Mixed Phase Compression High Efficiency Heat Pump

by

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A thesis
presented to the University of Waterloo
in fulfilment of the
thesis requirement for the degree of
Master of Applied Science
in
Mechanical Engineering

Waterloo, Ontario, Canada, 2014

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AUTHOR'S DECLARATION

I hereby declare that I am the sole author of this thesis. This is a true copy of the thesis, including any required final revisions, as accepted by my examiners.

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ABSTRACT

The objective of this thesis is the design and realization of a higher efficiency air source heat pump. The improved pump's operating cost must rival the cost of heating with natural gas, while incurring a minimal increase in the capital cost of the pump. A COP greater than 4 at –15C ambient is needed to achieve this goal. During winter season testing a COP of 4.25 +/- 0.11 was observed. This tracks well with a predicted COP of 4.4 and also against a commercial system with a claimed COP between 2.1 and 3.0.

ACKNOWLEDGEMENTS

I would like to acknowledge Roydon Fraser for supervision and guidance of this project, Ying Teo for helping with the conversion of the drawings to an electronic format, Eastern Refrigeration for allowing me to make small volume purchases, Copeland compressors for providing engineering data on their compressors and Sporlan for publishing application notes on their components.

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LIST OF ABBREVIATIONS

AUX - auxiliary
BTU - British thermal unit
Btu/hr - British thermal unit per hour
COP - Coefficient of performance
CPH - cycle per hour
DC - direct current
DC/DC- DC to DC power converter
DCR - DC resistance
EEV - electronic expansion valve
EMI - electromagnetic interference
FLA - full load Ampers
HFC - hydrofluorocarbon
HP - heat pump
HVAC - heating ventilation and air conditioning
LI - liquid injection
MODP – minimum operating differential pressure
MOQ - minimum order quantity
POE - polyol-ester
PTC - positive temperature coefficient
PWM - pulse width modulation
RC - resistor capacitor
RLA - rated load amps
SG - specific gravity
SH - Superheat
TOU - Time of use
TXV - thermostatic expansion valve
VFD - variable frequency drive

LIST OF NOMENCLATURE

cUL - Canadian underwriters laboratory
Cv - valve flow coefficient
di/dt - time derivative of current
dv/dt - time derivative of voltage
H - enthalpy per unit mass refrigerant
Hz - unit of frequency, Hertz
I^2R - resistive power loss
Nr - Reynold's number
Qmax - choke flow
R22 - cholorodifluoromethane
R32 - difluoromethane
R125 - pentafluoroethane
R134a –1,1,1,2 tetrafluoroethane
R290 - iso-propane
R407a – zeotropic mixture of R32, R125, R134
R410a - pseudo-azeotropic mixture of R32 and R125
SS304 – one of the popular chromium, nickel stainless iron alloys.
Ton - old English measure of cooling capacity. Approximately 12000 Bth/Hr

Chapter 1

INTRODUCTION AND PROBLEM STATEMENT

1.0 Introduction

Heat pumps have been used for many decades to provide heating and cooling for buildings. The operating principal of a heat pump is as suggested by its name. The device is a pump that moves thermal energy from one location to another; in most cases against the natural direction heat would flow. This thesis details an improvement to the conventional heat pump design that improves the efficiency and capacity of the pump by modifying the refrigeration cycle and through intelligent controls. The goal of this design was to improve the efficiency of an air source heat pump such that its operating cost equals that of a natural gas furnace even after factoring in the cost of the modifications. The increase capital cost amortized over the design life of the heat pump was considered. In his case study Lazzarin [1] examined the difficulty in adoption of HP technology during periods of cheap oil without correction to electricity prices. His study listed approximate COPs needed to match the heating costs of various energy sources. Competitive COP factors when compared with natural gas heating were in the 3.5 to 4.6 range. Other sources such as heavy oil were in the 4-6 range and around 2 for diesel.

For this project and estimate was made assuming current natural gas and electricity prices. A system COP target of 4 or greater was chosen to be competitive with natural gas heating in built up areas and to beat energy costs in rural areas where diesel fuel oil use is common. The cost of heating using a HP vs propane (also popular in rural areas) would be better than natural gas rates given the higher cost of purchase, transportation and storage of propane.

There are two types of heat pumps often used for residential and light commercial applications, air source and ground source [2]. The source specified refers to the source of heat when the pump is operated in the building-heating mode. Air source heat pumps draw heat from the ambient air. Ground source heat pumps, conversely, draw heat from either the earth or a water source. In many cases, a ground source pump is a more efficient option, but not practically possible. It is hoped that the modifications to the air source heat pump will make its efficiency competitive with ground source installations, without the inconvenience and capital expense.

1.1 Air Source Heat Pumps

All vapour compression heat pumps depend on a working fluid whose boiling point changes radically with pressure. By manipulating pressures at various points in the system, the pump forces the working fluid to boil or condense and in doing so absorb or reject heat from its environment. It is this behaviour that is used to shuttle thermal energy in a useful manner. Hence for a well-chosen working fluid, both the latent heat of vapourization and boiling point variation with temperature are substantial. Other favourable properties include high thermal conductivity, low toxicity and environmental compatibility.

Figure 1 shows the major elements of an air source heat pump. The operating principal of the pump is as follows. Liquid refrigerant under pressure is fed through an expansion device into the outdoor heat exchanger. The drop in pressure of the refrigerant as it passes through the expansion device decreases its boiling point. If the liquid was previously warmer than its new boiling point, it boils, cooling itself in the process. When correctly engineered, the resulting liquid temperature will be below the outdoor ambient air temperature. This allows the collection of heat from the outdoor air using the outdoor heat exchanger to boil the remaining liquid. The cold vapour at the exit of this heat exchanger is then compressed such that the high-pressure refrigerant vapour now has a temperature and boiling point above the indoor temperature. This allows both cooling and condensation of vapour inside the indoor heat exchanger. The heat rejected during condensation provides the bulk desired heating of the indoor air. The initial cooling of the gas and subsequent sub-cooling of the liquid also contribute a little to the indoor heating.

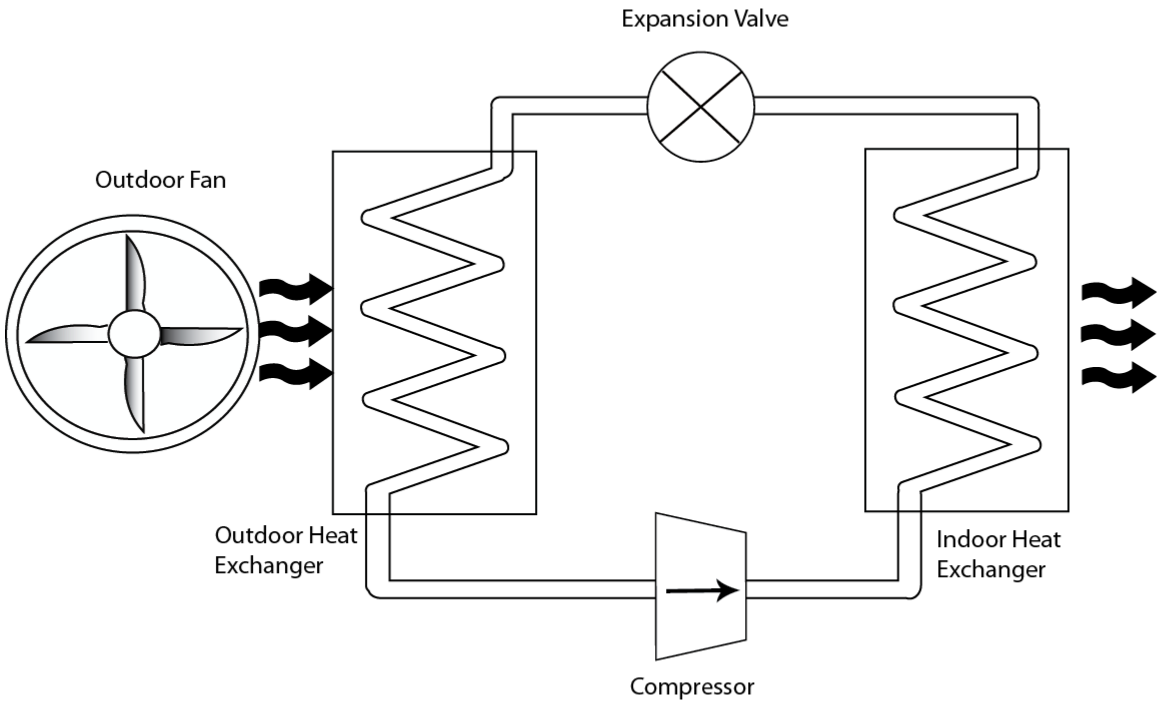

Figure 1 – Major elements of an air source heat pump

The operation of the air source heat pump will be more closely scrutinized in subsequent sections. For now, this simplified description is all that is necessary to introduce the reader to general goals of this design.

1.2 Ground Source Heat Pump

The ground source heat pump is similar to the air source heat pump except that the source of heat has been substituted. The outdoor heat exchanger no longer heats the refrigerant using air. Instead it draws its heat from the ground, an underground water source or from a deep enough body of surface water. Depending on the age and design of the ground source heat pump, it may incorporate an additional heat exchange step between the refrigerant and an antifreeze solution. Figure 2 shows the major elements of a typical ground source heat pump [3].

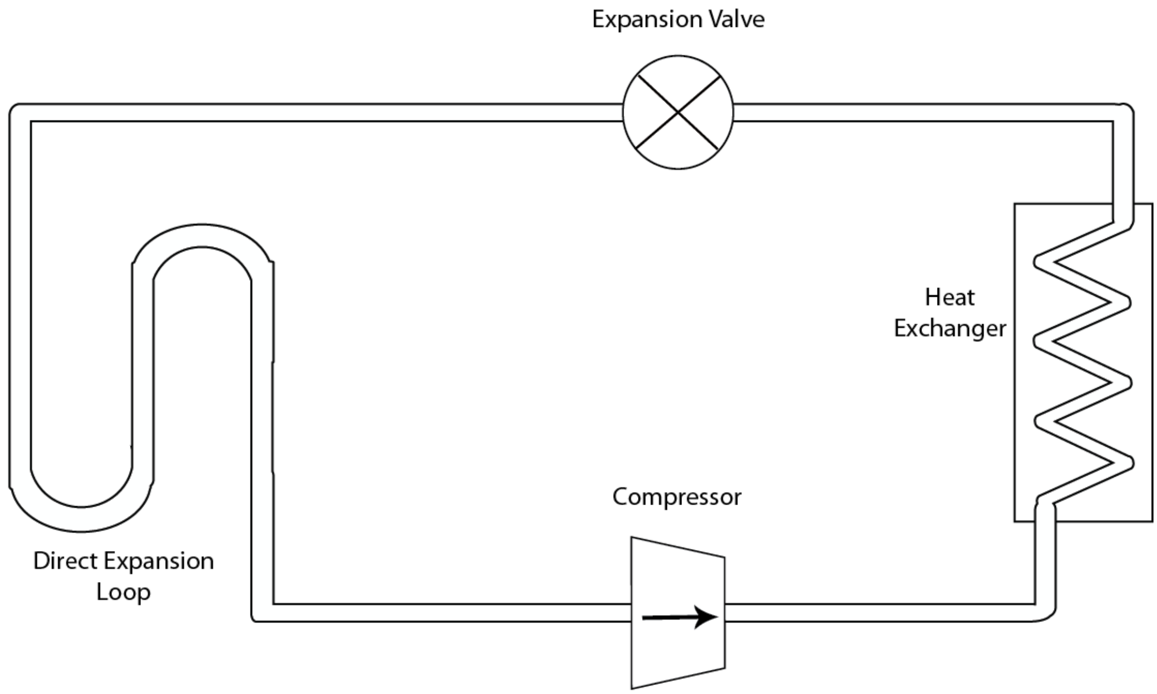

Figure 2a - Major elements of a ground source heat pump, direct expansion

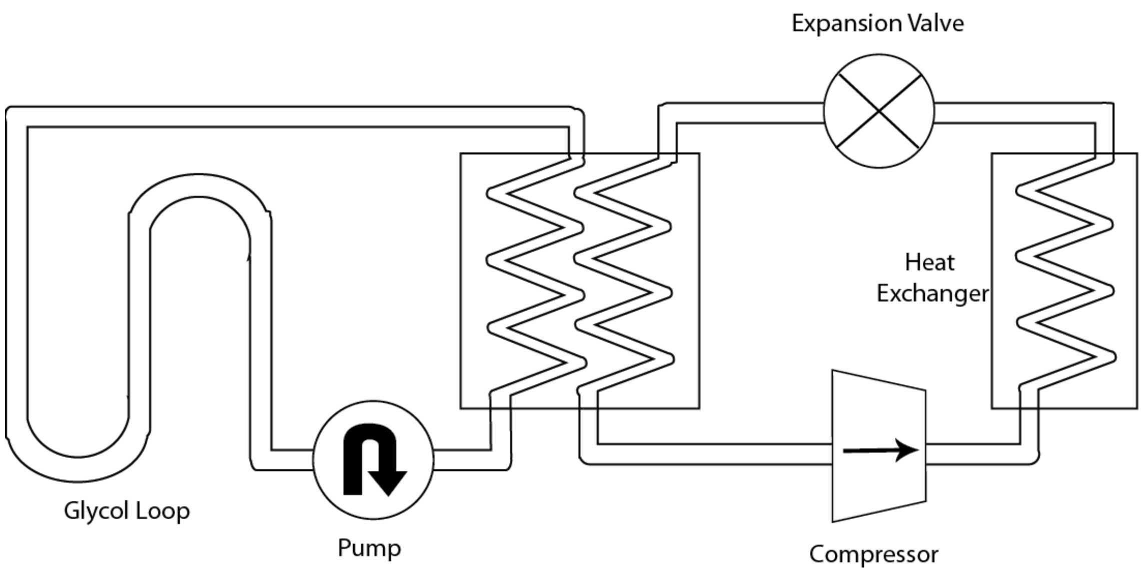

Figure 2b - Major elements of a ground source heat pump, with glycol loop

1.3 General Advantages and Shortcomings

The following table summarizes the major advantages and disadvantages of the two classes of heat pumps, as currently implemented [3].

Table 1 – Heat Pump Feature Comparison

	Air Source	**Ground Source**
Cycle Efficiency	The cycle efficiency is generally more variable and lower for an air source heat pump. This is owing to the variations in ambient air temperature. On warmer days the cycle efficiency and capacity tends to improve. Also of note is the, sometimes, lower efficiency design of the air source heat exchanger. The low thermal mass and low thermal conductivity of air necessitates a larger surface area for heat transfer.	Ground source heat exchangers tend to have more consistent source temperatures, as the mean soil temperature is more constant than the average air temperature. With designs where the refrigerant is fed through the ground loop, the system efficiency tends to be better than conventional air source designs. Some systems that use a separate antifreeze loop to first gather heat to boil the refrigerant are less efficient. Ground source heat pumps that draw heat from a water source are usually very efficient as water limits the minimum source temperature, is thermally conductive and has a high thermal mass.
Installation Costs	The installation costs of an air source heat pump are generally low. Little to no digging is needed in most cases and only a few holes need to be made through the building envelope.	The installation cost of a ground source heat pump can be many times the cost of the pump hardware. Long or deep trenches are usually needed for the heat exchangers. This is in addition to the holes normally needed through the building envelope. Additional costs usually include some landscaping work to restore the site to its original condition.
Installation Space and Zoning	An air source heat pump occupies roughly the same	Ground source heat pump installation normally requires a

	space as an air-conditioning condenser. In most cases, the outdoor unit can fit in a 3x3 foot space. There are space saving units from the Japanese market which occupy even less space. Most town zoning laws allow the installation of a heat pump as long as the drippage from the unit does not inconvenience anyone else (in condo installations).	large open area. Typically each Ton of capacity requires a 150 ft long trench with 15ft separation between trenches. In cities and tight subdivisions where this is not possible, there are vertical trenches systems that can be use so long as there is sufficient access for the drilling rig. There are often in city zoning restrictions on ground source heat pumps because of the drilling/trenching requirements.
Environmental Restrictions	Generally none	Horizontal ground source heat pumps need to be kept away from septic fields. Vertical systems are sometimes subject to environmental restrictions because of the risk of contaminating ground water with surface water. Water (well) source heat pumps are often prohibited due to the quantity of surface discharge. Systems that use a return well are also often scrutinized due to the risk of contaminating the aquifer through the return well.

1.4 Design Objectives

The objective of this design exercise is to make improvements to efficiency and capacity without resorting to brute force methods, such as upsized parts. Intelligent controls exploiting the natural strengths of various design elements are used rather than simply enlarging components. The heat pump's refrigeration cycle was augmented to use liquid injection sub-cooling to extract additional efficiency. The pump is engineered to use simple electronic methods to gain precise control of the refrigerant flow without the usual complexities associated with direct electronic control of the expansion device. Also considered is the cost of production, durability of parts and the ability to fail without total loss of heating capacity.

The heat pump construction was divided into a series of sub objectives that combine to achieve the desired system performance.

Sub-objectives:

1) Refrigeration cycle modifications. The mechanical components of a standard heat pump were altered to include the additional parts needed to provide additional sub-cooling and precise control of the liquid flow into the compressor inlet.
2) Sub-cooling control. Additional sensors and controls were added to allow precise control of the liquid sub-cooling and vapour superheat.
3) Intelligent defrost control. After the main mechanical and electrical systems were tested and tuned, additional real-world efficiency improvements were made by controlling the heat pump's defrost cycle.

Chapter 2

BACKGROUND TO PROBLEM

2.0 Background

The simplest method to improve heat pump efficiency is to select a warmer heat source[5]. Since this cannot be done with an air source heat pump, the only option is to improve the pump itself. This design aims to modify the conventional air source heat pump to make its efficiency comparable to a ground source system. With little control of the air source temperature, improvements to the cycle along with other optimizations must be used to gain efficiency. The modified air source system is slightly more costly than the ground source hardware (when considering just the cost of the pump itself), but costs many times less to install and requires little space.

The augmented heat pump uses two types of methods to raise efficiency. The first type of improvement addresses the fundamental efficiency and capacity limits of the refrigeration cycle. The second group of improvements relate to practical control issues. These improvements scavenge energy normally wasted due to these issues. In cost sensitive applications, some of these methods could be easily retrofitted to a conventional heat pump system. The performance of the resulting system would be less than one with the full modifications, but substantially cheaper. There have been efforts by the Japanese, as noted by Lazzrin, to gain performance enhancement from a heat pump with no regards to cost or complexity. This project does not attempt to do this, instead an attempt was made to balance cost and performance given the current development of the constituent technologies used.

