rate of 10 Hz. With this set of data, a linear model can be fit that captures the effects the valve and compressor have on superheat and evaporator refrigerant inlet temperature.
To generate the model, the MatLab function *ident* was used. *Ident* allows the user to first remove the means and also select sections of the data which seem to best capture the desired effects. With this new set of data, *ident* then can generate models of several types. For the MIMO control the state space model was selected. Models were generated of different orders to search for the best fit. In this case, the fourth order model gives good results. A comparison of results for the measured and simulated model is shown in Figure 4.5-2 and Figure 4.5-3.

The model generated by MatLab is in state space form, in the same form as shown in Phillips & Nagle (p. 150). The general state space form is given as

\[
\begin{align*}
    x(n+1) &= Fx(n) + Gu(n) \\
    y(n) &= Hx(n)
\end{align*}
\]

where

- \(x(n)\) is the state vector
- \(u(n)\) is the control signal vector to the plant (i.e. signal to the compressor and valve)
- \(y(n)\) is the output vector of the plant (i.e. the superheat and evaporator refrigerant inlet temperature)
- \(F, G, H\) are fitted matrices
To get a feel for the system, the poles of the plant were found with MatLab (using the \texttt{eig} command) to be \(0.0583 \pm 0.6102i, 0.9986, \text{ and } 0.9943\). This shows that there are two fast complex poles and two slow poles. This makes sense in that the plant step response shows some faster dynamics when steps are first made and then the general trends which are much slower and are similar to integrator ramps.
This model has four state variables. The physical meaning of these state variables is uncertain. The unknown states must be estimated from the input variables of superheat and evaporator refrigerant inlet temperature. To do this, a system is set up similar to the original state space form, with the state vector now being estimated from the inputs. The state space estimator equations, as seen in Phillips and Nagle [1995] (p. 345) are

\[ \hat{x}(n+1) = F\hat{x}(n) + Gu(n) \]  
\[ \hat{y}(n) = H\hat{x}(n) \]

where

- \( \hat{x}(n) \) is the estimated state vector.

In order to do this as accurately as possible, feedback of the actual plant output and the simulated plant output is added back into the model with an estimator gain, L. This leads to the estimator equation:

\[ \hat{x}(n+1) = F\hat{x}(n) + Gu(n) + L(y(n) - \hat{y}(n)) \]  

The estimated plant output in terms of the estimated state can be substituted to give

\[ \hat{x}(n+1) = F\hat{x}(n) + Gu(n) + L(y(n) - H\hat{x}(n)) \]

To calculate the estimator gain, pole placement was used with the MatLab place command with the poles placed just slightly faster than the plant poles. The poles used for the estimator were 0.05 ± 0.50i, and 0.90, 0.91. With the estimator gain calculated, the estimator equation was modified using the z-transform. Starting with Equation 6 and moving back one time step gives

\[ \hat{x}(n) = F\hat{x}(n-1) + Gu(n-1) + L(y(n-1) - H\hat{x}(n-1)) \]

substituting (using the z transform operator)

\[ \hat{x}(n-1) = \frac{\hat{x}(n)}{z} \]

gives

\[ \hat{x}(n) = F\frac{\hat{x}(n)}{z} + G\frac{u(n)}{z} + L\left(\frac{y(n)}{z} - H\frac{\hat{x}(n)}{z}\right) \]

This was then implemented in Simulink with the block diagram shown in Figure 4.5-4. The estimator was tested using the data collected earlier with the means removed, just as in the modeling process. The results are shown in Figure 4.5-5. The blue and green lines are the estimated values, while the red and purple lines are the actual measured temperatures. In addition, to better show the difference between the actual and estimated values Figure 4.5-6 shows a zoomed in section. These results show that the estimator works very well.
Figure 4.5-4 Estimator Simulink model

Figure 4.5-5 Estimator results (note that the estimated and measured data plot on top of each other)
The next step is to implement the estimator with control gains to control the system. The estimator and control equations are

\[
\hat{x}(n+1) = F\hat{x}(n) + Gu(n) + L(y(n) - H\hat{x}(n))
\]

(12)

\[
u(n + 1) = -K\hat{x}(n + 1)
\]

(13)

which are then combined and rearranged to give:

\[
\hat{x}(n + 1) = (F - LH)\hat{x}(n) - G\hat{x}(n) + Ly(n) - Lr
\]

(14)

with \( r \) being a reference input or desired values of superheat and evaporator refrigerant inlet temperature. This equation is then again written using the z-transform operator in a similar fashion as the estimator equation leading to

\[
\hat{x}(n) = (F - LH)\frac{\hat{x}(n)}{z} - GK\frac{\hat{x}(n)}{z} + L\frac{y(n)}{z} - Lr
\]

(15)

This is similarly implemented in MatLab with the block diagram shown in Figure . The contents of the estimator block in Figure  appear in Figure .
Figure 4.5-7 Controller Simulink model

Next the controller gains must be calculated. Initial attempts demonstrated that pole placement is at best difficult. It was decided to use the Linear Quadratic Gaussian method as described in Phillips and Nagle [1995] (p.382). The Linear Quadratic Gaussian method minimizes the cost function:

\[ J_N = \sum_{k=0}^{N} x^T(k)Q(x(k)) + u^T(k)R(k)u(k) \]  \hspace{1cm} (16)

where

- \( J_N \) is the cost function
- \( Q(k) \) is the state variable weighting function
- \( R(k) \) is the output vector weighting function

This method was implemented with the MatLab command \texttt{lqrd}, which solves for the poles and gains based on the state matrices F and G and the weighting functions Q and R. Initial testing revealed that the controller drove the command signals (motor speed and expansion valve opening) to unrealistic values. All the weight was placed on the controller signals in an attempt to keep them from saturating the system, i.e. Q was made zero, while R was made the identity matrix. This led to the controller poles being placed at 0.9923 ± 0.0606i, 0.9054, and 0.9050.

To check the gain values and controller capabilities, the model was put in place of the plant to take in control signals and output the temperatures using the discrete state space block in Simulink. It was discovered that the controller did not work under large steps such as would be seen during start up. For this reason, saturation limits were set up to ensure that the control signal stayed within a reasonable range. The electronic expansion valve limits
were set from fully closed to fully open. The upper limit was set to ensure that the stepper motor didn’t uselessly wind further open when in actuality the valve was no longer opening. The compressor saturation limits were also set up around the model’s mean value, which was city conditions, and allowed the controller to go from a speed lower than 800 RPM (idle) to slightly higher than 3000 RPM (highway, where the mass flow rate is saturated). Figure 4.5-8 shows how poorly the controller works for a step input. The control signals used in response to this step are shown in Figure 4.5-9. For this simulation, step inputs of superheat (5°C) and evaporator refrigerant inlet temperature (3°C) were applied to the respective set points. Notice that the error is larger than the step and that the compressor is stuck on the lower saturation limit. To try and test around these larger steps, the simulation was set up to start with zero error, (similar to a bumpless starting strategy). In practice, this could be implemented with the dual SISO controller handling start up and then control could be switched over to the MIMO controller when the system came into the operating range.

![Step Response](image)

**Figure 4.5-8 Error for superheat and evaporator refrigerant inlet temperature set point steps**

To further test the capabilities of the controller, as a regulator, pulses were fed into the reference signal. Initially pulses were implemented for both the superheat and evaporator refrigerant inlet temperature simultaneously, of 5°C and 3°C respectively. These results can be seen in Figure 4.5-9 and Figure 4.5-10. However the control signal still goes to the saturation limits. Therefore it was decided to implement the pulses separately for superheat and the evaporator refrigerant inlet temperature. Additionally, both pulses were decreased in magnitude until the controller stayed within the saturation limits set on the control signals.
Figure 4.5-9 Expansion valve and compressor command signal response to set point steps

Figure 4.5-10 Error for pulse input of 5°C superheat and 3°C evaporator refrigerant inlet temperature

The largest pulse the controller could handle without saturating the control limits were 0.05°C for superheat and -0.01°C for the evaporator refrigerant inlet temperature. The directions of these pulses were chosen because they simulated directions in error which correspond to the system losing superheat and the evaporator refrigerant inlet temperature getting even colder. The superheat pulse was implemented at 25 seconds and the evaporator...
refrigerant inlet temperature pulse at 75 seconds, while running the simulation for 100 seconds. The error of superheat and evaporator refrigerant inlet temperature for the overall simulation is shown in Figure 4.5-11. In addition the control signal produced to get the system back under control is shown in Figure 4.5-12.

Figure 4.5-11 Superheat and evaporator refrigerant inlet temperature error response after pulse inputs (see text)

Figure 4.5-12 Expansion valve and compressor command signal responses to pulse inputs (see text)
Figure 4.5-13 Error signals following superheat pulse

Figure 4.5-14 Control signals after superheat pulse
Figure 4.5-15 Error signals after evaporator refrigerant inlet temperature pulse

Figure 4.5-16 Control signal after evaporator refrigerant inlet temperature pulse

Figure 4.5-13, Figure 4.5-14, Figure 4.5-15, and Figure 4.5-16, show zoomed plots of the separate pulse responses. The controller can reject the disturbance; however a disturbance of such a small amplitude implies that any kind of system noise would disrupt the controller to at least this level. Even though the controller rejected the small impulses the zoomed figures show a strange behavior in that the error actually grows from the small impulse...
to almost 8°C. In addition, the error oscillates in sign on every time step. This is very unrealistic. A first guess at why this would happen comes from looking at the poles of the system. The two fast, complex poles could cause this type of behavior. Notice in Figure 4.5-6 that the estimated values actually show the fast “spikes” caused by the complex plant poles and that they are of fairly large magnitude.

In an attempt to see if this was the case a second order model was tested. The model was again found with the MatLab ident program. With a 2nd order state space model the estimator gains were again calculated with pole placement. The poles of the model were 0.9961 and 1.000. Therefore the estimator poles were placed at 0.9 and 0.91. The estimator shows very similar results to Figure 4.5-5 and Figure 4.5-6, except for not having the fast spikes caused by the complex poles. The controller gains were again found with the Linear Quadratic Gaussian method in MatLab. Similar to the 4th order model all the weight was placed on the control signal, which placed the poles at 0.9048 and 0.9052.

![Error signal pulse responses](image)

Figure 4.5-17 Error signal after pulse inputs (2nd order model)

To test the controller, pulses were introduced to the set point separately for the superheat and evaporator refrigerant inlet temperature. Just as with the 4th order model, pulse were tested until the maximum pulse that still permitted the control signal to remain below the saturation limits was found. For the 2nd order model the pulse were 0.1°C for superheat and -0.02°C for the evaporator refrigerant inlet temperature. The results for the new controller, based on the 2nd order state space model, are shown in Figures 14-20. Notice that the control signal in Figure 4.5-12, Figure 4.5-14, and Figure 4.5-16 are almost identical to the ones in Figure 4.5-18, Figure 4.5-20, and Figure 4.5-22. The major difference is that the error seen in Figure 4.5-19 and Figure 4.5-21 stays less than the initial pulse. In fact the pulse can now clearly be seen before the controller drives the errors to zero.
These results support the conclusion that the two fast complex poles produced the large increase in error with the 4th order model. There is another result that is unsettling. The fact that the error can swing back and forth with every sample rate seems unrealistic. The controller does not have rate limits so the model valve can change from fully open to fully closed within one sample time while the model compressor can instantly change shaft speed. This simplification may explain the unrealistic results observed in the simulation.
Figure 4.5-20 Control signals after superheat pulse (2nd order model)

Figure 4.5-21 Errors after pulse of evaporator refrigerant inlet temperature (2nd order model)
These results show the limitations of the Linear Quadratic Gaussian method. For this type of system that is dominated by behavior close to that of a double integrator the LQG method does not produce a suitable MIMO controller. In order for a controller of this type to work it must be designed with more robust methods and will probably require the ability to switch between different controllers based on different models for specific operating regions.
5. Results

5.1 Experimental Procedure

In order to gain understanding of the capabilities of different control strategies for mobile air conditioning systems and use these as a base line for comparison to modern control strategies, a test procedure was devised to capture the results of the controllers while being disturbed with situations as close as possible to real life. The most common type of disturbances include:

- small changes in compressor speed, as would be seen in engine speed changes while driving at nearly constant speeds
- steps in the evaporator air flow rate, as would be seen by driver changes in the fan speed
- step changes between standard driving conditions, i.e. moving from city to highway speeds
- start up conditions
- large additions of moisture into the system, as would be seen by opening the door on a humid day

These disturbances can be used to see the effectiveness of several types of controllers, including TXV, proportional, PI, and dual SISO control. The standard means of comparison would be to look at changes in the superheat and evaporator refrigerant inlet temperature throughout the experiment, as these have the greatest impact on efficiency. The exact configuration of the experiments can be seen in Table 5.1-1.
### Table 5.1-1 Test Matrix

<table>
<thead>
<tr>
<th>Test</th>
<th>Disturbance</th>
<th>Condition</th>
<th>Air Loop Temperatures</th>
<th>Compressor (RPM)</th>
<th>Condenser Air Flow Rate (cfm)</th>
<th>Evaporator Air Flow Rate (cfm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Evap.</td>
<td>Cond.</td>
<td>Initial</td>
<td>Final</td>
</tr>
<tr>
<td>1</td>
<td>Compressor PRBS</td>
<td>Idle</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>950</td>
<td>1045</td>
</tr>
<tr>
<td>2</td>
<td>Compressor PRBS</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1980</td>
</tr>
<tr>
<td>3</td>
<td>Compressor PRBS</td>
<td>Highway</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>3000</td>
<td>3300</td>
</tr>
<tr>
<td>4</td>
<td>Evaporator Air Flow Rate</td>
<td>Idle</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>950</td>
<td>950</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>Idle</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>950</td>
<td>950</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>Idle</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>950</td>
<td>950</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>Idle</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>950</td>
<td>950</td>
</tr>
<tr>
<td>5</td>
<td>Evaporator Air Flow Rate</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td>6</td>
<td>Evaporator Air Flow Rate</td>
<td>Highway</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>3000</td>
<td>3000</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>Highway</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>3000</td>
<td>3000</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>Highway</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>3000</td>
<td>3000</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>Highway</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>3000</td>
<td>3000</td>
</tr>
<tr>
<td></td>
<td>Evaporator Air Flow Rate</td>
<td>Highway</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>3000</td>
<td>3000</td>
</tr>
<tr>
<td>7</td>
<td>Condition Step</td>
<td>Idle to City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>950</td>
<td>1800</td>
</tr>
<tr>
<td></td>
<td>Condition Step</td>
<td>City to Highway</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>3000</td>
</tr>
<tr>
<td></td>
<td>Condition Step</td>
<td>Highway to City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>3000</td>
<td>1800</td>
</tr>
<tr>
<td></td>
<td>Condition Step</td>
<td>City to Idle</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>950</td>
</tr>
<tr>
<td>8</td>
<td>Start Up</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td>9</td>
<td>Humidity Addition</td>
<td>Idle</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>950</td>
<td>950</td>
</tr>
<tr>
<td></td>
<td>Humidity Addition</td>
<td>City</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>1800</td>
<td>1800</td>
</tr>
<tr>
<td></td>
<td>Humidity Addition</td>
<td>Highway</td>
<td>~80 °F</td>
<td>~80 °F</td>
<td>3000</td>
<td>3000</td>
</tr>
</tbody>
</table>

### 5.2 Compressor Speed Steps

The first set of tests were conducted to simulate small changes in compressor RPM. These types of changes would be seen with small changes in engine speed, while driving at fairly constant conditions. The tests were run at idle, city, and highway conditions. Included in these condition changes are the condenser air flow rate and compressor speed. The evaporator air flow rate was held constant at the equivalent of 250 cfm for all three conditions. The sequence used to step the compressor was generated using the Random Number Generator in MatLab, with a seed of 34,597,215,863 and a sample rate of 20 seconds. For all three conditions, steps of 10 percent in RPM were made. Table 5.2-1 shows an overall list of the values used for these experiments. In addition all the
controllers had a set point of 5°C; however the control computer and data acquisition computer read a temperature difference of approximately 1°C, which translates to the plots having the set point at 6°C. Since the dual SISO controller actually controls the compressor speed these experiments only include the TXV, proportional and PI controllers. Additional the control signal offsets were manually set. These values were tuned to a value as close to the steady state set point as possible, however there is of course some error in this, explaining why some of the steady state values are below the set point and some above.