2.1 Refrigeration Cycle Improvements

The majority of the improvement in the new heat pump design comes from a combination of sub-cooling of the liquid refrigerant line and flash intercooling the vapour exiting the outdoor heat exchanger. The efficiency gains come both directly from improvements to the cycle and indirectly from side effects of the changes. In many designs, the cycle efficiency is far from the theoretical limit because of practical limitations imposed by the conventional cycle [5]. This modified design aims to make some improvements through fundamental cycle modifications and more gains through practical improvements made possible as a secondary effect.

Figure 3 shows the refrigeration cycle for a conventional heat pump. Assuming steady state conditions, it is possible to predict the theoretical maximum efficiency and capacity of the cycle using the diagram. All that is necessary is to examine one cycle while taking note of the work input, the heat rejected to the living space and heat gathered by the exterior coil.

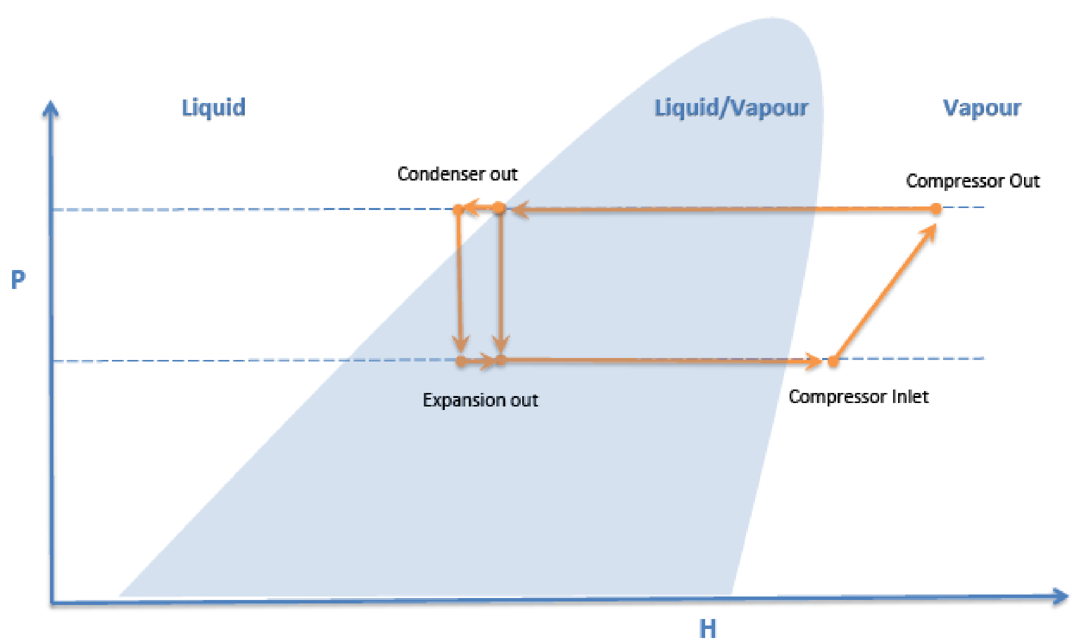

Figure 3 – Mollier diagram of a conventional air source heat pump

The lower left point in the cycle is a good starting point, pedagogically speaking, as this is where heat is first extracted from the outside air. Starting at the lower left corner, the refrigerant has just undergone a change in pressure at (almost) constant enthalpy. This is by virtue of it passing through the expansion device without the ability to gain heat from the environment. A portion of the liquid immediately flashes to gas as the now lower pressure will not support the refrigerant as a liquid without a change in enthalpy. The gas generated here will be of importance later. The temperature of the mixed phase refrigerant, of course, drops. With correct engineering and implementation, it will drop to a point 5-10K below the outdoor temperature. In practice this number may be much more during transient and sometimes even steady state conditions. The gas/liquid mixture passes through the outdoor heat exchanger where it boils at relatively constant pressure. There are of course frictional losses in the heat exchanger, they will be ignored for now. Near the end of the heat exchanger (assuming conventional tuning), the gas mixture gains additional heat such that its exit temperature is somewhat above the boiling point set by the operating pressure (low side pressure). The exit gas is almost always slightly super-heated as boiling is designed to complete before the end of the heat exchanger. While vapour super heat at the compressor is desirable to improve compressor durability, this design aims to minimize it with precise controls. The energy gained by the liquid evaporating in the outdoor heat exchanger is what was previously referred to as the heat gathered by the exterior coil.

The next stage of the refrigeration cycle is the compression stage. The low-pressure gas generated by the previous step is compressed. Work input to the compressor upgrades the usefulness of the gas by raising its pressure to the point where condensation can occur at a temperature suitably above the indoor temperature. Three important properties should be noted. 1) The isentropic compression curves must be corrected for compressor efficiency. 2) The isentropic compression curves typically decrease in slope as inlet super heat increases and 3) Practical systems show additional shifts to the right due to use of refrigerant to cool the compressor motor and crankcase oil.

Unfortunately, most compressor manufacturers are secretive about their full performance data. The exact compressor efficiency numbers are rarely supplied at all operating conditions, but rather as a single number for a specified operating condition and refrigerant. When making estimates of the cycle efficiency, the input work should be adjusted using the manufacturer's claimed efficiency (while being mindful of the differences in operating conditions). In practical calculations the exact figure is not needed as several other factors[1] have a larger influence than the exact number. The other influences are large enough that power measurements are needed to make meaningful use of the results. In particular, cooling of the motor and lubricant in the lower bearing sump has a large influence on measured discharge temperatures, making estimates based on pressure and temperature incomplete.

The falling slope of the isentropic compression curves, as inlet super heat increases, is a nuisance in conventional heat pump designs. Fortunately, for the proposed design this is a property that is exploited to compromise between efficiency and comfort.

The next stage of the refrigeration cycle rejects heat to the building interior. The hot gas at the compressor discharge is first de-superheated, and then condenses. Ideally super-heat is kept at a minimum such that the largest percentage of the heat exchanger is made available for condensing. The heat capacity per unit length is higher for condensation vs de-superheating as the former is a wet process and the latter is dry. This is because the heat transfer to the piping is typically higher where there is mass transport to the surface (gas turning to liquid) to allow heat transfer by phase change rather than just conduction through the boundary layer. As such, it is advantageous to minimize the gas super heat at the condenser inlet. The condenser also serves a second function in the conventional design. It provides sub-cooling of the liquid refrigerant prior to exit. In a conventional design this decreases the condenser capacity, but is a necessary step to ensure correct behaviour of the expansion device. In the case of a piston device, flow can be less than expected. With a TXV, pintle hammering may occur.

[1] crankcase heating, variable inlet superheat, motor core losses and cooling, adjustments due to variable inlet and outlet pressures, refrigerant type

2.2 Performance Shortcomings Due to Deviation from Ideal Behaviour

In the previous description of the normal refrigeration cycle, there was no distinction made between structure vs. scale related effects that decrease cycle capacity or efficiency. Such things as evaporator size, piping size, and compressor swept volume etc. are scale related. Increasing the effectiveness of a condenser by making it larger would be a scale effect whereas providing an alternate flow path for the flash gas would be a structural effect. The former trades off size and cost for effectiveness while the latter adds a measure of subtly to the design, which potentially improves the cost vs. effectiveness trade-off. Structure related effects shall be described separately from scale related effects as scale related effects are readily addressed by increasing or decreasing component sizes. Making changes to a design topology rather than only component sizes allows changes to the rules governing the design trade-offs. This is the driving principal behind the proposed design.

Four unfortunate behaviours of the conventional heat pump design were chosen for improvement in the proposed design. They are: 1) the loss of evaporator capacity as the indoor to outdoor temperature difference increases, 2) the ill effects of flash gas generation in the evaporator, 3) the need for significant super heat at the evaporator exit, 4) the backup of liquid refrigerant in the condenser.

2.3 Loss of Capacity at Low Temperatures and Ill Effects of Flash Gas

Conventional heat pumps will tend to lose capacity as the outside temperature drops. In designs where the evaporator is not underfed, this effect is largely due to loss of gas density at the exit of the evaporator. Piston meter heat pumps do not tend to exhibit this pathology as the refrigerant flow is driven by the high-side to low-side pressure difference. Some TXV designs will underfeed the evaporator at low temperatures due to mismatches between the thermostatic bulb charge and the refrigerant properties. In most small to medium size heat pump designs, the compressor is a constant swept volume design with piston metering. The loss of gas density at the inlet translates directly to a decrease in mass flow through the system. To quantify the loss in gas density, inspection of the Mollier diagram for the refrigerant will do. Please see figure 4. At lower evaporation temperatures there is a drop in gas pressure. With the drop in pressure there is an associated drop in gas density. There is also a secondary effect that can be insidious in practical systems. Consider the segment A in the cycle. The liquid refrigerant undergoes a constant enthalpy process as it is fed into the evaporator. A portion of the refrigerant must evaporate to allow the internal energy exchange needed to go from a room temperature, high-pressure liquid to a cold low pressure mix of liquid and gas. This process diminishes the amount of refrigerant available to evaporate per cycle, with a commensurate decrease in evaporator capacity. In addition to this drop in evaporator capacity there are other side effects that further decrease the evaporator capacity. The additional volume of gas flowing through the evaporator increases the frictional losses and necessitates designs with larger tube diameters. At 273K, liquid propane undergoes roughly 38-times expansion after boiling[2]. The larger tubes tend to further compromise the heat exchanger efficiency by decreasing the volume and surface area available on the atmospheric side of the heat exchanger. The loss of area on the atmospheric side of the heat exchanger can have an insidious effect on efficiency during frosting. As reported by Reindl and Jekel [6], the primary causes of efficiency loss during frosting are the greater thermal resistance added by the frost layer combined with the decreased airflow due to the additional restriction. They have suggested the use of variable pitch exchanger fins, which is made difficult when large tubing must be present throughout the exchanger depth. Also of note is the local cold spot that tends to form at the start of the evaporator tube. The cold spot can spread to other portions of the tube once frost build up has blocked off enough airflow, allowing other portions to cool down.

Often the expansion process is not complete before the mixed phase refrigerant reaches the start of the evaporator coil. This tends to happen in heat exchanger designs that use multiple tubes fed by a liquid distributor. The large pressure drop associated with the liquid distributor combined with the typical practice of feeding it a liquid (by ensuring

[2] Densities taken from National Refrigerants datasheet.

sufficient sub-cooling at the inlet) usually means that boiling will start very shortly after the individual exits. This effect is transient as the outlets are typically short, non-fined and not exposed to the main airflow. In normal use they will cool quickly to the point that boiling happens at the end of the distributor outlet pipes near the start of the evaporator tubes. The rapid boiling at this location often depresses the temperature there while increasing the temperature at the end of the coil. At near freezing outdoor temperatures, this can seed the frosting process and cause premature icing of the coil.

Frias and Aceves [7] have studied frosting behaviour. In their paper they allude to the difficulty in predicting frosting due to the variable nature of the structure depending on the conditions of formation. Their method for predicting frost growth rate is a mixture of first principals combined with empirical data. The frosting process equations assume a frost layer with a given thermal conductivity and porosity. Their study assumes densification of the frost layer as more vapour is absorbed, along with the resulting change in the thermal conductivity due to density and geometry. Unfortunately the predictive process is highly empirical and needs to be fitted with real data. This study is beyond the scope of this thesis.

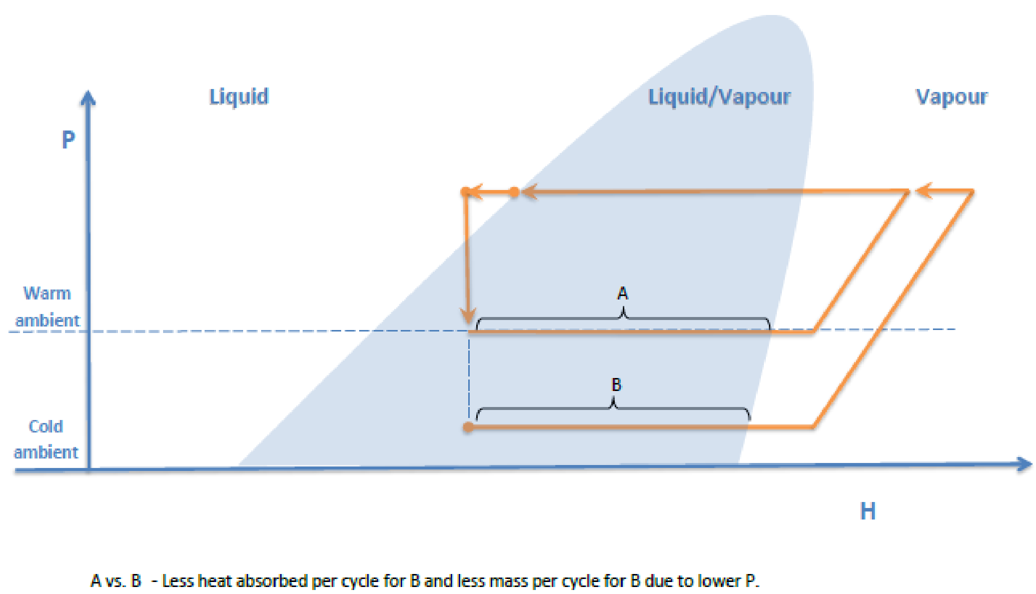

Figure 4 – Capacity loss as a function of decreasing evaporator temperatures

2.4 Poor Expansion Device Control

The ill effects of flash gas described above also influence the behaviour of the expansion device. The generation of flash gas tends to be an unpredictable process as the refrigerant flow through the system is often dependent on the difference between high side and low side pressures. The variation due to pressure is most apparent in fixed orifice or piston style expansion devices, but also apparent to an extent in designs using TXVs (thermostatic expansion valves) or EEVs (electronic expansion valves). When the evaporator is underfed the low side pressure drops and more refrigerant is drawn through the expansion device. This process is not always smooth due to the generation and presence of flash gas in the expansion device. The slugs of liquid refrigerant entering the evaporator cause pulses in the low side pressure, which in turn cause variation in the refrigerant flow. There is a degree of positive feedback in the process. With little damping in the system, it is quite possible for oscillations to occur. Systems employing TXVs can be better behaved if the time lags between the feed process and SH sensing process are short. EEVs are similar in nature, but have the flexibility of finer control and damping being added in software. In general all three methods of control result in some amount of SH margin being needed at the evaporator exit. This is to account for the natural fluctuations in the system and the need to keep the compressor inlet dry. The table below shows some typical SH margins for different designs. These values were inferred from a number of datasheets and tech manuals. The piston settings were taken directly from a setup calibration table for a Carrier 38YKC commercial heat pump. The TXV settings come from the Sporlan TXV setup instructions and selection guide. The EEV SH values come from Danfoss's claimed SH performance for the bundled stepper EEV and control box.

Table 2 – Typical Expansion Device Superheat Settings

Type of expansion device	**Typical superheat setting (Kelvin)**
Fixed orifice (piston)	15-20 K
TXV	5-10 K
EEV	3-5 K

Examining the Mollier chart for the refrigerant shows that it is beneficial, efficiency wise, to maintain as low a SH value as possible. The modified heat pump design helps minimize this value.

2.5 Liquid Backup in Condenser

A conventional heat pump design will typically keep a portion of the condenser flooded. This ensures a degree of liquid sub-cooling. This is at the expense of decreased condenser capacity. There are efficiency benefits to rejecting the sub-cooling heat to the indoor air, but they must be carefully balanced against the loss of efficiency due to the decreased effective size of the condenser. Figure 5 shows the temperature distribution and heat transfer per unit length in the condensers in various scenarios. It is usually best to maximize the effective condenser size and use the bare minimum of the coil to provide sub-cooling. The modified heat pump design eliminates the trade off requirement and allows the condenser to non-flooded.

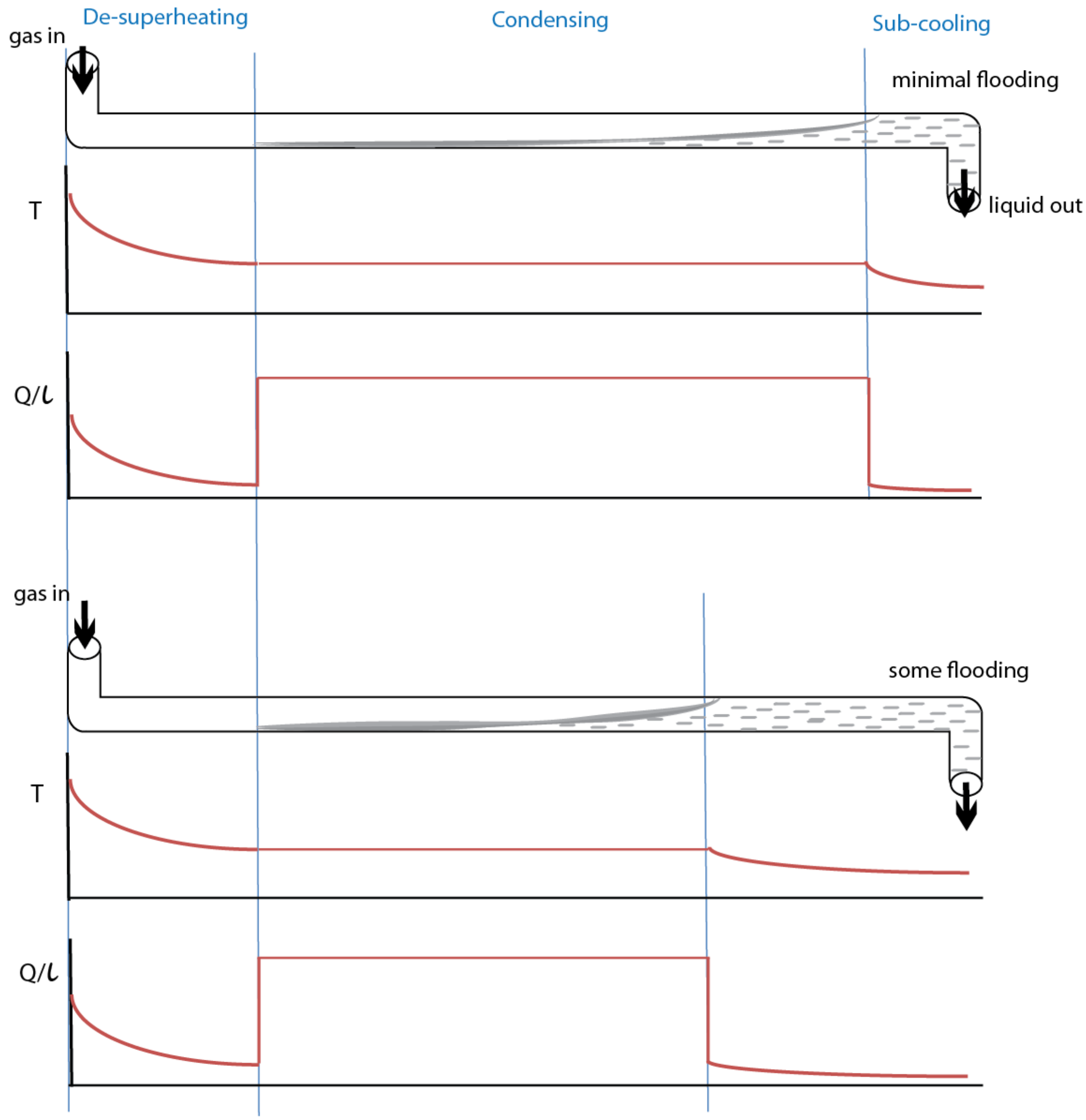

Figure 5 – Condenser Use vs. Flooding

Chapter 3

PRESENTATION OF IMPROVED DESIGN

3.0 Preliminary Considerations

The improved heat pump includes alterations to the refrigeration cycle, and applies intelligent controls to raise efficiency. During the design phase, the value, cost and reliability of each modification was considered. Only modifications that would make economic sense over the lifetime of the heat pump were kept. The efficiency of the pump could have been further increased with costlier parts, but in many cases, the additional upgrades were skipped as the initial increase in capital cost would not be recovered by the incremental improvement in efficiency[3].