<table>
<thead>
<tr>
<th>Evaporator Air Flow Rate (cfm)</th>
<th>Frequency Drive Setting (Volts)</th>
<th>Compressor Speed (RPM)</th>
<th>Frequency Drive Setting (Volts)</th>
</tr>
</thead>
<tbody>
<tr>
<td>150</td>
<td>0.00</td>
<td>950</td>
<td>-1.99</td>
</tr>
<tr>
<td>200</td>
<td>1.40</td>
<td>1045</td>
<td>-1.72</td>
</tr>
<tr>
<td>250</td>
<td>2.87</td>
<td>1800</td>
<td>0.23</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1980</td>
<td>0.65</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3000</td>
<td>3.42</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3300</td>
<td>4.25</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Condenser Air Flow Rate (cfm)</th>
<th>Frequency Drive Setting (Volts)</th>
</tr>
</thead>
<tbody>
<tr>
<td>800</td>
<td>1.10</td>
</tr>
<tr>
<td>950</td>
<td>2.00</td>
</tr>
<tr>
<td>1250</td>
<td>3.85</td>
</tr>
</tbody>
</table>

5.2.1 Idle Conditions

As mentioned above the compressor was stepped 10% along the pseudo random binary sequence, for idle conditions this is a step from 950 to 1045 RPM. Additionally at idle conditions the condenser air flow rate was held at 800 cfm and the evaporator air flow rate was held at 250 cfm, with both of them having approximate inlet temperatures of 80°F. Recall that the controllers were all tuned for city conditions and that due to the nature of the system, the results at idle speeds are not as good as the city and highway experiments.

First note, as expected, the TXV controller and the proportional controller show steady state error. Their results can be seen in Figure 5.2-1 and Figure 5.2-2. The TXV controller has a steady state around 3-4°C, or a steady state error of about 2-3°C. The TXV actually does a pretty good job of controlling the system for most of the experiment. However, when the compressor steps are repeated close together the superheat starts to oscillate strongly. As can be seen at the end of the experiment, superheat is nearly lost. It appears that the closed loop system with the simulated TXV response is very lightly damped when the excitation period is reduced to around 50 seconds.

The proportional controller also maintains superheat; however, it seems to fluctuate by much larger amounts than the TXV. Evident with proportional controller plot is the non-minimum phase of the system. This of course makes it even harder to control, and can explain why the proportional controller has larger fluctuations, as the superheat tends to goes down momentarily before it goes up. This causes the controller to try to compensate the
incorrect way and in fact cause the superheat to go slightly further in the incorrect direction. The fluctuations could be damped out by the TXV’s first order lag. The proportional controller has a steady state value of approximately 8-9°C, or steady state error of 2-3°C, just in the other direction from the TXV.

![Superheat changes with compressor steps with TXV control at idle conditions](image)

**Figure 5.2-1 Compressor steps with TXV control at idle conditions**

The PI controller has much better disturbance rejection than the TXV and proportional controllers, its results can be seen in Figure 5.2-3. Even at the faster switching section the PI controller maintains its steady state value at 6°C, which corresponds to no steady state error. The only problem with the controller is the large fluctuations in superheat, ranging in almost 8°C, this mostly likely stems from the controller being tuned at city conditions instead of idle conditions.
Superheat changes with compressor steps with Proportional control at idle conditions

Figure 5.2-2 Compressor steps with proportional control at idle conditions

Superheat changes with compressor steps with PI control at idle conditions

Figure 5.2-3 Compressor steps with PI control at idle conditions

5.2.2 City Conditions

For the city condition experiments, the compressor was stepped from 1800 to 1980 RPM. The condenser air flow rate was held at 950 cfm and the evaporator air flow rate was again at 250 cfm with both having air inlet temperatures around 80ºF. The results for these experiments turned out much better than at idle or highway conditions, however this is to be expected as the controllers were all tuned for this condition. In addition, the system was most likely designed to run at city conditions as these are the conditions under which most people drive.
The TXV controller actually does exceptionally well under these conditions. Notice from Figure, that almost all disturbances are rejected, however, at the end of the sequence the amplitude of the fluctuations is starting to increase with each additional step. The only problem, besides that, is the steady state error of 2°C as the controller has a steady state value of 4°C. The remarkable feature is that the superheat only fluctuates about 3°C around its steady state value for the entire test.

Figure 5.2-4 Compressor steps with TXV control at city conditions

Figure 5.2-5 Compressor steps with proportional control at city conditions
The proportional controller also does extremely well. Again notice from Figure 5.2-5 that all of the compressor step disturbances are rejected, only leaving a steady state error of around 3°C. The amplitude of the fluctuations are very similar to the TXV control, ranging from about 2-4°C. The PI controller results can be seen in Figure 5.2-6. The PI controller again does the best, as expected, and keeps the superheat right at 6°C, as desired. The controller does seem to fluctuate slightly more than the TXV and proportional controller, but at a very reasonable range of 4°C.

![Superheat changes with compressor steps with PI control at city conditions](image)

Figure 5.2-6 Compressor Steps with PI control at city conditions

### 5.2.3 Highway Conditions

The compressor steps made at highway conditions are from 3000-3300 RPM. The condenser air flow rate is held at 1250 cfm, with the evaporator air flow rate again at 250 cfm and the inlet air temperatures at 80 °F. The controllers do remarkable well at highway conditions. This is due to the fact the compressor is pretty close to its maximum mass flow and mass flow rate changes are not as significant.

The TXV, as seen in Figure 5.2-7, is remarkably steady as far as the amplitude of the fluctuations, as they are only about 2°C in range. All the disturbances are rejected, with the only problem seeming to be the steady state value. The value seems to start off at around 5°C and finish up at 4°C. This is only an error of 2°C. The proportional control, seen in Figure 5.2-8, is similar with the exception that the fluctuations are about 5°C and the steady state value decreases even more, starting at 7°C and finishing at 4°C. The PI controller is again best with fluctuations about 2-3°C and the steady state being dead on, and can be seen in Figure 5.2-9.
Figure 5.2-7 Compressor steps with TXV control at highway conditions

Figure 5.2-8 Compressor steps with proportional control at highway conditions
5.3 Evaporator Fan Steps

The following set of experiments simulate the automotive passenger turning the fan speed from low to medium to high, etc. The switches in air flow rate are made every 4 minutes in order to allow for the controller to settle to steady state. The air flow rates used for the three setting are 150 cfm, 200 cfm, and 250 cfm. The air flow rates are switched from the lowest setting upwards one step at a time and then back down in reverse order. Flowing this, the air flow rate is changed from low to high and back to show all possible combinations of changes. The entire sequence of steps is done for the TXV, proportional, PI, and dual SISO controllers at idle, city, and highway conditions. Just as in the compressor step changes a set point of 5 °C superheat is used for all of the controllers, and in addition for the dual SISO a set point of 0 °C is used for the evaporator refrigerant inlet temperature, which with computer measurement differences turns out to be -1°C.

5.3.1 Idle Conditions

For the idle condition experiments a compressor speed of 950 RPM was used. In addition, the condenser air flow rate was set to 800 cfm and the air inlet temperature for both the condenser and evaporator were set to 80 °F. The evaporator was switched as described in section 5.3. Just as in the compressor step experiments, the idle condition experiments give the worst results. This is again due to the tuning being done at city conditions and the difficulty of controlling the systems at idle conditions.
Figure 5.3-1 Evaporator fan steps with TXV control at idle conditions

The TXV controller does not do well at all for these conditions. The TXV results can be seen in Figure 5.3-1. The oscillations are much larger than in the case of compressor steps, and also get much worse on the steps when the air flow rate over the evaporator decreases. The most significant problem comes with the steady state values, which vary significantly. They range all the way from 14 ºC down to 1-2ºC on the last step down. These changes in steady state values are much larger than acceptable. In fact, it appears that superheat was lost near the end of the test.

The proportional controller, seen in Figure 5.3-2, does better as far as large changes in steady state values, as it starts around 6 ºC and ends up around 3-4 ºC. The proportional controller also has problems with decreasing steps in air flow rate, especially to the lowest setting. With these steps the amplitude of the oscillations becomes very large, 10 ºC in range. Notice that superheat is definitely lost several times. This is definitely unacceptable, but probably due to the fact that the controller wasn’t tuned for the low fan speed setting, but instead at 250 cfm.
The PI controller shows the exact same problems as the proportional controller in Figure 5.3-3. It does hold the oscillations around the correct steady state value; however the amplitude is again upwards of 10 °C, which is unacceptable.
Evaporator inlet refrigerant temperature changes due to evaporator air flow rate changes with PI control at idle conditions

**Figure 5.3-4** Evaporator fan steps with PI control at idle conditions (Teri)

In addition to the normal plots of just superheat and air flow rate, the evaporator refrigerant inlet temperature changes for the PI controller have been plotted in Figure 5.3-4. This is for comparison purposes to the dual SISO controller, whose results can be seen in Figure 5.3-5. The evaporator refrigerant inlet temperature for the PI controller changes all the way from -3 °C to -9 °C. This is much worse than the dual SISO controller and unacceptable, as the evaporator would frost under humid conditions. The dual SISO controller is able to hold the evaporator refrigerant inlet temperature within 1 °C of the set point. This tight control on the evaporator refrigerant inlet temperature in turn helps the superheat control. The controller has small fluctuations of superheat and always drives the superheat back to the set point of 6 °C.

Superheat changes due to evaporator air flow rate changes with dual SISO control at idle conditions

**Figure 5.3-5** Evaporator fan steps with dual SISO control at idle conditions

**5.3.2 City Conditions**

For the city condition experiments a compressor speed of 1800 RPM was used. In addition, the condenser air flow rate was set to 950 cfm and the air inlet temperature for both the condenser and evaporator was approximately 80 °F. The evaporator was again switched exactly as described in section 5.3.
The TXV controller does much better at city conditions; but is again the worst of the four controllers, as shown in Figure 5.3-6. The maximum amplitude of the oscillations is down to 6°C. The steady state superheat changes from 2°C above the set point to 2°C below. Again the errors and oscillations seem to be worse when the air flow rate steps down to the lowest setting.

Figure 5.3-6 Evaporator fan steps with TXV control at city conditions

The proportional controller, seen in Figure 5.3-7, has problems that are almost identical to the TXV. Most of the disturbances are rejected; however there are still some serious oscillations, which are still better than at idle conditions. Also when the air flow rate is switched to the lowest setting the steady state value starts decreasing from 8°C to 5°C and the oscillations increase in amplitude up to 8°C.

The PI controller does much better than at idle conditions and than the proportional and TXV controllers at city conditions. Its results are shown in Figure 5.3-8. The controller still oscillates; however, it continually drives the superheat towards the steady state of 6°C. The larger steps give it more problems, as would be expected, but the controller recovers quickly.
Superheat changes due to evaporator air flow rate changes with Proportional control at city conditions

![Graph showing superheat changes due to evaporator air flow rate with Proportional control at city conditions.]

Superheat changes due to evaporator air flow rate changes with PI control at city conditions

![Graph showing superheat changes due to evaporator air flow rate with PI control at city conditions.]

Figure 5.3-7 Evaporator fan steps with proportional control at city conditions

Figure 5.3-8 Evaporator fan steps with PI control at city conditions

The evaporator refrigerant inlet temperature is again plotted in a separate plot for comparison purposes to the dual SISO controller in Figure 5.3-9. The evaporator refrigerant inlet temperature again varies over a range of 6ºC. The patterns are very similar to the idle condition experiments, but are 2-3ºC lower. This is again unacceptable because of potential frost accumulation. The dual SISO controller again does exceptionally well, as seen in Figure 5.3-10. The evaporator refrigerant inlet temperature is again held within 1ºC of the set point.
Evaporator inlet refrigerant temperature changes due to evaporator air flow rate changes with PI control at city conditions

Figure 5.3-9 Evaporator fan steps with PI control at city conditions (Teri)

Superheat changes due to evaporator air flow rate changes with dual SISO control at city conditions

Figure 5.3-10 Evaporator fan steps with dual SISO control at city conditions

Similar to the idle speed experiments, the dual SISO controller benefits from the virtually fixed evaporator refrigerant inlet temperature, and drives the superheat towards its set point in a considerably smoother fashion, leading it to be the best controller. It is interesting to note the difference in settling time between the dual SISO controller and the PI controller occurring after the step from low to high fan speed. The settling time shows that, even though the dual SISO controller gains benefits from controlling the evaporator refrigerant inlet temperature, the dynamics are in fact coupled. This can be seen in that the settling time has increased with the dual SISO after this disturbance step, whereas the superheat transient for the PI controller settles in about the same length of time after every step disturbance.

5.3.3 Highway Conditions

The highway condition experiments were run at a compressor speed of 3000 RPM. In addition, the condenser air flow rate was set to 1250 cfm and the air inlet temperature for both the condenser and evaporator was
approximately 80 °F. The evaporator was switched exactly as described in section 5.3. The results of the highway condition experiments are in general a little worse than the city condition experiments, possibly due to tuning.

The TXV controller has similar oscillations of superheat at highway conditions as it did at city conditions, as shown in Figure 5.3-11. There is again a similar downward trend of the steady state value of superheat. However, this time the superheat starts out at the set point of 6°C, but the controller loses superheat with the last step of the air flow rate. The controller is starting to climb back from losing superheat, but it seems to be struggling. A possible cause for this struggle is that the evaporator could be starting to frost, causing the heat transfer rate to decrease. This would cause the control signal to be incorrect as it is just a proportional factor off of superheat.

![Superheat changes due to evaporator air flow rate changes with TXV control at highway conditions](image)

Figure 5.3-11 Evaporator fan steps with TXV control at highway conditions

The proportional controller shows similar trends, see Figure 5.3-12, but it starts out with a steady state value of 9°C and finish around the set point. The controller again handles the upward steps in air flow rates extremely well. With the step to the lowest air flow rate the steady state value starts to decrease and the amplitude of the oscillations starts to increase, but not to an extreme amount.
Figure 5.3-12 Evaporator fan steps with proportional control at highway conditions

The results for the PI controller are shown in Figure 5.3-13. The PI controller does very well at the highway conditions. The controller is always driving to the set point of 6°C in a reasonable quick fashion. The controller still has some problems with the large steps, but recovers the superheat and lowers the amplitude of the oscillations at the same time.

Figure 5.3-13 Evaporator fan steps with PI control at highway conditions

Additionally the evaporator refrigerant inlet temperature for the PI controller is plotted in Figure 5.3-14. Notice that the evaporator refrigerant inlet temperature is very low. This could cause frost to form on the
evaporator. While the PI control could integrate this problem out, the TXV and proportional controller would have problems, as they did in the plots shown above. The evaporator refrigerant inlet temperature of the PI controller follows the same pattern as in the idle and city condition experiments. The only difference is that the pattern has been lowered even more. The dual SISO controller manages the evaporator temperature extremely well as shown in Figure 5.3-15. The evaporator refrigerant inlet temperature again only gets about 1°C away from the set point. The superheat follows the PI controller’s trends, with fewer oscillations, which are also lower in amplitude. Again the dual SISO also shows effects of coupled dynamics with slower settling time, especially after the step from low to high fan speed.

![Evaporator inlet refrigerant temperature changes due to evaporator air flow rate changes with PI control at highway conditions](image)

Figure 5.3-14 Evaporator fan steps with PI control at highway conditions (Teri)

![Superheat changes due to evaporator air flow rate changes with dual SISO control at highway conditions](image)

Figure 5.3-15 Evaporator fan steps with dual SISO control at highway conditions

### 5.4 Full Condition Steps

The experimental results in this section showcase what happens when a car undergoes changes from one set of driving conditions to another set of driving conditions. For the superheat controllers the compressor speed and
condenser air flow rate were stepped from idle to city to highway and back down. Each one of the steps was held for 5 minutes before the next one was made. The compressor speed steps were 950 RPM, 1800 RPM, and 3000 RPM. The condenser air flow rate steps were 800 cfm, 950 cfm, and 1250 cfm. The evaporator air flow rate was kept at a constant flow rate of 250 cfm, and both air inlet temperatures were around 80°F. For the dual SISO controller only the condenser air flow rate was stepped as the controller has control over the compressor speed. To show the condition steps, RPM was chosen, as it is easiest to get a feel for what is happening. However, for the dual SISO controller the steps are shown with the changes in condenser air flow rate. The same condenser air flow rate steps were made in all the experiments, it is just easier to understand what is happening with the RPM values.