The following list shows an approximate cost break down of the major components associated with the pump enhancements. Unless otherwise noted, majority of the pricing applies to single quantity retail.

- VFD inverter – $250. The cost of the inverter would be partially offset by the removal of the higher current contactor, starting caps and run capacitors for the compressor. A typical contactor for a 3Ton compressor is in the $40 range. The two capacitors for phasing the motor are roughly $20. The three phase version of the compressor was also slightly cheaper than the single phase version by roughly $10.
- Wide temperature range TXV - $43. The TXV replaces a piston metering device which has a cost of less than a dollar. The cost of the piston metering device was ignored. An EEV was also considered for metering but not used due to

[3] A Lysholm twin screw compressor, for example, is a great way to increase efficiency and add variable capacity if cost were not a factor. For more details on screw compressors performance please see http://en.wikipedia.org/wiki/Rotary_screw_compressor.

availability and cost. The price of the EEV was $95 from the same supplier as the TXV.

- Heat exchanger fabrication materials – less than $20. The additional heat exchanger was fabricated using bent tubing. The retail cost of the materials was less than $20. In manufacturing volumes the exchanger would be half the cost in materials with roughly the same made up for in labour costs.
- Heat exchanger accumulator – $110. The heat exchange accumulator, a particularly large unit, was $20 more expensive than its regular variant.
- Miscellaneous electronics – Less than $10. In production quantities, the additional electronics would amount to less than $10.
- Additional solenoid – $20. The additional solenoid used was in the $20 range.

The total incremental cost of the modifications is estimated to be roughly $293 for a single prototype. In production volumes the cost difference would be in the $200 range[4]. This cost was considered acceptable given that it would require less than 2 years to pay for itself assuming only a 10% improvement in efficiency and a $1000 yearly heating bill. Appendix F describes the initial inspiration for this design.

3.1 Alterations Used in Improved Design

Three main structural changes were made to the conventional heat pump to improve performance. The three modifications are:

1) A sub-cooling heat exchanger on the high pressure liquid line with electronic controls of the liquid flow
2) An intercooler at the exit of the evaporator
3) Variable Frequency Drive for the main compressor

[4] This is an estimate based on nearly 12 years of experience in the electronics industry building high volume cost optimized products.

Please refer to Figure 6. The structural differences between a conventional design and the modified design are highlighted.

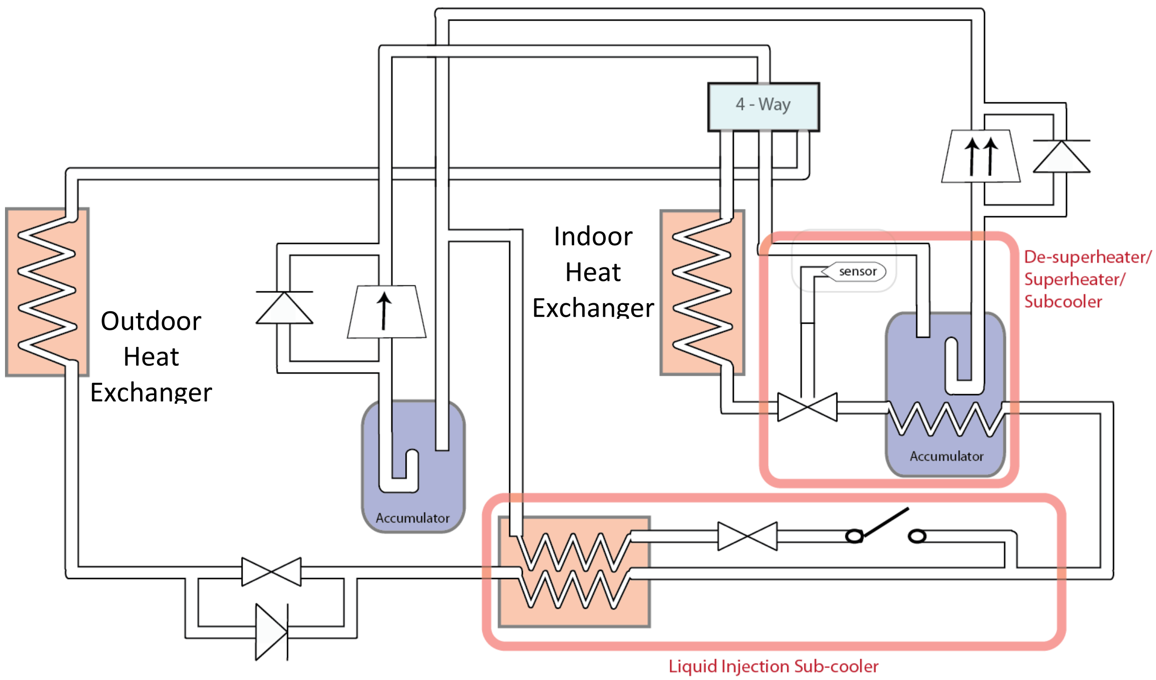

Figure 6 – Overview of Design Improvements

The important end results of the modifications are the following:

1) Zero or slightly negative SH at the evaporator exit
2) Minimal flash gas at the evaporator inlet
3) Minimum SH at the compressor inlet
4) Maximum sub-cooling at the expansion device inlet
5) Variable compressor flow

With the exception of item 5, the above improvements are the result of the modifications that improve the liquid line sub-cooling (without wasting energy) and the controlled intercooling that occurs before the compressor. These two modifications to the refrigeration cycle directly improve efficiency but also make other practical improvements possible.

3.2 Intelligent Control Optimizations

The efficiency of the heat pump is dependent not just on the configuration of the refrigerant loop, but also in the way that the heat pump is controlled. Additional intelligence, beyond what is necessary for basic functionality, can be added to the controls with the hopes of achieving higher efficiencies. These enhancements will be referred to as intelligent control optimizations.

In most installations, heat pumps never reach their rated efficiencies because they are not operated at steady state and excessive energy is wasted performing defrost cycles. The time to achieve steady state operation can exceed 10 minutes. Many HVAC installations have the thermostats programmed for short cycles (some as short as 15 minutes) and the defrost timers set to the minimum time to avoid callbacks to the job site. Transient behaviour is detrimental to efficiency as the heat output immediately after a restart is minimal while the energy input can be close to nominal. Defrost cycles waste energy in three ways. First, the building is cooled to provide heat for de-icing of the outdoor coil. Secondly, resistance heaters are frequently turned on to improve occupant comfort during a defrost cycle. Thirdly, the transitions between defrost and heating states delays steady state operation, which in turn prevents maximum efficiency from being achieved. The time between defrosts for a conventional heat pump can vary between 30 and 90 minutes and may last up to 10 minutes[5].

The modified design substantially decreases the need for defrost cycles. First the decrease in flash gas formation in the evaporator and the associated side effects prevents the initial formation of frost on the coils. This is very important, as the rate of frost formation is very dependent on there being a nucleation point on the coil surface. By making the coil surface temperature as uniform as possible, the heat pump gets maximum use out of the coil before frost starts forming.

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[5] The defrost options were taken from a Carrier 38YKC technical manual and a Diamond Air service manual.

When many conventional heat pump designs are first cycled, the evaporator pressure and temperature initially dip far below the steady state settings. This is because the evaporator tends to be underfed when the pump is first started due to the initially low high side pressure. The evaporator coil must first be pumped down to redistribute the refrigerant such that there is sufficient in the high pressure side to achieve a pressure differential great enough to obtain the designed flow rates into the evaporator. The same system that decreases the flash gas formation also improves the pump's transient response. The additional gas flow, through the extra heat exchange loop, during start-up eliminates the pressure dip (and surplus cooling) that is associated with conventional heat pumps. This has a two-fold benefit. The minimum temperature peaks are lessened and the transition into normal heating mode is accelerated.

Transients are further improved by using an electronically controlled injection system to control gas intercooling before the compressor. The heat pump is able to increase refrigerant flow during start-up without the risk of slugging the compressor. More aggressive TXV settings that allow faster settling can be used because the electronics provide rapid corrections by modulating the injection solenoid valve. The additional fine control here improves transient response and also increases compressor efficiency by allowing a large decrease in compressor inlet super heat. A side benefit of doing this is a reduction in compressor operating and discharge temperatures. The cooling effect on the motor has a small effect on efficiency and a large impact on lifetime. The reduced discharge superheats improve condenser efficiency.

3.3 Additional Improvements

One of the modifications to the design is not a structural type modification, but is still very important to the overall operation of the system. The compressor has been replaced with a 3-phase compressor and fitted with a VFD (variable frequency drive). This allows continuous control of the compressor speed and better electrical efficiency. Continuous speed operation allows the following

1) Improved capacity control through VFD – Continuous speed control of the compressor provides finer control than schemes with on/off or two-step speed control.

2) Improved efficiency through VFD use – The VFD provides an energy efficient method of speed/capacity control vs. other digital compressor control techniques[6]

3) Pressure Triggered Defrost. It should be noted that several articles (Frias & Reindl) have listed the pressure drop this design uses for frost sensing as a direct contributor to evaporator efficiency loss due to the loss of airflow. The defrost controller is in effect directly measuring rather than estimating the effect that decreases efficiency.

4) Reduced Piping Sizes due to VFD drive – piping sizes can be reduced when the compressor is VFD driven. At ambient conditions above freezing, the compressor can be slowed such that the pipe capacity is not exceeded while still maintaining capacity at low temperatures. In a conventional design the piping must be sized for worst case conditions: high ambient temperatures and full compressor capacity. With intelligent controls, compressor speed is limited as ambient temperatures rise so large pipes are not needed. While an ideal pipe diameter can be optimized for, the design is limited, economically, by the standard pipe size offerings. In practice, it was found that a single pipe size reduction was possible.

[6] Copeland digital scroll is an example of a capacity control method is less efficient as the motor is run at full speed but capacity reduced by separating the running scrolls. Copeland themselves have made mention of their VFD scroll designs being most efficient of their capacity modulation methods.

http://www.emersonclimate.com/en-us/Products/Compressors/Scroll_Compressors/copeland_scroll_commercial/Pages/modulation_technologies.aspx

5) Good power factor and the ability to run on generator power. Almost all VFDs are equipped with active input power factor correction. This combined with a programmable soft start allows system operation from generator power without the need to oversize the generator.

3.4 Mechanical Systems

The heat pump design was configured to allow a series of tests using single stage and two-stage compression. The intent was to run using a single compressor at medium to high ambient temperatures and to use the second stage compressor as needed during low ambient conditions. Figure 7(a) shows the configuration of the pump setup.

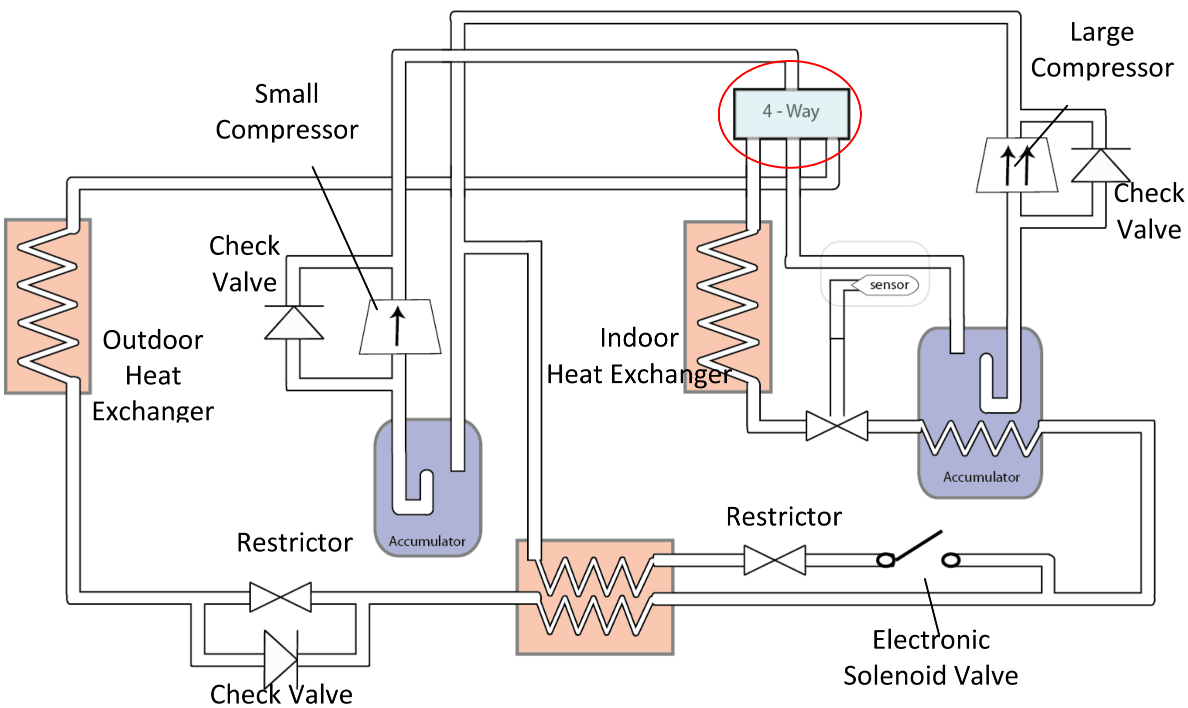


Figure 7(a) – Mechanical modifications

The 4-way valve's symbol, circled in red in Figure 7(a), matches the industry standard configuration. The two possible states of the valves are depicted in Figure 7(b).

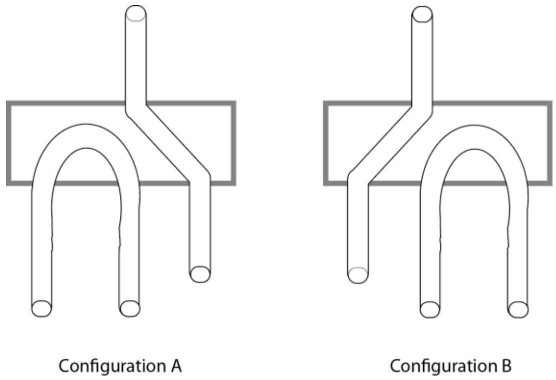


Figure 7(b) – 4-way valve configurations

Two-compressor operation is only intended for use during low ambient operation. Operating both compressors during warm conditions was not planned for due to the likelihood of overheating the second compressor.

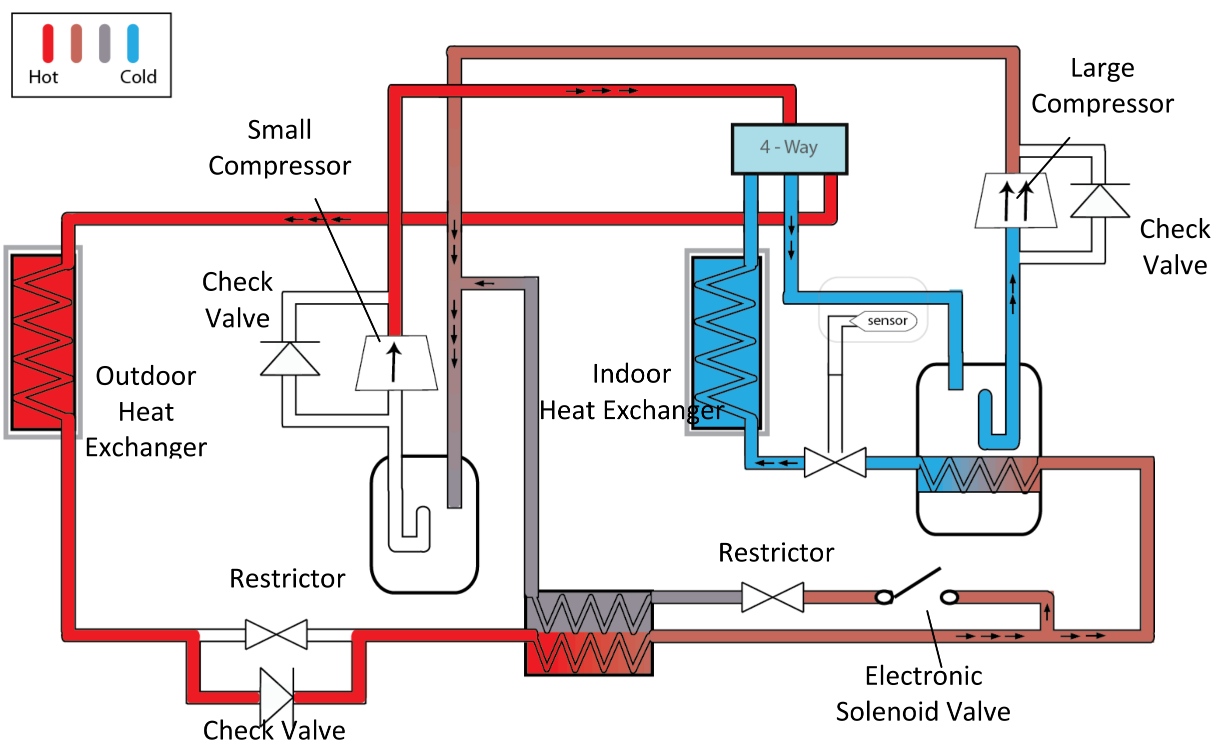


Figure 8 – Two compressor mode flow diagram

During two-compressor operation the refrigerant flows as indicated in figure 8. A portion of the hot refrigerant exiting the condenser is flashed to provide subcooling for the rest of the liquid before it is fed to the outdoor evaporator. This portion of the refrigerant is evaporated at the system's intermediate pressure and only needs to undergo partial compression to reach the high side pressure. This configuration is similar to a vapour phase injection economizer cycle except that this design has independent control of the two compression stages and the ability to reduce the inter-stage super heat to zero by over injecting liquid. The standard vapour phase injection economizer cycle includes a reservoir at an intermediate temperature and pressure. The liquid exiting the condenser is partially evaporated in the reservoir to yield a mid temperature, mid pressure mix. The vapour from this reservoir is fed to an intermediate tap in the compressor (usually mid scroll or mid screw) while the liquid is expanded then fed to the evaporator. In the conventional economizer cycle, having both compressors implemented on a single shaft locks the stage-to-stage compression ratio. Also, using the conventional design does not allow inter-stage liquid injection without scroll erosion. This is because most economizer style compressors are implemented by machining a third port in the middle of the stationary scroll. Unlike the main inlet port, which is fed through the crankcase, there is little free volume before the scroll. There is little tolerance for liquids present at the third port in these designs.

A further refinement, unrelated to the compressor configuration is the use of excess refrigerant in the evaporator to sub-cool the incoming liquid. This is done with a heat exchanger in the suction accumulator as shown in Figure 9. The evaporator is deliberately overfed to ensure that liquid is present in the accumulator. The evaporation of this liquid in the accumulator provides the heat removal for cooling the incoming liquid. During two-compressor operation, control of superheat at the inlet of the first compressor is not as precise as the liquid injection solenoid is used to regulate the inter-stage superheat.

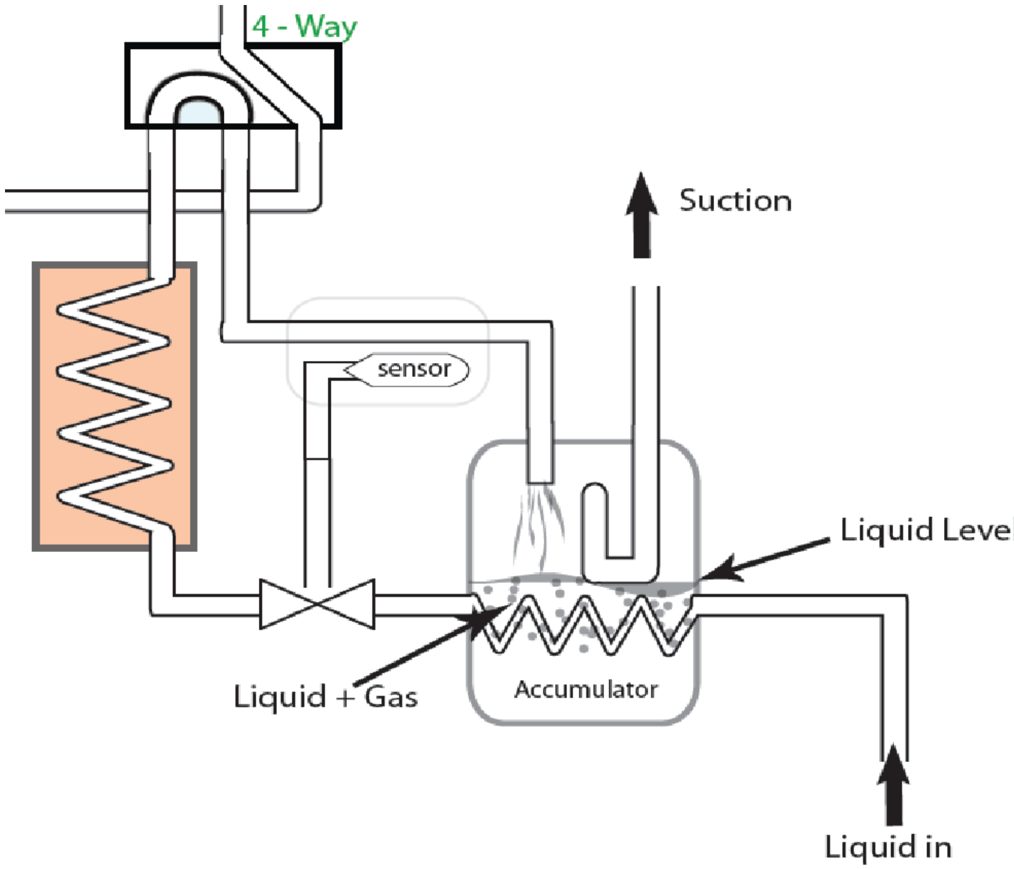
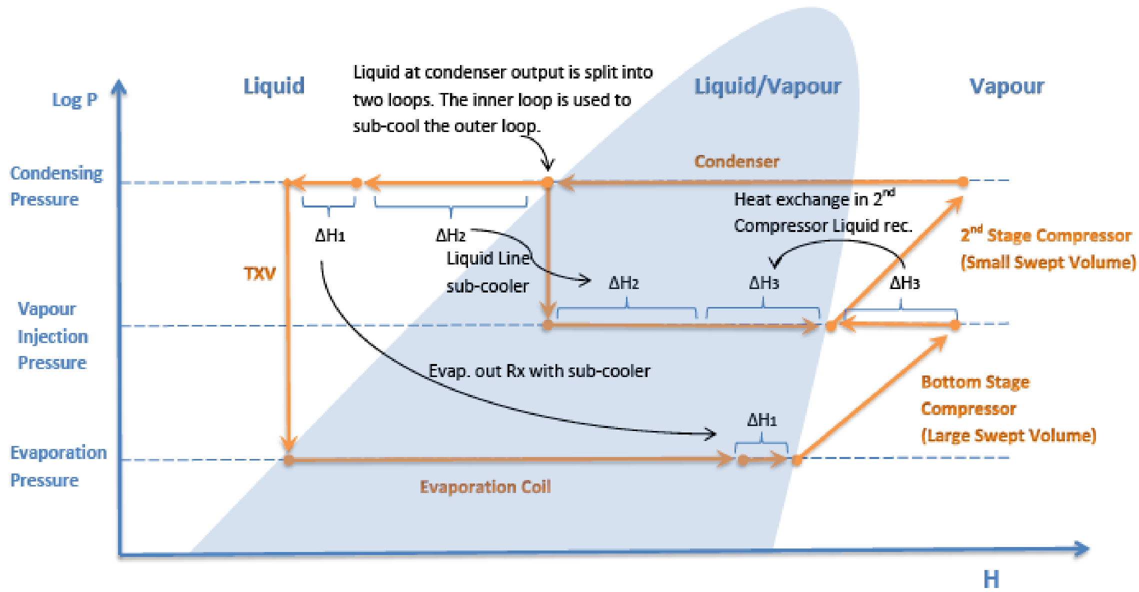


Figure 9 – Heat exchange in suction accumulator

The following Mollier chart shows the expected two compressor operating condition.



ΔH_1 Heat exchange between liquid line to 1st compressor's liquid accumulator.

ΔH_2 Heat exchange between liquid line and vapour injection line.

ΔH_3 Internal heat transfer in 2nd compressor liquid accumulator.

➔ Compression super heat from 1st stage boils excess liquid from vapour injector line.

Figure 10 – Mollier diagram for dual compressor mode

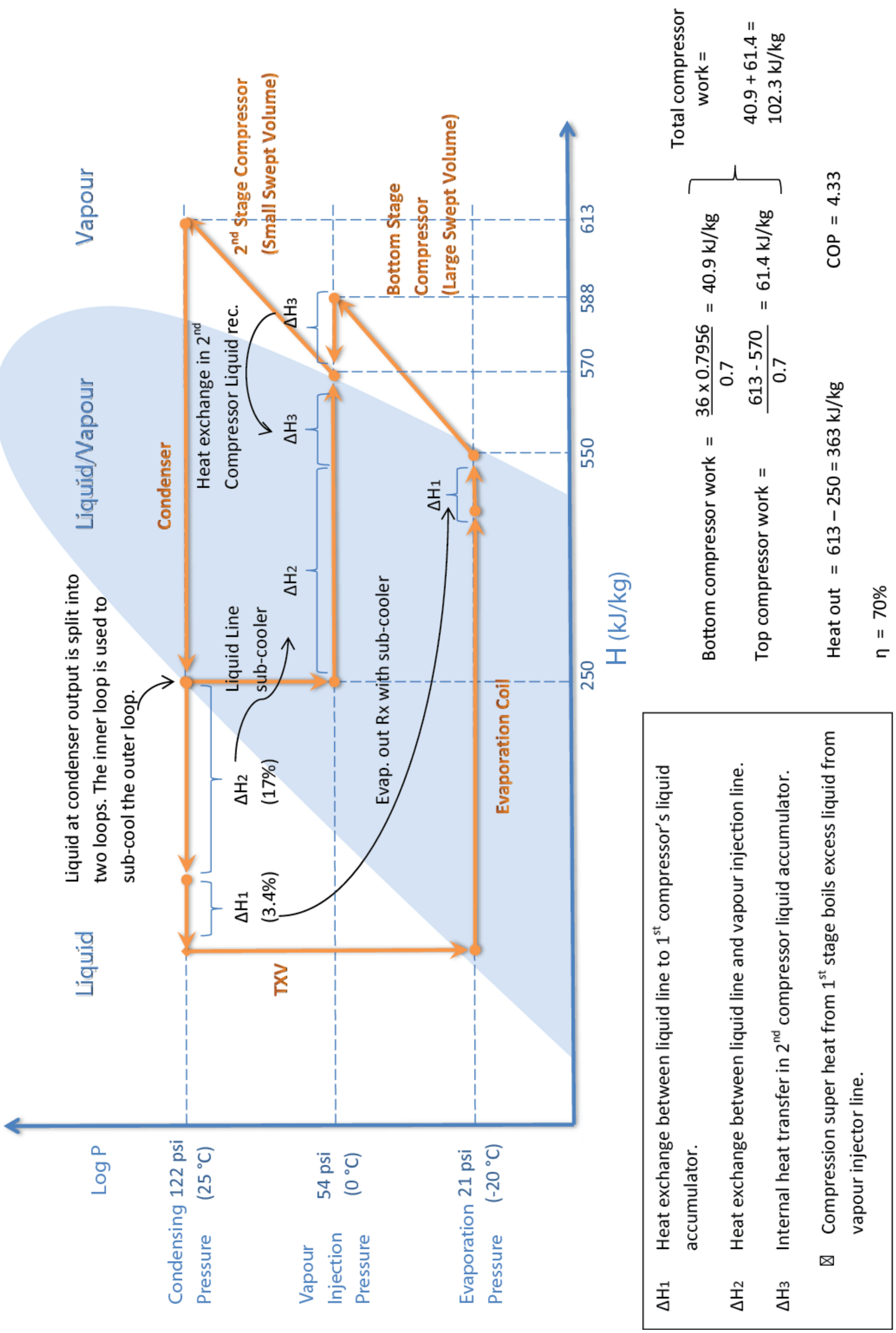


Figure 11 – Mollier diagram for dual compressor setup, with expected operating point

During moderate and warm conditions, only the smaller upper compressor is run. Refrigerant bypasses the lower compressor through a check valve in parallel with the compressor. Figure 12 shows the refrigerant flow in single compressor mode.

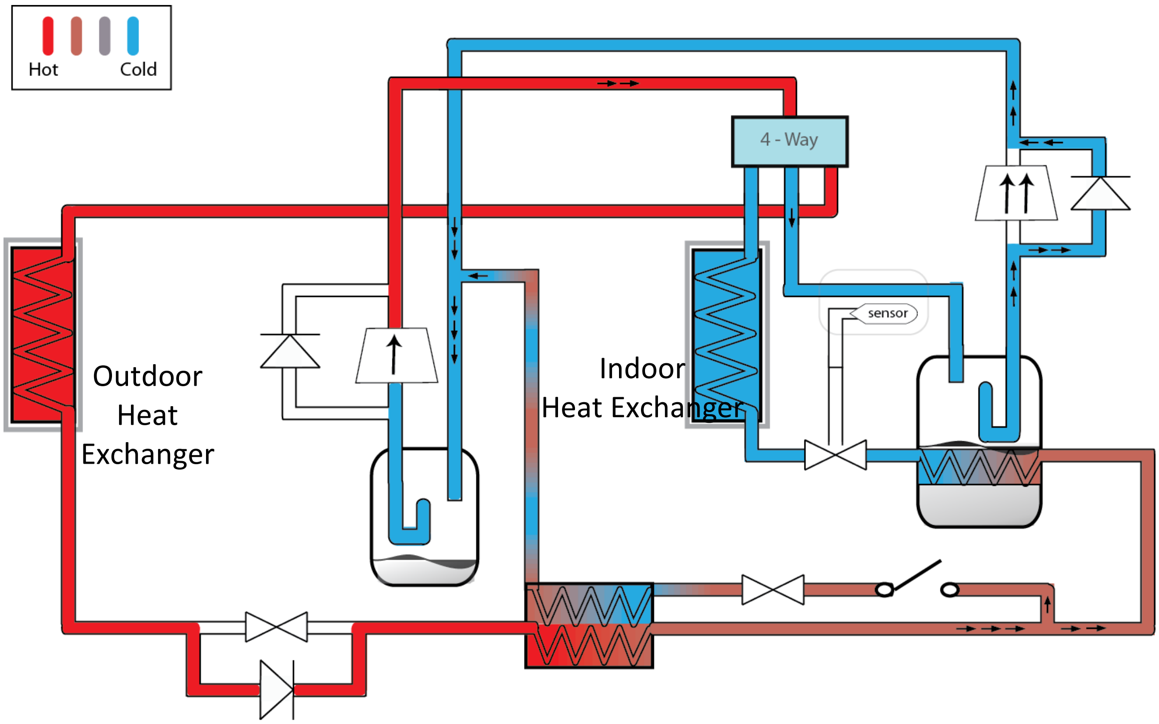


Figure 12 – Single compressor flow diagram

During single compressor operation, very precise control of the inlet super heat is possible as the liquid injection solenoid that feeds the sub-cooler also controls the compressor inlet superheat. To control the inlet superheat, sufficient liquid must be present at the exit of the sub-cooler. Figure 13 shows the accumulation of this liquid after it exits the evaporator. Under these conditions, the maximum possible liquid sub-cooling has already occurred and controller is at liberty to modulate the flow to achieve the desired superheat at the compressor inlet. Despite the liquid present in the accumulator, there tends to be some vapour super heat resulting from not all of the heat exchanger coils being submerged. In the extreme condition, all of the coils are exposed. The TXV is set to maintain a few degrees of superheat after the accumulator heat exchanger and the liquid injection system provides the remaining superheat control.

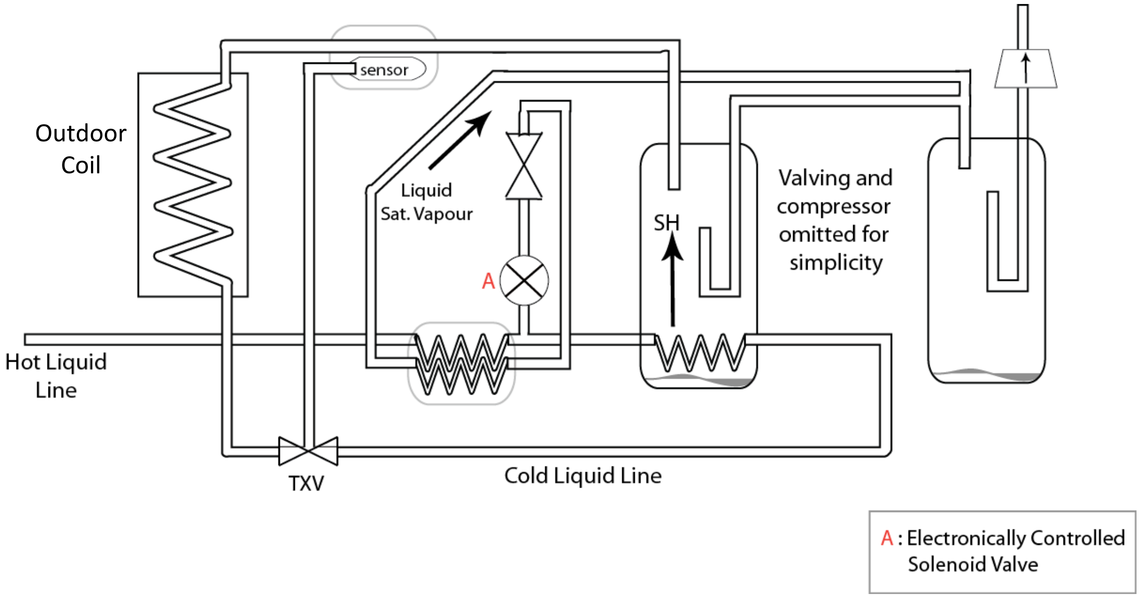


Figure 13 – Sub-cooling and Superheat Control

The system oscillates slowly between two states. When the liquid injection solenoid is on (Figure 14), liquid refrigerant is evaporated to cool the bulk of the liquid flow before the accumulator heat exchanger. During this state little heat exchange occurs in the accumulator, the evaporator is slightly overfed and the liquid level in the accumulator rises. The TXV is a laggy element and will not respond immediately. During this period, the inlet super heat drops. Fortunately, the electronic controls are quick and detect the drop in compressor inlet superheat. At a set minimum superheat, the liquid injector is closed.