The TXV controller, shown in Figure 5.4-1, does an intermediate job of controlling the superheat on the upwards steps. The only major problem on the upward steps is the large overshoot. The controller really runs into problems on the steps back down. On the first step down, it almost loses superheat, but maintains it and through pretty large oscillations drives the superheat back to its steady state value of around 8°C, which is off by 2°C. The last step really causes problems as the controller completely loses superheat. After 2.5 minutes, it finally has started to get superheat back.

Figure 5.4-1 Condition steps with TXV control
Figure 5.4-2 Condition step with proportional control

As seen in Figure 5.4-2, the proportional controller definitely does better than the TXV, but it still has some problems. The steps up still cause a good bit of overshoot. Unlike the TXV, the proportional controller controls the first step down fairly well; however, on the second step it also loses superheat. The proportional controller is able to regain superheat quickly and is in the process of ramping back towards steady state. It is again interesting to note how much more trouble these controllers have at idle conditions. Even though the controllers are tuned at city conditions, they have had simple preset scheduling done as discussed in section 4.1

Figure 5.4-3 Condition steps with PI control
The PI controller makes definite improvements over the TXV and proportional controllers. The results for the PI controller can be seen in Figure 5.4-3. The controller is always driving the superheat back to 6°C. Even after losing superheat at the last step it fairly quickly is back at the set point. In addition to the superheat plot, a plot of the evaporator refrigerant inlet temperature was again made for the PI controller and is shown in Figure 5.4-4. The evaporator refrigerant inlet temperature goes through a range of 5°C. This is better than seen with the evaporator step changes, but is still much worse than with dual SISO controller.

Evaporator inlet refrigerant temperature changes due to condition steps with PI control

Figure 5.4-4 Condition steps with PI control (Teri)

Superheat changes due to condition steps with dual SISO control

Figure 5.4-5 Condition steps with dual SISO control

The controller that produced the best results for these experiments is by far the dual SISO controller, shown in Figure 5.4-5. With the added help of controlling the compressor speed, it is able to keep evaporator refrigerant inlet temperature at the set point for the entire experiment. The superheat also stays at the set point throughout the experiment, even though it seems the oscillations change some with each new step.
5.5 Start Up Response

The start up response experiments shown in this section represent what the systems goes through when the air conditioning system is switched on while driving at city conditions. The system was first allowed to come to rest with the sufficient time for the refrigerant to redistribute and warm up to room temperatures. The condenser air flow rate was set at 950 cfm and the evaporator air flow rate was set at 250 cfm. Both air inlet temperatures were set to 80°F. For the TXV, proportional, and PI controllers the compressor was set to 1800 RPM, however this is still free to control for the dual SISO controller.

![Superheat changes during startup with TXV control at city conditions](image)

Figure 5.5-1 Start up response with TXV control

The TXV again does the worst of the controllers, as shown by Figure 5.5-1. The overshoot is the largest reaching upwards of 20°C. The controller also seems to be reaching a steady state value, which is much higher than normally seen, until it completely looses superheat. The TXV is then able to regain superheat and through one more peak in overshoot, settles at its final steady state value of 8°C. The whole process takes upwards of four and a half minutes to complete.

The results for the proportional controller are shown in Figure 5.5-2. The proportional controller does much better than the TXV. It again shows the dip seen after the controller starts decreasing the superheat at a slower rate. The proportional controller, however, rejects this and stays at its superheat of about 9-10°C. The process to get to steady state takes about two and two-third minutes. The overshoot for the proportional controller has decreased down to around 18-19°C.
The PI controller again makes improvements in start up control over the last two controllers. Its results are shown in Figure 5.5-3. The largest improvement being that the superheat is driven directly to the set point value of 6°C. The settling time is slightly reduced from the proportional controller to just over two and a half minutes. However, it is interesting to note the long period before superheat starts to increase and the spike seen in the upward slope of the superheat. Anti-windup techniques were not used for the integral term and it is possible that it is having some effect in the start up response. The overshoot is again around 18-19°C.
The dual SISO controller does by far the best of the controllers. For the dual SISO controller the evaporator refrigerant inlet temperature is driven to the set point with only slight overshoot in about a minute. In comparison the PI controller’s evaporator refrigerant inlet temperature, shown in Figure 5.5-4, takes much longer to settle out, and at that settles at -6°C. With the added help of having the evaporator refrigerant inlet temperature steady at its set point, the dual SISO controller is able to settle out to the superheat set point of 6°C under 2 minutes, almost a full minute faster than the PI controller. The dual SISO results can be found in Figure 5.5-5.

Figure 5.5-4 Start up response with PI control (Teri)

Figure 5.5-5 Start up response with dual SISO control
5.6 Humidity Addition Response

The humidity addition experiments discussed in the following sections were run to simulate the effects of a door or window of an automobile being opened. In this case the relative humidity would increase significantly. In addition, it is very interesting to see how the system behaves as it pulls the humidity back down, i.e. the door or window has been closed again. For this reason, the system was allowed to come to steady state and then the humidity was added.

For the humidity addition experiments, the humidity addition system was turned on full blast until the relative humidity at the inlet of the evaporator reached approximately 80 percent. At this point the humidity addition system was turned completely off and the system was allowed to come back towards steady state. The humidity was added in this fashion at idle, city, and highway conditions. During all the experiments the evaporator air flow rate was held constant at 250 cfm and both inlet air temperatures were held at 80ºC.

1.1.1 Idle Conditions

For the idle condition experiments, the humidity was added as described in section 5.6. The compressor speed was held at a constant 950 RPM and the condenser at a constant 800 cfm. Of course, for the dual SISO controller the compressor speed varied as directed by the controller. As expected, the TXV controller does the worst and each step up in controller complexity does better. The results for these experiments are shown in Figure 5.6-1, Figure 5.6-2, Figure 5.6-3, Figure 5.6-4, and Figure 5.6-5.

![Superheat changes due to humidity addition with TXV control at idle conditions](image-url)

Figure 5.6-1 Humidity addition with TXV control at idle conditions
Superheat changes due to humidity addition with Proportional control at idle conditions

The TXV controller has the largest overshoot of superheat as it rises all the way up to 19°C. It is however comparable in recovery time to the proportional controller. The controllers decrease the superheat back to steady state at an approximately equal rate. It takes both of them over 4 minutes to get back close to steady state. The proportional controller does better in overshoot than the TXV as it only rises to just shy of 16°C. The PI controller does much better in handling the overshoot, as it only peaks up to just over 14°C. In addition its settling time is significantly better. It only takes the PI controller a little over two minutes to settle out. Again the evaporator refrigerant inlet temperature results are shown for the PI controller for comparison purposes with the dual SISO controller. The evaporator refrigerant inlet temperature changes about 8°C for the PI controller, while the dual SISO controller does much better only changing about 2°C. The dual SISO controller takes almost exactly the same amount of time as the PI controller to reach steady state.