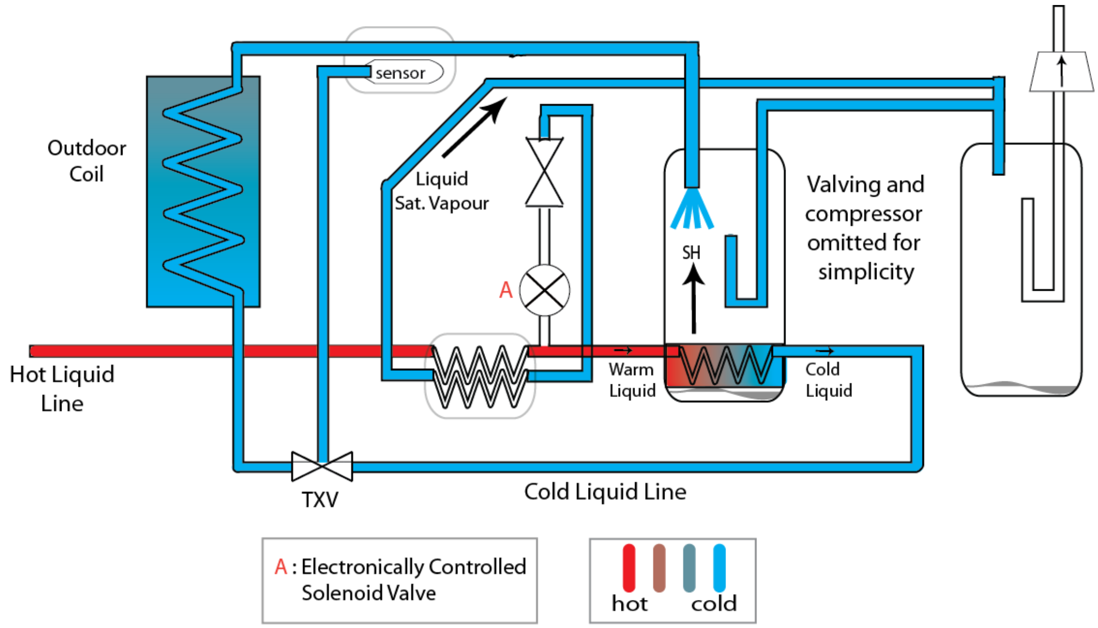


Figure 14 – Superheat Control, Injection On

After the liquid injector has been turned off, the second operating condition occurs. There is little heat transfer in the sub-cooler, but significant heat exchange in the suction accumulator's heat exchanger. As a result, the liquid feeding the TXV continues to be sub-cooled. The hot liquid refrigerant gradually evaporates the liquid refrigerant in the accumulator (left over from the injector on state). Because a large volume of liquid is in contact with the refrigerant vapour at the inlet pressure, the resulting super heat remains close to zero. The internal heat exchange ensures near zero gas super heat despite the heat transferred from the hot liquid refrigerant in the accumulator heat exchanger. Eventually, the reservoir of evaporating liquid refrigerant is depleted and the compressor inlet super heat rises. Please see Figure 15. The injector-controller senses this and the liquid injector is turned on. Ideally (from a performance perspective) this process would be controlled continuously, but the cost and complexity associated with continuous liquid level sensing and valve modulation were too much when compared to the incremental improvement in performance. The observed injector cycle times are several minutes while performance is only degraded for seconds during the transitions between modes.

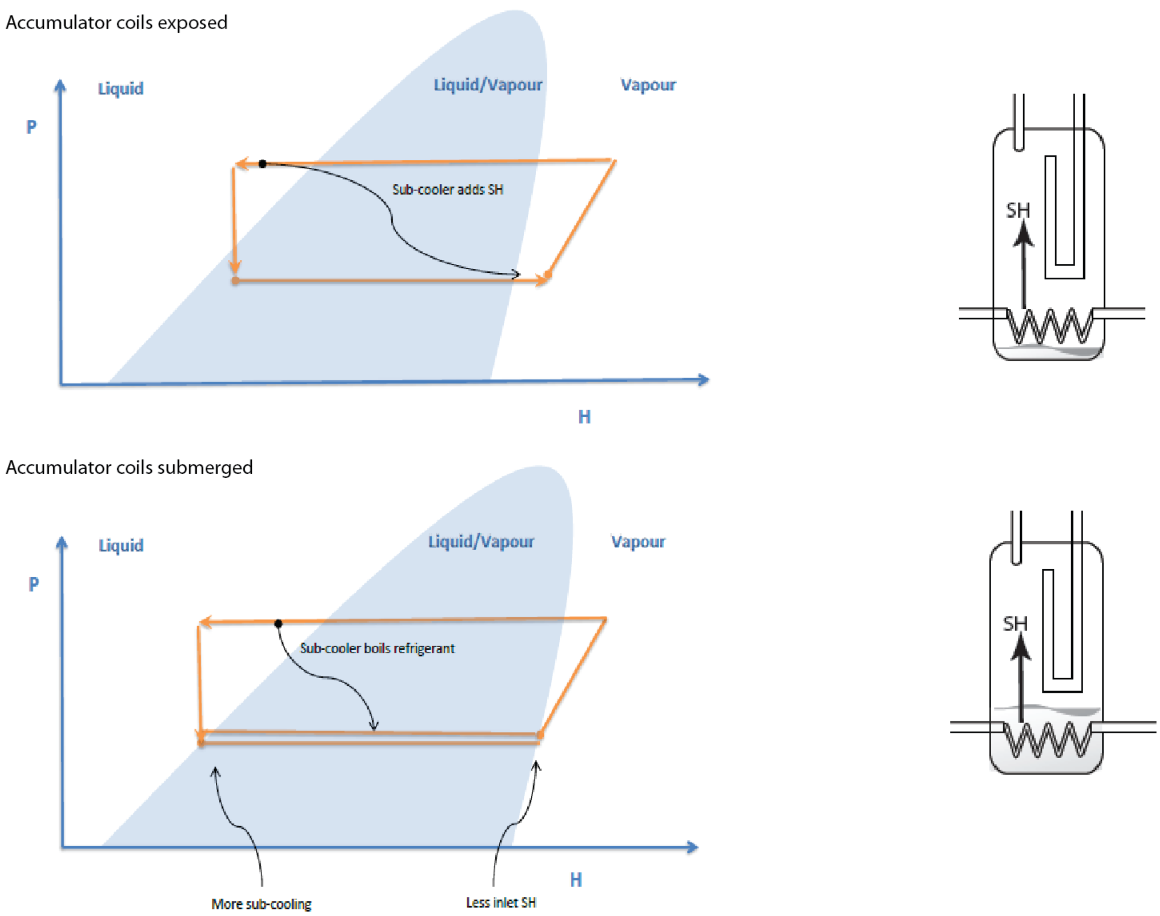


Figure 15 – Mollier Diagram for Superheat Modulation, Injection Off

Figure 16 shows the expected system pressures and flow during single compressor operation.

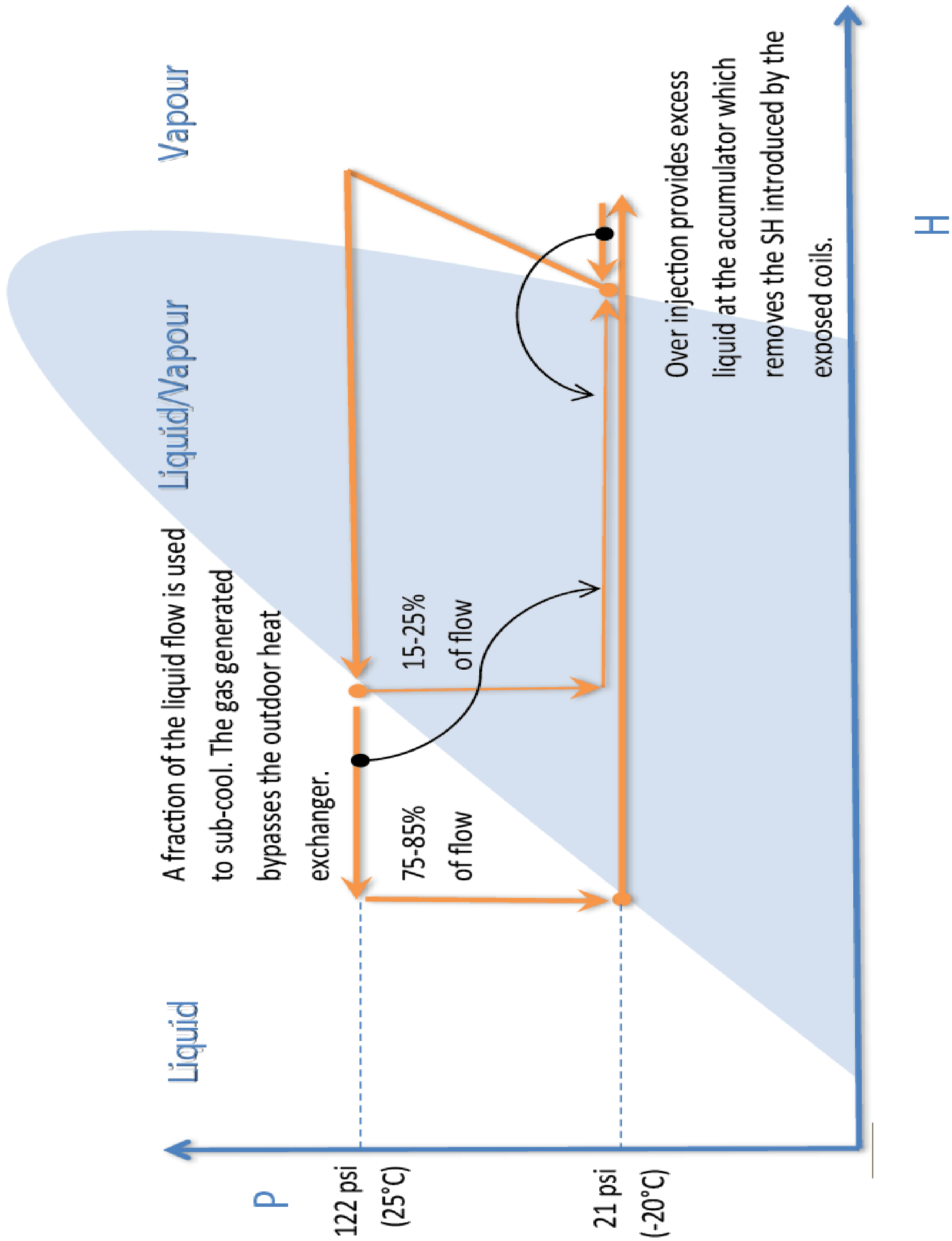


Figure 16 – Mollier Diagram in Single Compressor Mode, Injection On

3.5 Component Selection and Sizing

The target heat output of the design was 30000 Bth/hr. Normally, this figure would determine the compressor size, and to a lesser degree the indoor and outdoor heat exchanger sizes. In this design, however, there is some flexibility in the compressor swept volume as the compressor shaft speed can be varied[7]. The compressor mass flow rate is now a product of swept volume and shaft speed instead of being fixed. The main constraints to component selection now becomes a fitting exercise where the pressure/flow relationship at the compressor is related to a torque/rpm pair at the motor, which is ultimately expressed as a voltage/current/frequency relationship at the drive inverter.

3.5.1 Inverter Selection

First, the maximum size driver inverter that would run off of a single-phase 240V feed was selected. This was done to give maximum flexibility for the subsequent component choices. A single-phase input type inverter was selected because of the power available at the test site. Three-phase output was selected so that smooth and efficient variable speed operation would be possible. All single-phase induction motors require a split winding with some form of phase shift to generate a rotating field. The phase splitting mechanism is usually only efficient and/or functional at a single frequency. This is not compatible with the design goals, hence the selection of a three-phase inverter and motor combination.

4 Drive inverter brands were considered. Brand A was selected for this project. The largest of their single-phase 200V class units was picked.

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[7] The use of a variable frequency drive (VFD) inverter allows compressor speeds other than 60Hz. The lower speed limit is typically set by lubrication and sealing requirements and the upper speed is limited by vibration and bearing durability.

Table 3 – Inverter Brand Comparison

Brand[8]	Pros	Cons
A	Good Power Efficiency Many aux features and programmability Well specified electrical interface and controls Well documented de-rating factors and service intervals Good Motor Protection Features Good reputation for durability	Cost Programming Complexity
B	Cost Local Availability	Programming interface lacking Unknown reliability
C	Cost Good power efficiency	No serviceable parts (main filter cap or cooling fan) Limited programmability of aux features
D	Designed for compressor control. Good Efficiency Software designed for HVAC usage	Cost Prohibitive MOQ 3 month lead time

Many of the Brand A drive inverter auxiliary features were used in this design. Their use will be discussed in the electrical section. The only portion relevant to the mechanical section is the maximum output of the drive inverter as relates to the compressor selection. The selected inverter's maximum output is rated at the lesser of 11A/4.9 kVA.

[8] Brands considered are listed in appendix A

3.5.2 Compressor Selection

Several compressor manufacturers were evaluated for this project: Mitsubishi, Copeland and Danfoss. Mitsubishi does not deal with individuals or small companies. Danfoss, while having excellent quality, has a poor distribution network in North America. There were no Danfoss compressors in stock at the largest HVAC wholesaler in Toronto. Their lead times were too uncertain for this project. Copeland compressors were selected based on availability, cost and inter-changability. The next table shows the feature choices (other than swept volume) driving the compressor model selection:

Table 4 – Compressor Selection Considerations

Feature	Notes
Scroll or Piston	A scroll compressor was selected for two reasons. Volumetric efficiency over a wide range of compression ratios & scroll compliance. Scroll compliance is needed for this design as there is a chance that liquid will be introduced into the compressor. With a piston style compressor, liquid ingestion is usually a catastrophic event, where as with a scroll compressor, there is increased wear but no sudden failure.
Mineral Oil or POE lubricant	Mineral Oil was selected. Due to the experimental nature of the system, there would definitely be significant amounts of moisture introduced to the lubricant during prototyping. POE lubricant degrades rapidly in the presence of moisture. Also, desiccation of the system is much more difficult with POE lubricant present.
Single vs Three Phase Motor	A three phase motor was selected so that continuously variable speed control would be possible.

3.5.3 Swept Volume Selection

4 Copeland scroll compressors were evaluated for pairing with the drive inverter. The first digit pair in the compressor part number is the nominal capacity when used with R22. This number is sufficient for a rough estimate of compressor size. The maximum compressor speed of 1.4 times nominal was selected to limit the bearing loads to 2 times nominal. Also, the minimum compressor speed was limited to 75% of nominal to ensure reliable lubrication. Both Copeland and Danfoss have indicated that their compressors can be expected to function at 50% nominal speed. Testing on the Copeland unit showed scroll mating at 30- 40% of nominal speed. ~ The 75% figure was selected so as to be conservative as the compressors' oil pumps are centrifugally driven and there is little to no compressor flow till 40% speed.

The largest model considered, when run at nominal speed would give the target heating capacity. Unfortunately that compressor is too large to allow sufficient reduction in capacity at lower compressor speeds. In addition, the largest compressor's current consumption is too close to the inverter's maximum output to allow significant over-speed operation. The smallest model considered, did not have sufficient swept volume to reach the target capacity at a safe operating frequency. The smaller of the two remaining compressors was selected so there would be more margins for lubrication and minimum capacity vs maximum capacity. Maximum capacity is anyway limited by the inverter output, so there would have been little benefit to using the ZR28K3 compressor.

Table 5 – Compressor swept volume considerations

Model	Swept Volume	Rated Load Amps	Full Load Amps	Watts
ZR21K3	1.77	7.7	6.1	1980
ZR26K5	2.2	8.6	7.0	2340
ZR28K3	2.37	10	7.9	2550
ZR30K3	2.54	10.7	8.3	2740

Additional adjustments are needed to account for the rating system used on the compressor datasheets and for changes in refrigerant composition. They will be considered later.

3.5.4 Refrigerant Selection

A number of factors are considered when selecting a refrigerant. The most obvious candidates were refrigerant mixtures designed as R22 replacements. The compressor is of a R22 design and it would make natural sense to select a working fluid which retains, as much as possible, the properties of R22. Sami and Tulej [8] studied a ternary mixture comprised of HFCs. Their work showed results typical and consistent with many of the other R22 replacements candidates. Sadly, COP was degraded. Comparing their results to a commercial R22 heat pump shows 10-20% degradation vs R22 (despite their claims of good performance).

Reassuringly, it has been suggested by Devotta [9] that flammability is one of the compromises that has to be made in selecting a R22 replacement with equal or better performance. During the development of this heat pump, flammable refrigerants were considered. R22 refrigerant, in practice, is flammable due to the presence of dissolved mineral oil. In Devotta's paper, he notes a European study on the exaggerated risks of R290 in heat pump.

The first constraint on the refrigerant is chemical compatibility with the compressor lubricant and compressor design. R717 was immediately discounted, as the selected compressor is a hermetic type that exposes the motor windings to the refrigerant. R717 is corrosive to copper. R410A, although very popular in modern designs, was also discounted, as it requires the use of a POE lubricant. POE lubricants were avoided in this design due to moisture sensitivity and the difficulties in desiccating the piping sufficiently after brazing. HFC1234 was not considered due to toxicity concerns. R22 was considered but was not used for environmental and cost reasons. While R22 is an excellent performer, it has been banned from manufacturing. Remaining stocks of R22 are very costly and difficult to obtain. Refrigerant R290 (propane) was ultimately picked because its pressure temperature curves are similar to R22 while its thermal conductivity is substantially higher. This permits the use of many R22 rated components with only minor adjustments. The heat transfer per unit volume for R290 is 5-7% lower than R22 but this did not make much of a difference in practice as higher thermal conductivity allow lower condensing pressures and higher evaporation pressures than would have been expected with R22. One of the major benefits of R290 in this design is the lower condenser operating pressure. The lower high-side pressures allowed the compressor to be run a substantially higher speeds without overloading the motor or drive inverter. Efficiency gains were found by using a smaller compressor scroll at a higher operating speed.

3.5.5 TXV Sizing

TXV sizing is required to take into account both the maximum and minimum flows in the system. A TXV that is too small will not allow enough flow to achieve the desired system capacity. An oversized TXV will chatter at low flows due to a combination of too much control gain and increased control hysteresis from stiction. Most conventional TXVs are able to operate smoothly between 110% and 50%[9] of their nominal flow. This design has a total range of operation that exceeds these limits. Fortunately the proposed design has a fortuitous operational property that increases the operational range of the typical TXV. During warm outdoor ambient conditions, where less capacity is needed, the decrease in compressor speed allows further reduction in the TXV flow, without fluttering, due to the decrease in pressure differential.

Most commercial TXVs are flow rated assuming nominal condensing and evaporation pressures and also based on the density of the refrigerant passing through them. The flow

[9] Sporlan TXV sizing guide

charts are usually provided for R22, R407a and R134a and R410a. The figures for R22 were adjusted to account for the use of R290 whose density is substantially lower than R22. The now lower condensing pressures were also accounted for when the TXV was sized. Please see appendix C for the adjustment method. It should be noted that the flow regime through the TXV was such that viscosity made little difference to the flow estimates. The flow through the restrictor was almost completely dictated by fluid density.

3.5.6 TXV Thermostatic Charge Selection

Thermostatic charge selection is critical in a wide temperature range application. The superheat setting of a TXV will vary as a function of temperature depending on the bulb charge. Certain gas mixes in the TXV will result in wildly varying SH settings and or a non-functional system at certain temperatures. It is also important to consider the effects of charge migration for certain gas fills. Charge migration is an ill effect where the thermostatic charge evaporates from the bulb, condenses in the TXV power head and is unable to return to the bulb. Loss of valve control occurs under this condition as there is no longer a mix-phased mixture in contact with bulb on the suction line. Charge migration can occur in cases where there is a liquid fill in the TXV bulb and the power head is installed lower (and non-inverted) than the bulb. For this design, it was very important to select a thermostatic charge that is immune to charge migration. Firstly, the temperature range of the design is very wide. Secondly the TXV body and power head tend to run much cooler than conventional designs due to the sub-coolers in line with the liquid line before the TXV. Thirdly, a malfunctioning TXV will be masked by the liquid injection system, making the problem much more insidious. For this design a wide range gas thermostatic charge that is adsorbed in a charcoal brick was selected. At no point does the gas condense, thus preventing charge migration. The uniformity of SH vs. temperature was given lower priority since the liquid injection system is capable of making significant corrections to regulate SH.

3.5.7 Liquid Injection Solenoid Selection

The liquid injection solenoid selection was simple as its needs are modest. First material compatibility and operating pressure constraints must be satisfied. All valves rated for R22 pressure will function at R290 pressure levels. A refrigeration valve with a 304 stainless steel pintle and bronze seats was selected for good longevity. Also, since the flow rates were quite small, the smallest direct acting solenoid in the catalog was sufficient for this application. Direct acting solenoids, vs. pilot operated, tend to be more reliable and have the side benefit of having lower pressure[10] drops at low flows. The flow direction through the selected direct acting valve needs to be observed as it not a pressure balance type and uses fluid pressure and flow to assist closing.

3.5.8 Heat Exchanger Design for Liquid Injection

Please see appendix B for the rough hand calculations used to size the liquid injection heat exchanger. Two assumptions were made to simplify the calculations. First, the mix phase side of the heat exchanger is isothermal at the evaporation temperature set by the low side pressure. Second, the flow rate through the liquid side is sufficient to thin the boundary layer within the pipe, such that the heat flow is dominated by the conductivity of the solids making up the heat exchanger. After the heat exchanger was sized, the length was roughly doubled for margin. Given more time to optimize the design, a sight glass would have been added at several points in the heat exchanger to allow careful study of the minimum required length of heat exchanger. Because of the difficulty in forming the pipes the sight glasses were not added. Due to limited supplies of refrigerant, multiple size heat exchangers were not experimented with.

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[10] Small refrigeration pilot operated valves derive their piloting pressure source by introducing a pressure drop in the line. Typically 1-2 psi MODP (minimum operating pressure differential) is required to allow actuation of the solenoid valve. At extremely low flows some pilot valves will flutter, usually greatly accelerating wear.

3.5.9 Accumulator Sizing

Unlike a conventional design, the liquid accumulators serve two purposes in the design. First they provide compressor protection by allowing a volume where refrigerant can accumulate while the levels in the condenser and evaporator are being established. Secondly, the accumulator provides a convenient location to install a sub-cooler.

The extra system volume is especially important during the start-up transient when a large quantity of liquid refrigerant may have accumulated in the outdoor coil[11]. During initial cold start-up, refrigerant will have migrated from the warm condenser and indoor lines to the cold outdoor accumulator. The indoor liquid line may be filled with vapour or bubbles under these conditions. When the suction is first applied, the vapour or bubbles in the liquid line, combined with a high low side pressure, can result in a large volume of liquid refrigerant being pushed out of the evaporator without actually evaporating. This is because vapour easily passes through the TXV orifice and the high low side pressure may push the evaporation point above the outside temperature. The accumulator gives the liquid refrigerant somewhere to accumulate other than the compressor's oil sump. Many designs that have very long lines also have a solenoid valve prior to the evaporator inlet to decrease charge migration effects. This was not done for this design, as the cost of the extra solenoid was more than upsizing the accumulator. The accumulator was sized large enough to hold nearly 2/3rds of the refrigerant capacity to account for the extra line lengths associated with the injection sub-cooler. The passages for an accumulator of this size are more than adequate for the suction flow rates as the lines are conventionally sized to match the intended capacity. One important accumulator specification is the size of the orifice used to allow oil return. Lubricating oil tends to accumulate at the bottom of the accumulators and is carried back to the compressor through a pinhole in a portion of the suction tube that dips to the bottom of the accumulator. If the accumulator is too large in a conventional system, proper oil return does not occur, as the gas velocity in the dip tube is too low. Fortunately the choice of R290 as the refrigerant mitigates this problem.

[11] Refrigerant has an unfortunate tendency to accumulate in the coldest part of the system.

Mineral refrigeration oils are very soluble in R290 and return to the compressor very easily.

The second use of the accumulator in this design is to provide sub-cooling and ensure near zero suction superheat. Normally the heat exchanger in the accumulator is used to assist boiling off surplus refrigerant making it out of the evaporator and to provide positive superheat to the suction gas. In this design, the evaporator is deliberately overfed so as to keep the heat exchanger coils in the bottom of the accumulator submerged. The key difference in use is the degree of flooding of the accumulator. The liquid level must be kept high enough to ensure that are few exposed heat exchanger coils. Exposed coils increase SH whereas submerged coil do not.

3.5.10 Condenser and Evaporator Sizing

In a conventional heat pump design, sizing of the heat exchangers is critical to maintaining the correct level of refrigerant in the coils. The relative sizing of the two coils has a large impact on the refrigerant balance, which in turn sets system capacity, vapour super heat and liquid sub cooling. This design is much less sensitive to the sizing of the heat exchangers and benefits from being fitted with the largest heat exchangers that the available space can accommodate. This is because sub-cooling is provided external to the condenser, SH regulated separately and excess refrigerant stored in the suction accumulator. Rather than depending on fluid backing up the liquid line into bottom portion of the condenser for sub-cooling, the condenser can be more fully drained of condensate. This increases the useable capacity of the condenser and improves transient response by minimizing the volume of refrigerant in the loop. A sight glass was installed at the outdoor end of the liquid line. The sight glass allows observation of the state of the liquid line and optimization of the refrigerant volume.

3.6 Electronics Description

To save cost, the bulk of the components used in this heat pump design were taken from a commercial heat pump. After the ban on R22 refrigerant, new old stock[12] units dropped in price as they have limited commercial value due to the dwindling refrigerant supply. When this project was undertaken, the cost of the components scavenged from a complete unit would have cost more than double the unit had they been purchased separately Purchasing a new old stock R22 unit, then rebuilding it, was the most economical plan. The commercial heat pump included a set of electronics, which monitor safety interlocks and perform rudimentary defrost control. Because of the retrofit nature of the project, most of the new electronics consist of glue circuits to interface the existing heat pump electronics to the new mechanical parts. The remaining new circuits are present to modify the behaviour of the existing electronics.

Figure 17 shows the wiring for the drive inverter and the new 3 phase compressor motor. Appendix G contains a list of common HVAC signal designations. The compressor motor has an integrated thermal protector which opens the centre of the Y connected field windings. It is important to program the drive inverter's thermal protection and current limiting such that they trigger before the compressor's protection circuit. This is to decrease the chances of damage to the drive inverter due to an unexpected open circuit at its output. Also important is the grounding of both the drive inverter chassis and the three phase lines. There is high current leakage to the chassis of the driver inverter, which poses a dangerous condition should the ground connection be broken. The three phase line must be shielded to reduce EMI as the output of the driver inverter is an unfiltered PWM signal.

[12] New old stock is a term used in the surplus market to describe an item that has not been used, hence new, but is coming from a old stock pile. They are frequently discounted to decrease inventory.

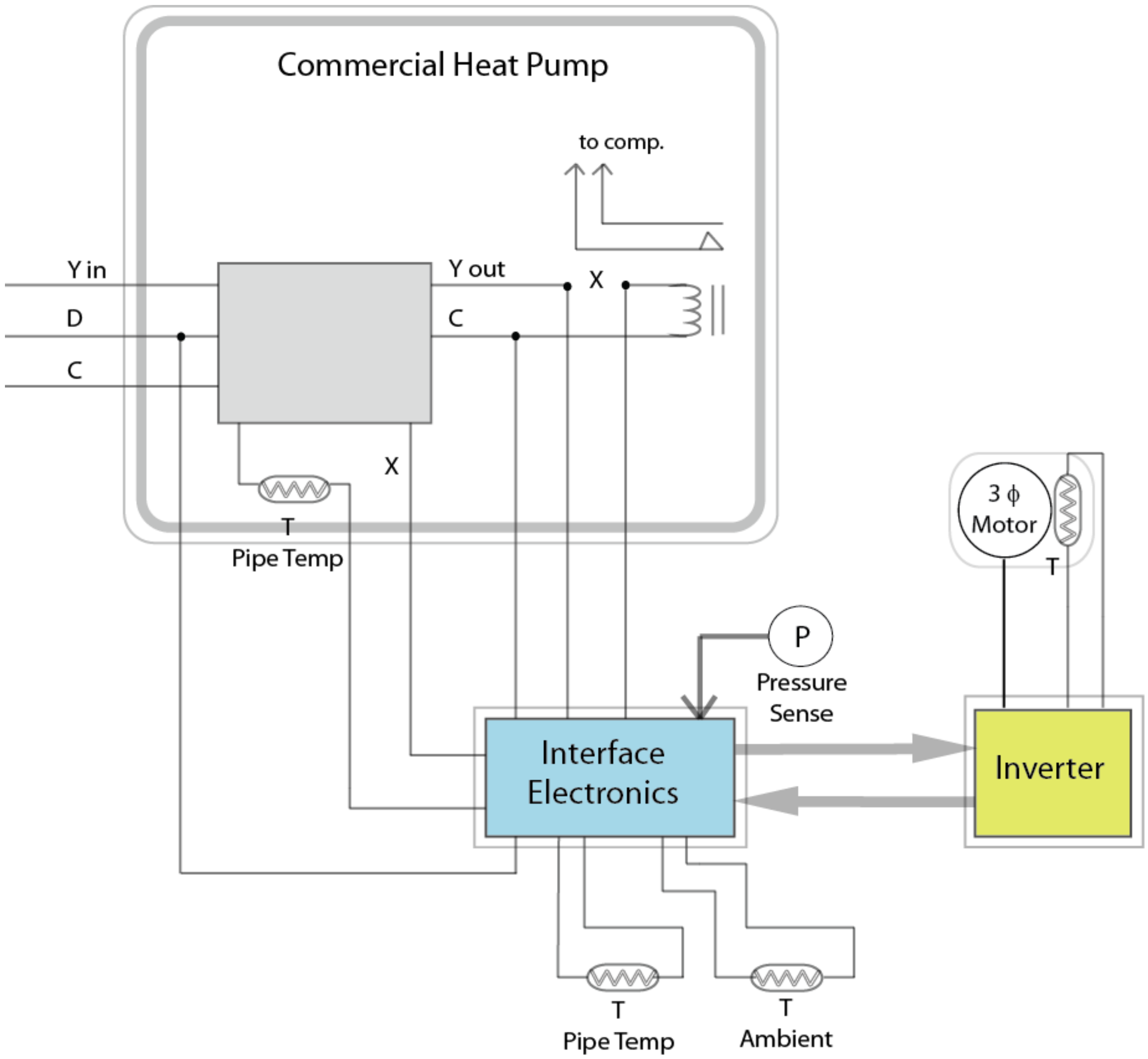


Figure 17 –Block Level Diagram of Inverter Hookup to Existing Pump Electronics

Figure 18 shows the addition of the thermal protection for the compressor on the prototype. Note that the cut in the pipe wrap was made on the bottom to facilitate drainage. A PTC (positive temperature coefficient) thermister is used to monitor the compressor discharge pipe temperature. In the event of blocked flow, or excessive refrigerant charge, the compressor discharge temperature sensor will shut down the drive inverter before the compressor's internal temperature protections opens the motor windings. The positive temperature coefficient of the thermister adds additional protection in the event that the wiring is accidentally cut or damaged. An open thermister circuit results in the drive inverter shutting down.

Figure 18 – Photo of Compressor Thermal Protection Placement

Figure 19 shows the interface between the existing heat pump electronics and the new drive inverter. Functions of the electronics provided with the heat pump include under-pressure protection, over-pressure protection, over temperature protection, under temperature (ambient) lockout, reversing valve control, and compressor start-up delay protection. The safety and sequencing functions of the existing electronics are maintained but modified to suit the new mechanical system. Control of the main contactor in the existing heat pump is intercepted and used to provide a run signal for the new drive inverter. In addition, the defrost auxiliary heat request signal is intercepted and used to command a separate compressor run speed during defrost cycles. This is important to avoid impairing system capacity when running at lowered compressor speeds. It should be noted that the interface electronics provide isolation between the old electronics and the new parts so varying ground potentials do not cause unintended effects or current flow. Also, the interface electronics take into account that most HVAC systems use 24V AC signalling. The 24V AC signals are rectified and filtered to provide appropriate drive for the optical isolators, which then interface to the driver inverter's DC logic inputs. While more elegant circuits could have been used, there is a certain durability that comes with the larger components and longer time constants. This is important when interfacing with an inherently noisy system, such as a typical HVAC system's. The time delays introduced by the filtering process are negligible in comparison to the soft-start times programmed into the drive inverter and the time constants associated with the inertia of the motor compressor system.

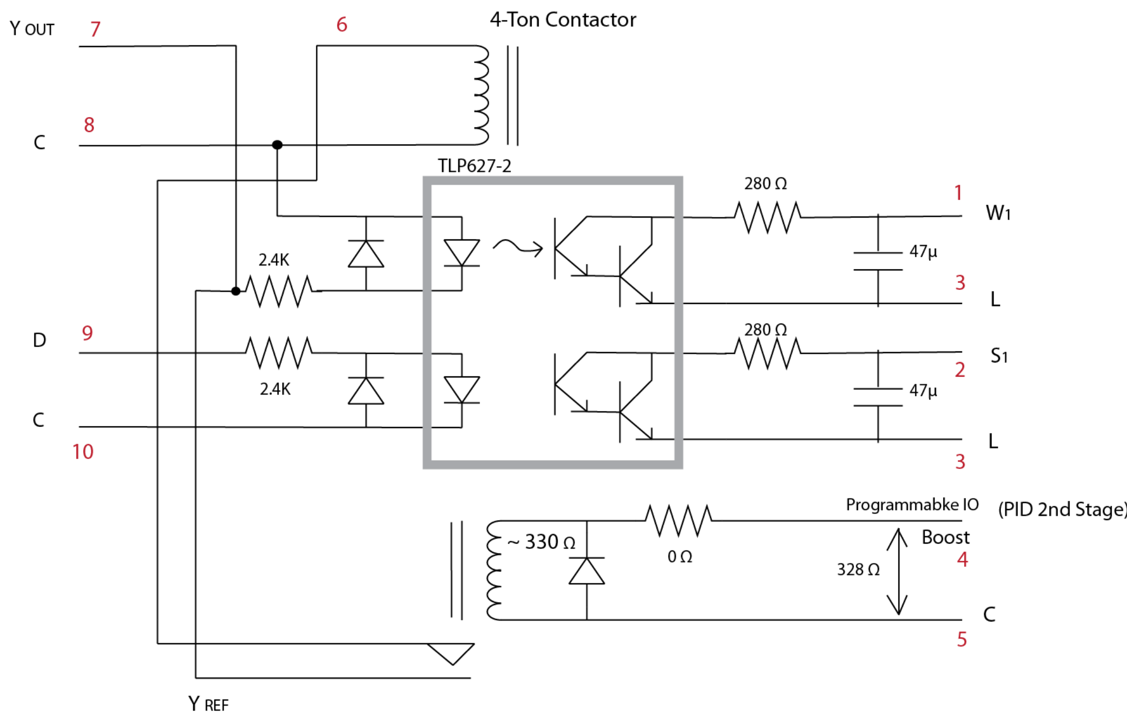


Figure 19 - VFD to pump electronics interface

The signals on the left of the diagram are from the heat pump's control board. Y_{out} is the drive signal to the heat pump's compressor contactor. D is the defrost signal, which is normally used to call for auxiliary heat from the indoor air handler. The signals are opto-isolated and used to drive a set of biased control signals on the VFD. W1 and S1 are run and speed control signals respectively. L is the VFD's isolated return for these control signals. A provision was made to let the saturation output form the VFD’s PID controller sequence the pre-existing compressor in the heat pump. The function was not used in this project.

Figure 20 shows the details of the liquid injection sensing electronics. The liquid injection system is controlled by measuring a pair of temperatures. The details of the control will be explained later. The design of the electronics for the liquid injection system takes into account the slow nature of the thermal sensors. The electronics are slowed to the point that good noise immunity is achieved. There is no point in designing quick readout electronics given the inherent lags in the thermal sensors themselves. The input clamp structure provides protection from ESD and accidental shorts to the 24V system. Shorts to the 240V line were not designed for, though the input stage would likely survive them, such a short would be indicative of a much more serious wiring problem.