Figure 5.6-2 Humidity addition with proportional control at idle conditions
Superheat changes due to humidity addition
with PI control at idle conditions

Figure 5.6-3 Humidity addition with PI control at idle conditions

Evaporator inlet refrigerant temperature changes due to
humidity addition with PI control at idle conditions

Figure 5.6-4 Humidity addition with PI control at idle conditions (Teri)
5.6.2 City Conditions

Similarly, for the city condition experiments, the humidity was added as described in section 5.6. The compressor speed was this time held at a constant 1800 RPM with the condenser at a constant 950 cfm. All of the controllers handle the humidity addition better at city conditions than at idle conditions. The TXV still has the poorest performance, with the PI and dual SISO producing similar responses. The results are shown in Figure 5.6-6, Figure 5.6-7, Figure 5.6-8, Figure 5.6-9, and Figure 5.6-10.
Superheat changes due to humidity addition with Proportional control at city conditions

Figure 5.6-7 Humidity addition with proportional control at city conditions

The TXV shows improvement in overshoot from idle conditions as it only climbs up to 16°C. It still needs over 4 minutes to settle. The proportional controller has very similar results as the TXV, even though at city conditions the proportional controller seems to do slightly better than the TXV. The proportional controller settles in just under the settling time of the TXV and it on reaches a peak of just under 14°C. The PI controller again has remarkably better performance than the TXV and proportional controllers. The PI controller overshoots to 12°C and has a settling time of around 2 minutes. The PI controller does allow a change in evaporator refrigerant inlet temperature of 5°C. The dual SISO controller actually does worse in overshoot, as it reaches 15°C. However it is able to keep evaporator refrigerant inlet temperature from fluctuating more than 2°C. The dual SISO controller has approximately the same settling time as the PI controller.
Superheat changes due to humidity addition with PI control at city conditions

Figure 5.6-8 Humidity addition with PI control at city conditions

Evaporator inlet refrigerant temperature changes due to humidity addition with PI control at city conditions

Figure 5.6-9 Humidity addition with PI control at city conditions (Teri)
5.6.3 Highway Conditions

For the highway condition experiments, the humidity was again added in the same fashion as the other experiments. This time though, compressor speed was held at a constant 3000 RPM and the condenser at a constant 1250 cfm. All comparative results are virtually the same for the highway conditions as the city conditions and are shown in Figure 5.6-11, Figure 5.6-12, Figure 5.6-13, Figure 5.6-14, and Figure 5.6-15. The TXV again is worst, followed by the proportional controller. The overshoot for the TXV and for the proportional controllers are actually slightly higher than at city, at 16°C and 14°C respectively. The settling times for the TXV and proportional controllers are just slightly less than at city conditions, however the trends are exactly the same. The PI controller and dual SISO controller have the same overshoots, settling times, and variations at city and highway. The only noticeable difference being that the evaporator refrigerant inlet temperature changes for the PI controller are shifted down 2°C.
Superheat changes due to humidity addition with TXV control at highway conditions

Figure 5.6-11 Humidity addition with TXV control at highway conditions

Superheat changes due to humidity addition with Proportional control at highway conditions

Figure 5.6-12 Humidity addition with proportional control at highway conditions
Superheat changes due to humidity addition with PI control at highway conditions

Figure 5.6-13 Humidity addition with PI control at highway conditions

Evaporator inlet refrigerant temperature changes due to humidity addition with PI control at highway conditions

Figure 5.6-14 Humidity addition with PI control at highway conditions (Teri)
5.7 Power Consumption

Several experiments were run to verify the strategies for reducing power consumption outlined in Section 1.3. In order to eliminate the effects of a somewhat varying sample rate, the work was calculated for a set 23.5 minute experiment. The work was calculated by integrating power over a set length of time. The power was calculated using the compressor torque and shaft speed. Work was then calculated by multiplying power by the sample rate at that instant and adding up the individual pieces to get the total work for the 23.5 minute span. The work for this time span was calculated for several steady state experiments. With each new experiment either the superheat levels or evaporator refrigerant inlet temperatures set points were changed. The results for these experiments have been summarized in Figure 5.7-1.

There are several interesting trends to note from Figure 5.7-1. The first being that the fill of the evaporator makes a difference in the amount of work required to run the experiment. Noticing from first two experiments (shown by the two bars on the far left of the figure) decreasing superheat, which increase the evaporator filling, decreases work. The important (and expected) trend is that increasing the evaporator refrigerant inlet temperature reduces power consumption. These experiments were run at a slightly higher humidity for two reasons. The first being to look at the overall trend that evaporator load has on work, and the second being to allow the compressor to be run at a higher speed for the set point evaporator refrigerant inlet temperature. The results are seen in experiments 3-5 in Figure 5.7-1. This result is exactly as expected and is discussed in Forrest and Bhatti [2002]. Finally the last experiment shows that increasing the load on the evaporator increases the power consumption.
For raw comparisons only, the work needed to run the evaporator air flow rate experiments, shown in Section 5.3, was also calculated. The controllers picked for the work comparison were the PI controller and dual SISO controller. The work was calculated from the start of the first step for 23.5 minutes, in the same fashion as above. These results are shown in Figure 5.7-2. This is not truly a fair comparison, as the evaporator refrigerant inlet temperature for the PI controller drops below acceptable levels. Normally if this were to happen the compressor would be clutched off. However, the startup which would ensue would create more work than just running the evaporator inlet temperature at a lower temperature. A key point to note is that while the PI controller’s work fluctuates with conditions, the dual SISO is able to keep the work relatively constant over all conditions. This is exactly what is wanted for a regulator designed for this type of power reduction strategy.
Figure 5.7-2 Power Consumption Savings
6. Conclusions and Recommendations

Presented in this thesis is a comparison of several types of control for automotive air conditioning systems. These controllers varied from the widely used TXV to a dual SISO controller. Comparing just the superheat controllers, the PI controller does by far the best in all aspects of control. However the PI controller is not the best controller when looking at the ultimate goal of reducing power consumption. This is where the dual SISO controller takes the lead. The dual SISO controller’s ability to raise the evaporator refrigerant inlet temperature allows adjustment of the power required to drive the cooling system to the minimum needed to meet the cooling requirements.

With the dual SISO controller’s ability to keep control of both the evaporator refrigerant inlet temperature and superheat it makes it possible to implement the strategy discussed in the motivations section (1.3). The main idea of this strategy is to use the air conditioner as a regulator set to minimize power consumption and then regulate the final temperature with the reheat section. The amount of reheat in this strategy would be much less than traditional systems. Forrest and Bhatti [2002] discussed a similar strategy that showed saving up to almost 25% during actual tests. This amount of savings is not a trivial amount and could be implemented rather simply. The dual SISO controller fits into the role of regulating the system and ensuring that the evaporator refrigerant inlet temperature is up to the desired level, as well as making full use of the evaporator. With this increase in evaporator refrigerant inlet temperature less reheat is needed and energy is saved. The harder to control and slower responding air conditioner can be set at a base cooling level and the climate control system can provide cool, dry air by regulating mixing dampers (or air movers) to get the correct temperature to the passenger compartment. Controlling a damper, with the air conditioner fixed is a much easier task than trying to change the set point of the vapor compression system.

Even though the dual SISO control shows potential for a large amount of energy savings, there are some possibilities to reduce the power consumption even further. One method is keeping even tighter controller and having the air conditioner reach set points at a faster rate. A method to do this was discussed in section 4.5. In this section a MIMO controller is presented that is designed with standard methods. Even though this particular MIMO controller does not work, a MIMO controller could possibly give control improvements. A MIMO controller, by using the coupled effects seen in air conditioners, could improve settling times and therefore arrive at new set points or reject disturbances at a faster rate. More complex strategies for design and use of a MIMO controller would be needed to make this work and increase the power consumption savings.
Bibliography

Chappell, J., C. Cusano, P. Hrnjak, N. Miller, and T. Newell, 2000: “Measurement of the Relationship Between Oil Circulation and Compressor Lubrication in a Mobile A/C System: Part Two.” Department of Mechanical and Industrial Engineering University of Illinois at Urbana-Champaign, ACRC TR-169

Drozdek, J., C. Cusano, P. Hrnjak, N. Miller, and T. Newell, 2000: “Measurement of the Relationship Between Oil Circulation and Compressor Lubrication in a Mobile A/C System: Part One.” Department of Mechanical and Industrial Engineering University of Illinois at Urbana-Champaign, ACRC TR-169


Solberg, J., P. Hrnjak, and N. Miller 1998: “Regulation of the Liquid-Mass-Fraction of the Refrigerant Exiting an Evaporator.” Department of Mechanical and Industrial Engineering University of Illinois at Urbana-Champaign, Masters Thesis

Weston, P., W. Dunn, and N. Miller, 1996: “Design and Construction of Mobile Air Conditioning Test Facility for Transient Studies.” Department of Mechanical and Industrial Engineering University of Illinois at Urbana-Champaign, ACRC TR-97

Whitchurch, M., W. Dunn, and N. Miller, 1997: “Humidity Effects in Mobile Air-Conditioning Systems.” Department of Mechanical and Industrial Engineering University of Illinois at Urbana-Champaign, ACRC TR-126

Zhang, R., and A. Alleyne. The University of Illinois at Urbana-Champaign. http://mr-roboto.me.uiuc.edu/evps/index.html
Appendix A: Modification Details

The modifications to the refrigerant loop included removing an oil separator, which was right after the compressor, but had been routed around with valves. In addition, the oil sampling section after the condenser was removed. The oil sampling section was replaced with the proper tubing and adapters, as well as a receiver. The third item changed was the liquid line venturi, which was removed for use in another lab and replaced with the correct tubing and fittings. The new schematic drawing can be seen in Figure 2.2-1.