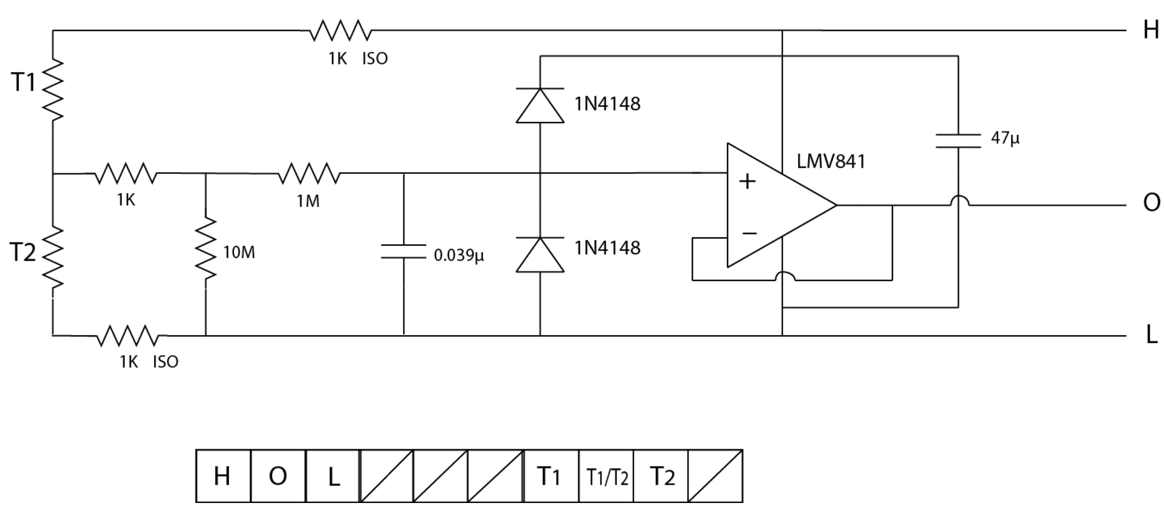


Figure 20 – Liquid Injection Interface, Sensor Side

The added electronics provide a voltage indicative of the imbalance between the temperatures of the two temperature sensors. It is this imbalance that is used to control the liquid injection logic. The signal representing the imbalance is fed to the drive inverter where one of the built in functions is used. The scaling for the signal is based on a reference voltage provided by the drive inverter. The intent of doing this is to maintain an output that is ratio-metric to the reference potential, and thus decrease the variation of the system's performance with time, initial calibration and temperature. The drive inverter provides the ability to monitor an analog line and compare it against a set of limits. This was one of the useful features that drove the selection of the drive inverter. The particular model selected has many integrated functions, including programmability, which save on development time and testing. The output of this function, along with the limits, can be assigned to various functions via programming. In the case of the liquid injection, the software is designed to look for a compressor run condition in addition to a given set of temperature differences. When all conditions are met an output is driven.

The liquid injection control output from the driver inverter is optically isolated and used to control the injection solenoid. Please see Figure 21. The design of the isolation circuit was driven by cost and certification requirements. A cUL[13] listed module was chosen for ease of future certification but it also had the side benefit of being more cost effective and manufacturable than a discrete solution. The isolation circuit is a fairly standard design. The input drive is scaled to provide 2 times margin to account for device aging. The isolation module is opto-triac based and requires an output snubber to limit current and voltage slew during commutation. The output snubber design was optimized to commutate the 2x the expected transient from the solenoid based on measured parameters. The snubber was designed to provide 3x safety margin during commutation, then fitted to the closest standard component values. Please see appendix D for snubber design notes.

[13] Canadian UL mark

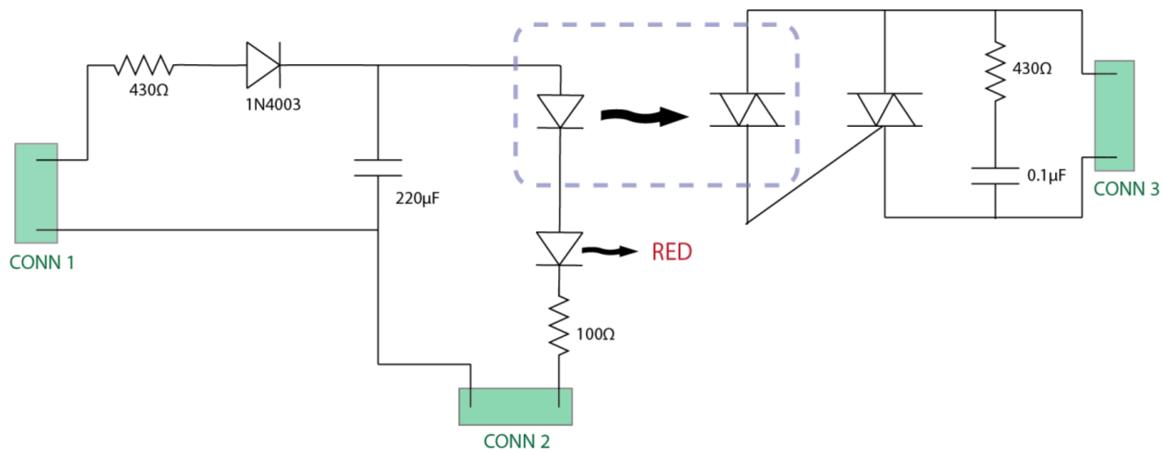


Figure 21 – Liquid Injection Interface, Solenoid Driver

Figure 22 depicts the defrost cycle inhibitor. The system level design details (trigger points and control algorithm) will be provided in a future section. This section will only describe the implemented function. The defrost inhibitor circuit works by tricking the existing heat pump electronics into thinking that the evaporation temperature is above freezing when defrost is not desired. As implied there are two parts to the circuit. One portion determines if defrost is necessary, and the second portion manipulates the existing heat pumps signals to alter the original defrost behaviour.

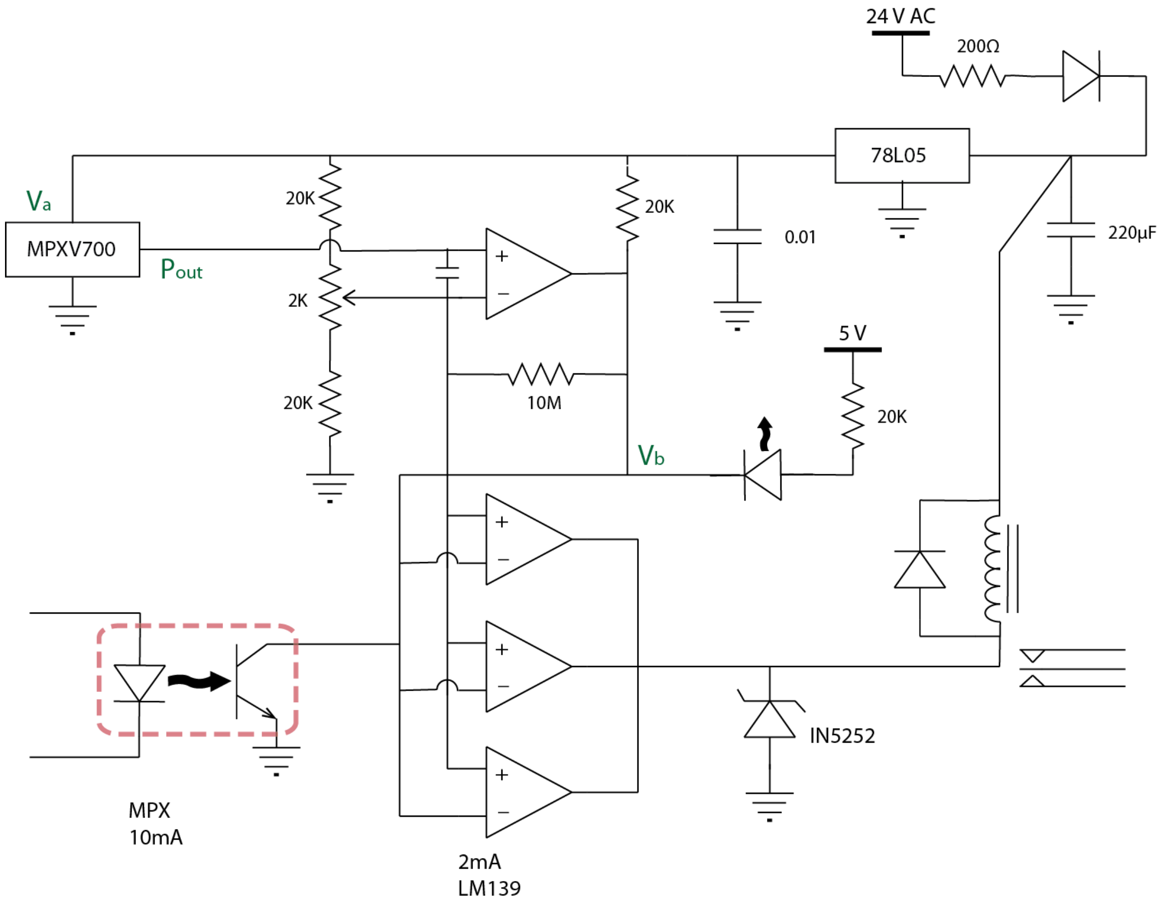


Figure 22 – Schematic of Defrost Inhibitor

The need for defrost is determined by monitoring the pressure drop across the outdoor heat exchanger. A differential pressure sensor was installed for this purpose. Care was taken when routing the pressure sensor hoses downwards to avoid water ingestion. As frost accumulates on the fins, the void area decreases with a resulting increase in pressure drop. This pressure drop is sensed by the differential pressure sensor and compared against an adjustable set point. The comparison circuit is designed with a small amount of hysteresis so as to avoid chatter when the pressure nears the switching threshold. The result of the comparison is buffered and interlocked against the defrost indicator cycle. The interlocking in important so that the inhibit signal maintains the correct state during the defrost cycle when the fan is disabled. The design of the buffering circuit took into account the cost savings associated with using a quad packaged comparator. It also took into account the cost and reliability improvements that occur by reducing the current draw on the 5V supply rail.

The fan control interface consists of software in the drive inverter and an external isolated buffer. The software control provides a delay between compressor run and fan activation. The intent of the delay is to allow faster system pump down by allowing the high side pressure to build more rapidly. The quicker rise in high side pressure allows liquid refrigerant to be more quickly transferred to the evaporator. A side benefit of the delay is improved user comfort as the initial blast of cold air from the heat exchanger is shortened. The optically isolated buffer is similar in design to the liquid injection system, with the snubber re-optimized for switch line voltages instead to 24V.

3.7 System Level Optimizations

A number of system level optimizations were made in this design to achieve the overall design goals. Most of the optimizations were of an efficiency nature. A few, however, were done to address practical control issues.

3.7.1 Efficiency Optimizations

One of the main design goals of this design is to provide improvements in system efficiency through continuous control of the compressor speed. In addition to flow capacity, there are a number of losses that vary based on compressor speed. Both losses and flow capacity are non-linear with speed. Hence there is a shape to the efficiency vs operating speed curve. A system level optimization that operates the compressor at the best speed for every operating condition can generate substantial efficiency gains. The next section outlines a number of losses associated with the compressor and the electromechanical system. Afterward the losses have been considered flow related effects are examined. Finally, compressor design related effects are touched upon, but not discussed in detail, as low volume consumer has little control of the compressor design other than selection from the available units. As such, compressor design is beyond the scope of this project.

First consider a conventional single speed design. Small compressors are almost always of the hermetic variety where the compressor and motor are housed in a single package. The compressor motor pairing is typically designed to run at one frequency. Generally the compressor size will be adjusted to provide a good impedance match to the mechanical system, but this can only occur at one pressure/flow combination. The motor compressor system will also sometimes be compromised to allow easier start-up. Providing variable frequency drive circumvents these limitations. Another compromise to consider is the dual frequency ratings of many single-phase motors. The phase shifted winding in the motor will typically have a single sweet spot for the external run capacitor. The capacitor selection will typically be a compromise that allows operation at both 50 and 60 Hz, albeit at decreased efficiency. The capacitor will also be optimized in value for the nominal motor loading. Maximum efficiency requires continuous tuning of the capacitor value in a single phase system.

To run the compressor at variable speeds, the motor is driven by a drive inverter. There are losses associated with the generation of the variable frequency signal and also losses associated with operating the motor at speeds other than it’s nominal speed. System losses need to be qualified to gain a sense of where the efficiency peaks will be found.

3.7.2 Core Losses vs. Drive Frequency

Practical magnetic materials exhibit hysteresis in their magnetization curves. As a result a small amount of the energy used to magnetize a material is not recovered when the material is demagnetized. The power loss due to this effect is dependent on the depth of magnetization and the number of magnetization/demagnetization cycles per unit time. Please see Figure 23.

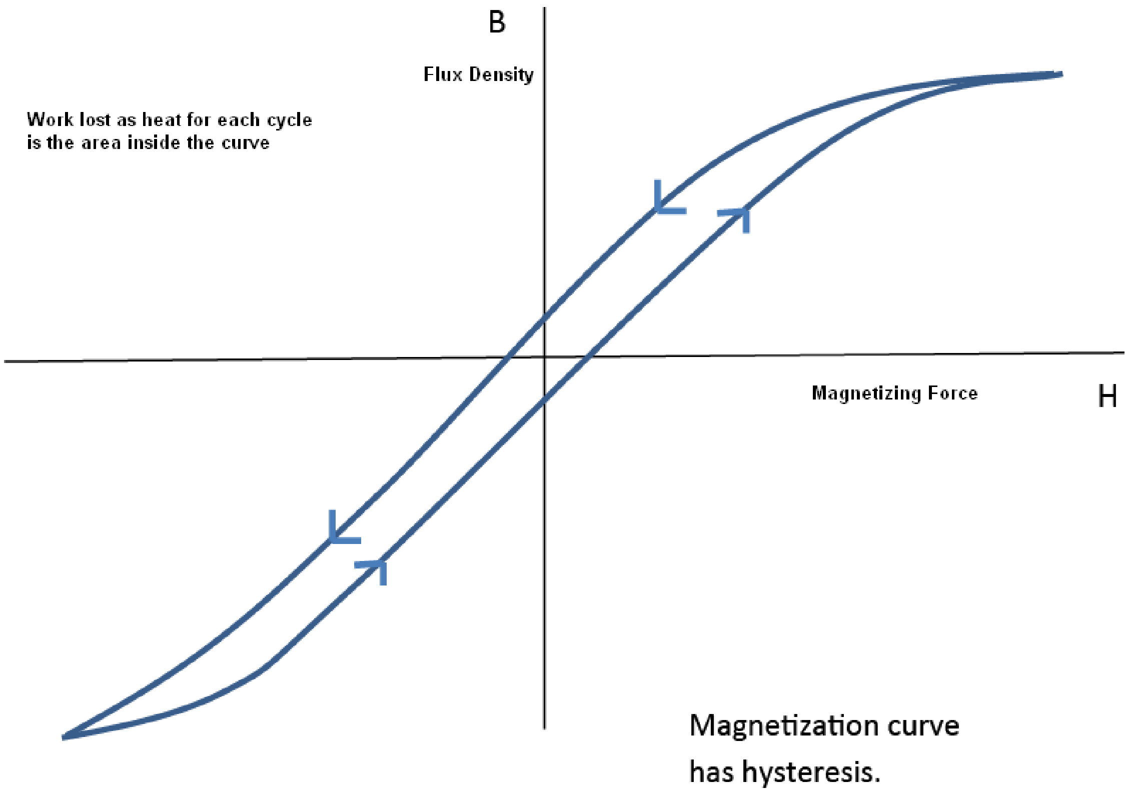


Figure 23 – Magnetization Losses

The loss vs. frequency curve is linear over a short range of frequencies, they then increases rapidly if the core material is electrically conductive. The non-linear increase with frequency is largely due to eddy current losses in the material. If the physical size of individual conducting pieces (in the core) is too large[14], they begin to act like a shorted turn in a transformer and dissipate power in the resistance of the material. To prevent this from happening, cores are frequently made of individually insulated laminations. At nominal operating frequencies the eddy current losses are minimal. Unfortunately, there is a break point where the eddy current losses will increase rapidly with frequency. Core losses due to the repeated magnetization/demagnetization cycles are mostly predictable,

[14] Most magnetic materials that support high flux densities are also electrically conductive.

except for a second order effect that comes from self-heating. Material losses are a function of core temperature and this function normally has an inflection point. A well designed magnetic core will be sized such that the self-losses heat the core to the maximum efficiency point.

3.7.3 Core Losses vs. Carrier Frequency

Core losses vs. carrier frequency are usually not a large contributor to total losses. The drive inverter generates a variable frequency output digitally so as to reduce the ohmic losses within the inverter. The incoming line voltage is rectified, then run through a gang of switches. The switches are modulated at a higher carrier frequency to generate the desired signal. Most drive inverters do not include a reconstruction filter to extract only the low frequency signal at the output. This is done for cost and efficiency reasons as the typical application has the inverter connected to a motor where the winding inductance provides smoothing of the signal anyway. The side effect of synthesising a low frequency sine wave in this fashion is many modulated high frequency components at multiples of the carrier frequency. While the high frequency components detract from system efficiency, there is not much efficiency gained from filtering them, as the filter parts would decrease efficiency. The high frequency components don't tend to induce much current flow in the motor anyway due to the inductance of the coils. The only benefit to reducing the high frequency components is a reduction in EMI, both for regulatory reasons and to prevent self-interference.

3.7.4 Copper Losses vs. Frequency

Motor losses due to copper losses are due to I^2R heating. As drive frequency increases the effective copper area decreases due to the skin effect[15]. This effect is quite mild at our

[15] High frequency currents tend to confined themselves near the surface of a conductor.

frequency of interest. The copper winding losses in our system are dominated by straight DC resistance as there is little difference in skin effect with double digit frequencies.

3.7.5 Friction vs. Drive Frequency

At very low frequencies the motor bearing friction will be very high, as the fluid film will not have been established in the journal bearings. It is not wise to operate at low speeds for prolonged periods, as the bearing wear is far too high under boundary lubrication conditions. The friction under these conditions does not need to be considered; as such low speeds cannot be sustained. Once hydrodynamic lubrication is established, the frictional losses drop dramatically and increase linearly with speed. Power loss due to fluid friction is approximately a quadratic function of speed in this application. As the oil warms and thins, there is a decrease in friction, but less than would be the case for plain oil as the refrigerant solubility in the oil also decreases with temperature. The second order effects again make efficiency predictions inaccurate and empirical measurements more valuable.

To gain a rough sense of the speed required for hydrodynamic lubrication, the compressor was run with inlet and outlet disconnected. The motor input power was observed while increasing the drive speed. At roughly 20 Hz there was a drop in motor power consumption. Whether the decrease in consumption was due to the transition to hydrodynamic lubrication or due to the centrifugal oil pump priming could not be determined without destroying the compressor. A minimum compressor speed of 45Hz was selected.