For the air side loops additional sections were added to help with air mixing and the pressure drop associated with the ducting contracting and expanding again for the air side temperature measurement before the blower. On the evaporator side a section of ducting the same dimensions as the existing ducting was added to help the air mix fully before it reaches the thermocouple grid. The drawing of the section can be seen below. On the condenser side two sections were added. One was added to increase the length between the condenser and the thermocouple grid. In addition a new nozzle was made to help with the pressure drop that is seen when ducting is contracted and expanded again. Both sections can be seen below.

Additional Evaporator Section:
**Additional Condenser Section:**

Material: 22 gage galvanized sheet metal
All units inches

**Condenser Nozzle:**

Material: 22 gage galvanized sheet metal
All units inches
Appendix B: Wiring Diagrams and Details

Control Wiring Diagram:
Signal Conditioner Wiring Diagram:
Air Flow Rate Inverters Wiring Diagrams:

8.4.3 Frequency Setting Signal (A-G-R-B-I, P, F, P2, F-P2, d G-Sb):

(1) RR terminal frequency setting signal characteristics

The characteristics of the frequency setting signals input to the RR terminal and output frequency signals are as shown in the figure below.

- Adjustment of the RR analog terminal input
  - At the standard default setting, the adjustment has been given an allowance to output the inverter output for the first time with a slight voltage applied to the RR terminal.
  - To determine the allowance, increase the r - r - G, value. If the value is too large, the inverter output will be output even when 0V is input.

- Adjustment of the RR input terminal gain (r - r - G-R-B-I)
  - At the standard default setting, the maximum frequency will be reached when the RR input is slightly below the upper limit voltage (this can be set at 10V/5V with the jumper pin). To set so that the maximum frequency is reached with the upper limit voltage, lower the r - r - G value. However, if this is lowered too far, the maximum frequency will not be output even when the upper limit voltage is input.

- To adjust the RR input terminal so that 0V corresponds to an inverter output of 0Hz, set r - r - G - b - 0 (a change of one corresponds to about 0.001 Hz).

- To adjust the RR input terminal so that the maximum output voltage corresponds to the maximum inverter output frequency, set r - r - G - 0 (a change of one corresponds to about 0.001 Hz).

- To adjust, adjust the r - r - G value first, and then adjust r - r - G.

- Setting of the RR terminal frequency setting signal characteristics (built-in gain)
  - If the r - r - G function is used, the RR terminal frequency setting signal characteristics should be decreased in proportion.

---

8.4 Frequency Setting Signals when set with external signals

The "faster frequency" is controlled externally using the RR, IV, CC and CU terminals on the control circuit terminal block shown in Fig. 9-13.

8.4.1 Types of frequency setting signals

The frequency setting signals are changed over with the jumper J1 and J2 on the terminal PCB. The jumper positions and combinations of parameters (L, L, L, L, L, L, L, L, L, L, L, L) and each function of the parameters are shown in Fig. 8-14.

8.4.2 RR terminal input priority (r - r - C, r - r - L, r - r - D, G-Sb)

The frequency setting signals input from the terminal block can be changed over.

---

Table 8-2: Typical emergency stop and operations

<table>
<thead>
<tr>
<th>Setting of ESE</th>
<th>Function</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Coarsen stop. (memory E stop)</td>
</tr>
<tr>
<td>1</td>
<td>E stop after deceleration stop</td>
</tr>
<tr>
<td>2</td>
<td>Min DC braking stop when set at E stop, and voltage set at r - r - b, the inverter will E stop</td>
</tr>
</tbody>
</table>

---

Fig. 9-17 Frequency setting signal characteristics (50 Hz)

- Separate point 1 and 2 by at least 10%. The r - r - L error will be displayed when point 1 and 2 are close than 10%.

---

Fig. 8-14 Changeover of frequency setting signals and each function
Compressor Inverter Wiring Diagram:

Fig. 5 shows the connection diagram of VS-616GII for operation by external signals.

Notes:
1. Indicates shielded leads and twisted-pair shielded leads.
2. External terminal 13 of +15V has maximum output current capacity of 20mA.
3. Either external terminal 13 or 14 can be used.
4. Terminal symbols: ◆ shows main circuit; ○ shows control circuit.
HPVEE Name Scheme:
Zone Box Diagram Naming Conventions

HP DAS Inputs
HP VXI
(#) ##

Multiplexer ______ Terminal
Number ______ Number
Condenser Loop Box Diagram:

![Diagram showing various components and connections for a condenser loop box.](image-url)
Evaporator Loop Box Diagram:
Refrigerant Loop Box Diagram:
### HP VEE Wiring Descriptions:

<table>
<thead>
<tr>
<th>Signal Name</th>
<th>Description (Listed is voltage signal with equations can produce following)</th>
<th>Bundle-Wire</th>
<th>Mux-Term</th>
<th>VXI Channel</th>
</tr>
</thead>
<tbody>
<tr>
<td>V Teri</td>
<td>Evaporator Refrigerant Inlet Temperature</td>
<td>R-1</td>
<td>1-0</td>
<td>0</td>
</tr>
<tr>
<td>V Tero</td>
<td>Evaporator Refrigerant Outlet Temperature</td>
<td>R-2</td>
<td>1-1</td>
<td>1</td>
</tr>
<tr>
<td>V Tkri</td>
<td>Compressor Refrigerant Inlet Temperature</td>
<td>R-5</td>
<td>1-2</td>
<td>2</td>
</tr>
<tr>
<td>V Tkro</td>
<td>Compressor Refrigerant Outlet Temperature</td>
<td>R-6</td>
<td>1-3</td>
<td>3</td>
</tr>
<tr>
<td>V Tcri</td>
<td>Condenser Refrigerant Inlet Temperature</td>
<td>R-3</td>
<td>1-4</td>
<td>4</td>
</tr>
<tr>
<td>V Tcro</td>
<td>Condenser Refrigerant Outlet Temperature</td>
<td>R-4</td>
<td>1-5</td>
<td>5</td>
</tr>
<tr>
<td>V Tl v</td>
<td>Liquid-line VFT Inlet Refrigerant Temperature</td>
<td>R-7</td>
<td>1-6</td>
<td>6</td>
</tr>
<tr>
<td>V Teai</td>
<td>Evaporator Air Inlet Temperature</td>
<td>E-1</td>
<td>1-7</td>
<td>7</td>
</tr>
<tr>
<td>V Teao</td>
<td>Evaporator Air Outlet Temperature</td>
<td>E-2</td>
<td>1-8</td>
<td>8</td>
</tr>
<tr>
<td>V Teav</td>
<td>Evaporator Air Loop VFT Inlet Temperature</td>
<td>E-3</td>
<td>1-9</td>
<td>9</td>
</tr>
<tr>
<td>V Tcai</td>
<td>Condenser Air Inlet Temperature</td>
<td>C-1</td>
<td>1-10</td>
<td>10</td>
</tr>
<tr>
<td>V Tcao</td>
<td>Condenser Air Outlet Temperature</td>
<td>C-2</td>
<td>1-11</td>
<td>11</td>
</tr>
<tr>
<td>V Tcav</td>
<td>Condenser Air Loop VFT Inlet Temperature</td>
<td>C-3</td>
<td>1-12</td>
<td>12</td>
</tr>
<tr>
<td>V Toil</td>
<td>Refrigerant/Oil Temperature at Oil Concentration Sensor</td>
<td>R-8</td>
<td>1-13</td>
<td>13</td>
</tr>
<tr>
<td>V dPdv</td>
<td>Discharge-line VFT Pressure Differential</td>
<td>R-16</td>
<td>2-0</td>
<td>16</td>
</tr>
<tr>
<td>V dPlv</td>
<td>Liquid-line VFT Pressure Differential</td>
<td>R-17</td>
<td>2-1</td>
<td>17</td>
</tr>
<tr>
<td>V Micro</td>
<td>MicroMotion mass flow rate</td>
<td>C-6</td>
<td>2-2</td>
<td>18</td>
</tr>
<tr>
<td>V dPeav</td>
<td>Evaporator Air Loop VFT Pressure Differential</td>
<td>E-5</td>
<td>2-3</td>
<td>19</td>
</tr>
<tr>
<td>V dPcav</td>
<td>Condenser Air Loop VFT Pressure Differential</td>
<td>C-5</td>
<td>2-4</td>
<td>20</td>
</tr>
<tr>
<td>V TH2O</td>
<td>Temperature in H2O for RTD probes from PID Controller</td>
<td>E-6</td>
<td>2-5</td>
<td>21</td>
</tr>
<tr>
<td>V Oil</td>
<td>Percentage of Oil at oil sensor ?????</td>
<td>C-11</td>
<td>2-6</td>
<td>22</td>
</tr>
<tr>
<td>V RTDea i</td>
<td>Evaporator Air Inlet Humidity Probe Temperature</td>
<td>E-7</td>
<td>2-7</td>
<td>23</td>
</tr>
<tr>
<td>V RTDeao</td>
<td>Evaporator Air Outlet Humidity Probe Temperature</td>
<td>E-10</td>
<td>2-8</td>
<td>24</td>
</tr>
<tr>
<td>V RTDcao</td>
<td>Condenser Air Outlet Humidity Probe Temperature</td>
<td>C-8</td>
<td>2-9</td>
<td>25</td>
</tr>
<tr>
<td>V RHeai</td>
<td>Evaporator Air Inlet Relative Humidity</td>
<td>E-8</td>
<td>2-10</td>
<td>26</td>
</tr>
<tr>
<td>V RHeao</td>
<td>Evaporator Air Outlet Relative Humidity</td>
<td>E-11</td>
<td>2-11</td>
<td>27</td>
</tr>
<tr>
<td>V RHcao</td>
<td>Condenser Air Outlet Relative Humidity</td>
<td>C-9</td>
<td>2-12</td>
<td>28</td>
</tr>
<tr>
<td>V Tmstr</td>
<td>Thermistor, located in constant temperature bath</td>
<td>R-19</td>
<td>2-13</td>
<td>29</td>
</tr>
<tr>
<td>V dPer</td>
<td>Evaporator Refrigerant Pressure Drop</td>
<td>R-14</td>
<td>2-14</td>
<td>30</td>
</tr>
<tr>
<td>V Pero</td>
<td>Evaporator Refrigerant Outlet Pressure</td>
<td>R-9</td>
<td>2-15</td>
<td>31</td>
</tr>
<tr>
<td>V Pkri</td>
<td>Compressor Refrigerant Inlet Pressure</td>
<td>R-11</td>
<td>2-16</td>
<td>32</td>
</tr>
<tr>
<td>V Pkro</td>
<td>Compressor Refrigerant Outlet Pressure</td>
<td>R-12</td>
<td>2-17</td>
<td>33</td>
</tr>
<tr>
<td>V Pcr i</td>
<td>Condenser Refrigerant Inlet Pressure</td>
<td>R-10</td>
<td>2-18</td>
<td>34</td>
</tr>
<tr>
<td>V dPcr</td>
<td>Condenser Refrigerant Pressure Drop</td>
<td>R-15</td>
<td>2-19</td>
<td>35</td>
</tr>
<tr>
<td>V Plv</td>
<td>Liquid-Line VFT Inlet Refrigerant Pressure</td>
<td>R-13</td>
<td>2-20</td>
<td>36</td>
</tr>
<tr>
<td>V Peav</td>
<td>Evaporator Air Loop VFT Inlet Pressure</td>
<td>E-4</td>
<td>2-21</td>
<td>37</td>
</tr>
<tr>
<td>V Pcav</td>
<td>Condenser Air Loop VFT Inlet Pressure</td>
<td>C-4</td>
<td>2-22</td>
<td>38</td>
</tr>
<tr>
<td>V RPM</td>
<td>Compressor Speed</td>
<td>E-12</td>
<td>2-23</td>
<td>39</td>
</tr>
<tr>
<td>V torque</td>
<td>Compressor Torque</td>
<td>E-13</td>
<td>2-24</td>
<td>40</td>
</tr>
<tr>
<td>V 5V</td>
<td>5 Volt input for comparison basis ?????</td>
<td>R-18</td>
<td>2-25</td>
<td>41</td>
</tr>
<tr>
<td>V Somat</td>
<td>Somat input (we can change to valve input voltage)</td>
<td>-</td>
<td>2-26</td>
<td>42</td>
</tr>
<tr>
<td>Sensor Old</td>
<td></td>
<td></td>
<td></td>
<td>43</td>
</tr>
<tr>
<td>Sensor New</td>
<td></td>
<td></td>
<td></td>
<td>44</td>
</tr>
<tr>
<td>Oil Return Micro Density</td>
<td></td>
<td></td>
<td></td>
<td>45</td>
</tr>
<tr>
<td>Oil Return Micro m dot</td>
<td></td>
<td></td>
<td></td>
<td>46</td>
</tr>
</tbody>
</table>
Appendix C: Test Loop Operating Procedure

1. Make sure all the refrigerant has been evacuated and open the system to the atmosphere with the red refrigerant transfer hoses connected to the access valve closest to the compressor inlet.

2. Disconnect the airside venturi pressure transducer air hoses at the pressure transducer side. (They are attached with quick disconnects)

3. Turn on the following equipment:
   - Condenser blower disconnect switch
   - Sporlan valve control circuit box
   - Sporlan valve voltage supply
   - Instrumentation power supply
   - Evaporator blower disconnect switch
   - Evaporator heater disconnect switch
   - Evaporator heater control circuit box
   - Compressor motor disconnect switch
   - Compressor clutch control circuit box

4. Open HP VEE program C:\Marten\VEE Programs\test2.vee. Make sure that the File Output Name reads C:\Marten\Data\Test1.txt. Press the run button and allow the program to collect data for at least 600 samples (can keep track of on the graphs).

5. Take pressure reading from barometer on sidewall and record in lab book. (Conversion from mmHg to Pa is as follows: Pa = (X mmHg)(13.55)(9.81)).

6. Open Excel file C:\Marten\Data\Calibration.xls and Enable Macros when prompted. Enter in the zero as the atmospheric pressure in the space labeled Omega Pressure Reading. Press the “Get New Calibration Data” button and save the Calibration file as Calibration **-**-**.xls with the blanks being the current date.

7. Enter the new values for the air side venturies into the calculation section in the HP VEE as these tend to change the most from day to day. (Optional)

8. Plug in the air lines to the airside venturi pressure transducers.

9. Hook up the red hose to the middle inlet of the refrigerant transfer pressure gauges. Hook up the yellow hose to the vacuum pump and the left side of the refrigerant transfer pressure gauges. Hook up the blue hose to the liquid side of the tank of clean R134a and the right side of the refrigerant transfer pressure gauges.

10. With the R134a tank closed and the two valves on the refrigerant transfer pressure gauge open, turn on the vacuum pump for several minutes to pump out all of the air out of the system as well as the hoses.

11. Close both the refrigerant transfer pressure gauge valves and then turn off the vacuum pump.

12. Open the liquid refrigerant valve so it just fills the hose from the tank to the refrigerant transfer pressure gauges. Set the tank on the scale and zero the scale.

13. Open the right valve of the refrigerant transfer pressure gauges and allow the system to fill as close as possible to 3lbs and 9oz, closing the refrigerant transfer pressure gauge valve to stop flow first. Record the exact weight in lab book.

14. Press run on the evaporator and condenser blower inverters.

15. If the system doesn’t fill complete to 3lbs 9 oz, turn on the compressor with the valve open and allow it to suck in the extra need refrigerant.
16. Disconnect the red hose from the access valve and put on its cap.

17. Run the system according to the test plan, i.e. compile the desired MatLab controller file and press start on the Wincon tool box to start the program and on the HP VEE program to start data acquisition.

18. After running experiments, hook up the recover pump to the dirty tank (air and oil) and to the access valve using the hoses from earlier. Pump out all of the refrigerant in the system.

19. Turn off all of the equipment listed earlier in step 3.