3.7.6 Volumetric Efficiency vs. Drive Frequency

The volumetric efficiency of the compressor changes with drive frequency. Re-expansion losses are not being considered in this design, as the compressor selected is a scroll design rather than reciprocating piston. The effects of leakage between the scrolls, however, need to be considered. The selected compressor has a compliant scroll design. As such, the scrolls move through an orbit that generates slight interference of the scroll pairs (when at speed). The stationary scroll is compliantly mounted to prevent destruction of the compressor. Because of the interference fit between the scrolls, the compressor's volumetric efficiency is near 100%, as long as sufficient speed is enough to cause mating of the scroll surfaces. The compressor design also depends on outlet to inlet differential pressure to allow mating of the scrolls in the axial direction. This is done deliberately to decrease motor starting loads. Restarting a HVAC system that has been recently run but briefly interrupted can result in a large pressure differential across the compressor. The motor starting circuit may not have sufficient capacity to start rotation, causing motor failure due to high starting currents. The selected compressor is designed to unload the compressor scrolls to facilitate safe start up without having to allow system pressure equalization. These two features cause a large decrease in volumetric efficiency below a minimum speed threshold. The minimum operating speed of the compressor and the minimum start-up ramp rates are constrained by the need for the scrolls to mate. If the scrolls spin too slowly, they will not wobble enough to mate. Also, if the start-up acceleration is too gentle, there will not be enough of a pressure rise to force the scrolls together along their axis, which in turn prevents the pressure rise needed to seal the scrolls.

The scrolls of the selected compressor were found to seal effectively at drive frequencies above 25 Hz with minimum frequency[16] ramp rates of 5 Hz/s. Prior to scroll mating, the volumetric efficiency is below 25%. Once mating has occurred, the volumetric efficiency ramps to nearly 100%. The first observation was made with the compressor pumping air to find the threshold of operation. The second observation is based on capacity estimates.

[16] Earlier tests with lower ramp rates yielded a higher minimum frequency for mating

3.7.7 Inverter Effects at Reduced Voltage

When the drive frequency is modified, the motor voltage must be altered as well. At low running speeds, the voltage must be reduced to prevent motor core saturation and winding damage. The drive inverter does not have constant efficiency as a function of its output voltage. The drive inverter's losses are of three types. 1) Constant losses associated with supervisory functions and overhead 2) Switching losses 3) ohmic losses. The constant losses are most prominent when operating at low output powers. This typically coincides with low output voltages associated with minimum operating speeds. Switching losses are typically linear with output current and carrier frequency. Switching losses are the unavoidable packets of energy that are dissipated as heat during the imperfect act of turning on or off the switches in the inverter. The linear relationship with frequency is expected as the number of cycles per second dictates the number of times the losses occur per second. The linear relationship with current comes from the increase in stored charge in the switching element that increases as the current density increases. Ohmic losses have a quadratic to cubic function of current. They are due to the resistances in circuit elements. Ideally the ohmic losses are I^2R and quadratic. Self-heating, however, will sometimes increase R and cause losses to increase. These three sources of loss in the drive inverter cause the efficiency vs. power curve to be shaped like an inverted bathtub. The efficiency vs. voltage curve takes on a similar shape, with exaggeration at the low and high ends of the curve due to the more than linear increase in power at higher voltages.

3.7.8 Supplemental Heat Losses

In a hermetically sealed compressor design, compressor oil is exposed to refrigerant. The compressor oil is specifically selected such that it is soluble in the refrigerant. This choice is important because the refrigerant must be capable of returning any discharged compressor oil to the compressor by washing it out of the piping and heat exchangers. This causes significant amounts of refrigerant to accumulate in the compressor oil during off periods. The dissolved refrigerant can cause lubrication failure through its thinning effect on the compressor oil. To prevent this, a heater is usually added around the base of the compressor. The amount of supplemental heat needed to purge the compressor oil of refrigerant is dependent on both the ambient temperature and the compressor operating power. As the motor dissipation increases, less supplement heat is needed. The waste heat from the motor is sufficient to keep the oil warm. As ambient temperatures drop, more supplemental heat is needed to offset the losses to the environment.

Figure 24 shows the compressor supplemental heat vs. ambient temperature required to maintain a shell temperature 15C above ambient. The data was found empirically.

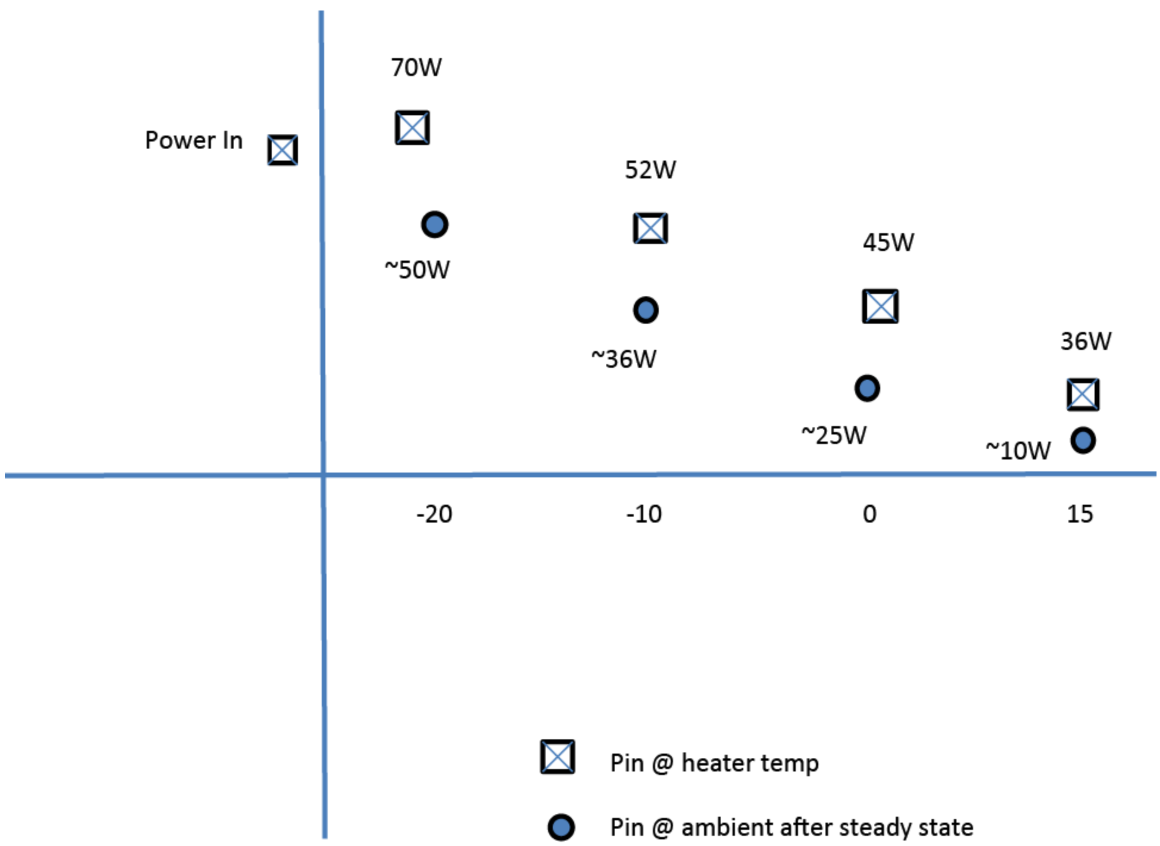


Figure 24 – Supplemental Compressor Crankcase Heat Curve

3.7.9 Compressor Speed Re-Optimization for R290

The selected compressor was originally designed for R22 service. The heat pump was charged using R290, which has a lower operating pressure at a given temperature. The lower operating pressure results in decreased motor torque and current draw at nominal speed. Because the motor is driven from a variable frequency drive inverter, it can be operated at decreased voltages at nominal speed or, at elevated speeds and nominal voltage. The former allows lower power consumption when reduced capacity is sufficient and the latter allows higher capacity on demand. These two modes are possible because the drive inverter converts from one frequency and voltage to another reactively. As a result, reduced voltages at the output result in a commensurate current reduction at the input. Baring losses due to imperfect efficiency in the conversion process, power out equals power in. This allows power savings by reducing the compressor voltage, load permitting, without using resistive losses to drop the voltage. There is also the opportunity to achieve overall efficiency gains by running a smaller displacement compressor at higher speeds as needed. If sized correctly to the load, the decreased frictional losses of the smaller compressor result in power savings without loss of peak capacity, as there is always the option of running the compressor faster when needed. In the results section, measurements of efficiency vs. compressor speed are tabulated.

3.7.10 Compressor Cooling Optimization

Because the compressor is of a hermetic design, most of the motor cooling is provided by the incoming refrigerant. In heating mode, the waste heat from the motor is not squandered as it eventually makes its way to the indoor heat exchanger. In cooling mode, some of the outdoor heat exchanger capacity is used to dissipate this heat, but there is usually surplus cooling capacity in systems large enough to satisfy the heating requirements. During defrost mode, the additional heat helps speed the defrost cycle.

While it is tempting, for cooling purposes, to allow some liquid refrigerant to enter the compressor suction line this is not advisable. The liquid refrigerant would be both very effective at removing heat from the motor and reduce the degree of super heat at the compressor's[17] inlet. It would, however, also thin the compressor oil and likely cause cavitation in the compressor bearings. Most hermetic scroll compressor manufacturers advise against continuous liquid flow at the inlet[18]. This design regulates the inlet superheat very close to zero, but does not deliberately introduce slugs of liquid refrigerant at the compressor inlet.

3.7.11 Pipe Sizing

Piping in a heat pump system must be carefully sized to accommodate the pressure drops in the lines and to maintain sufficient flow velocity. This design, for reasons that will be explained presently, is less sensitive to piping sizing than a conventional heat pump design.

Consider first the effects of piping size on pressure drops in an HVAC system. Smaller piping in either the liquid line or the vapour line result in larger pressure drops across the line. In the liquid line this can result in a loss of effective sub-cooling by the end of the line. In heating mode the pressure at the inlet of the liquid line is set by the condensation temperature. The liquid line typically runs indoors for the majority of its length. The liquid in the line will continue to cool somewhat as it travels towards the outside unit, but its final temperature will never be lower than room temperature. The pressure of the liquid in the line will, however, continue to drop along the length of the line. If the line is too narrow for its length, the resulting pressure drop can lower the refrigerant boiling point to the point of bubble formation. Bubbles in the liquid line cause difficulties with accurate refrigerant metering. This can cause problems with system capacity, durability

[17] This refers to the super heat at the scrolls' inlet, not the compressor housing inlet

[18] There are some scroll compressors with a vapour injection port intended for economizer cycle use. They tolerate limited injection into the scrolls, but it should be noted that the motor assembly itself is not subject to liquid flow into its lubricant.

and control stability. Pressure drops in an undersized vapour line cause loss of system capacity and efficiency by increasing compressor discharge pressure during heating and lowering compressor suction pressures during cooling. The loss of efficiency during heating is of more concerning as this cannot be easily compensated for with additional refrigerant flow. System capacity is also most taxed during heat mode.

Two other related effects, and a third which is unrelated, must be considered when selecting piping sizes in a conventional heat pump design. Compressor flooding can occur with too much pipe volume (either too large or too long) and refrigerant distribution can difficult to adjust with long lines. With large volume lines, refrigerant can flood the compressor during the off periods, if the accumulator before the compressor is too small. This is because the refrigerant will tend to migrate from the warm interior of the building to the cold outdoor unit. Transient refrigerant distribution can also be problematic if the ratio of line volume to heat exchanger volume is large. During the start-up transient pockets of refrigerant gas may be present in the liquid line. These pockets must travel through the refrigeration loop before normal refrigerant levels are established in the heat exchangers. Large lines allow large pockets to form, which in turn cause larger deviation from the optimal refrigerant levels in other components. The previous two effects were related to refrigerant flow. The third effect has to do with oil flow. While larger diameter piping is good for limiting pressure drops, they are not good for returning oil to the compressor. A minimum flow velocity is needed to ensure that the compressor lubricating oil is returned to the compressor.

The proposed heat pump design is much less sensitive to piping because it has a variable capacity compressor and has secondary means of providing liquid line sub-cooling. Variable compressor speed allows the use of smaller lines as the compressor speed can be reduced when outdoor temperatures are higher. Decreasing the compressor speed can offset the increase in evaporation pressure at warm ambient temperatures. This allows a smaller piping diameter to be used as compared to a fixed speed design. The reverse is true at low temperatures. It is safe to over-speed the compressor at low ambient temperatures, as the mass flow through the pipes remains reasonable despite the higher compressor speeds. On the liquid line side, slightly more pressure drop is permissible without efficiency or durability concerns. The pressure drop in the high-side line makes little difference to system efficiency as additional pressure drop in the lines can be subtracted from the pressure drop across the expansion device. This is only true when there is sufficient sub-cooling of the liquid to prevent bubbling before the expansion device. Because of the additional sub-coolers in this design, additional pressure loss in the lines can be tolerated by decreasing the pressure drop across the expansion device to compensate.

Should an installation require the use of large diameter vapour lines, the oil return problem can be easily avoided by forcing periodic scheduled high speed operation of the compressor.

3.7.12 Defrost Cycle Losses

Virtually all air source heat pumps require a periodic defrost cycle to remove frost accumulations from the outdoor heat exchanger coils. The energy used for the defrost cycle is typically taken from the interior of the building. The defrost cycle decreases both the system efficiency and system capacity. Several optimizations are incorporated into this heat pump design to decrease the efficiency losses due to defrost cycles.
A traditional heat pump often uses a time/temperature based defrost cycle. In some more advanced traditional heat pump designs, the current defrost cycle is modified based on the duration of the previous cycle. In both cases, a timer counts down when the coil temperature and/or ambient temperature is near or below freezing. The timer starting value is usually dependant on a combination of coil and ambient temperatures. Near-freezing temperatures usually cause the most coil frosting, as the ambient air is both cold and moisture laden. Very cold and warm conditions do not tend to cause as much frosting as there is either little moisture content in the air or the air is too warm to permit freezing on the coils. Consequently, the timer is set for a shorter period at conditions near freezing. When the timer expires, the heat pump is reversed and the outdoor fan is

disabled. After a set period of time, or when the outdoor coil has reached a temperature target, normal operation is resumed. During the defrost cycle, a bank of resistive heaters is normally enabled to prevent cold air from blowing into the occupied space.

This heat pump design modifies the defrost cycle in several ways. First, the defrost cycle is triggered by the accumulation of frost on the coils, not based on an estimate of the running time needed to accumulate frost on the coils. The degree of frost accumulation on the coils is measured by observing the air pressure drop across the coil when the outdoor fan is running. This eliminates many unnecessary defrost cycles. The decrease in defrost cycle frequency will be discussed in the result section.

Secondly, the efficiency of the defrost cycle heat source is improved. At moderate outdoor temperatures, the auxiliary heat source is inhibited during defrost. This way the cost of the energy used for the defrost cycle is less. Rather than using resistively generated heat for the cycle, heat that was brought into the house at a COP greater than unity is used. As the ambient temperature drops, there comes a breakeven point where it is better to enable some auxiliary heat during defrost rather than decrease the system capacity and increase heating cycle times. For this reason, the auxiliary heat inhibit during defrost is controlled by the outside temperature.

Thirdly, the compressor speed is optimized for maximum defrost efficiency. During defrost, the rate at which heat can be transferred to the outside coil has an effect on defrost efficiency. Slow heat transfer during defrost is wasteful as more of the heat is lost to heating the outside air. This design optimizes defrost efficiency by running the compressor at the highest speed possible, regardless of the set-point during heating, during a defrost cycle. In doing so, a larger percentage of the heat transferred goes to melting the coil frost instead of heating the outside air.

A fourth method used to improve efficiency is pulsed outdoor fan operation. The traditional heat pump turns off the outdoor fan for the duration of the defrost cycle. After the frost has been melted not all of the water falls from the coil. The water that remains on the coil is removed by evaporation. As expected, the energy associated with the drying process can be substantial. This heat pump design pulses the outdoor fan after melting the frost accumulation. A portion of the water is removed from the coils mechanically so less energy is expended drying the coils.

Another design optimization included is a synchronized defrost option. When several heat pumps are operating close together, the vapour from a single defrosting unit can rapidly ice the coils on nearby pumps. Synchronized defrost allows several heat pumps to defrost simultaneously to avoid capturing freshly expelled moisture.

All of the above optimizations help to decrease the detrimental effects of defrost on cycle efficiency. It should be mentioned that the loss of efficiency during defrost extends to a period after defrost has occurred. The act of reversing the heat pump for defrost produces a transient similar to the initial pump start up transient. Even after heating mode has resumed, the pump's heat output remains low for several minutes. This is why it is critical that both the frequency and duration of defrost be minimized

The last system level optimization used to improve efficiency is continuous compressor speed throttling. Most commercial heat pumps operate the compressor at one speed. Some premium units incorporate a two speed compressor using a switchable number of poles in the motor (through external selection of different windings). While there are commercial heat pumps with variable compressor capacity, they typically cost more than $10K and are not being used for comparison due to the price point. This design was optimized to compete at a price point slightly above that of an entry-level unit. A conventional one or two speed design requires cycling of the heat pump to regulate the indoor temperature. Continuous throttling allows the compressor capacity to exactly match the heating or cooling demand. As a result the compressor does not need to cycle to regulate the indoor temperature. The lack of cycling avoids operating the heat pump in an inefficient manner right after compressor start up.

The magnitude of the efficiency loss due to cycling can be estimated by logging the transient start-up behaviour and then comparing it against the thermostat cycle length. A casual observation of a single speed heat pump showed a 12 min transient period where the heat pump was not producing its rated heat output while still consuming nearly rated power. This design with liquid injection active had a start-up transient almost 2 times faster. A typical residential grade thermostat will have a heat pump setting that calls for 1 to 3 cycles per hour. Most come pre-set for 3 cycles per hour to minimize customer complaints about comfort. With a 20 min cycle there is a heavy efficiency hit with a conventional design, and a moderate penalty with this modified design.

With continuous throttling, the electrical consumption of the compressor, per unit heat transferred, is decreased. There is however an energy usage penalty due to the continuous operation of the fan motors. The results section will more closely compare the efficiencies in both modes.

A side benefit of continuous throttling is decreased wear of the compressor. Wear is maximized during compressor start-up and shutdown due to lack of lubrication at start-up and the loss of lubrication just before compressor stoppage. Wear due to lubricant dilution is also reduced as the compressor spends less time cold.

Chapter 5 contains the experimental validation results. In order to validate the design presented in this chapter a series of experiments were designed. The experiments were designed to 1) measure the heat pump output and efficiency under steady state and transient conditions 2) observe the performance during defrost and 3) calibrate the measurement system.

Additional validation for engineering purposes (component evaluation, reverse engineering tests etc.) was done during assembly of the prototype, but much of it is ancillary to the design and not covered in detail in this document.

Chapter 4

THEORY AND COMPUTATION

4.0 Theory and Computation

A heat pump is essentially a heat engine in reverse. Instead of thermal energy flowing from a heat source to a cold sink with work extracted, work input is used to move heat from a cold source to hot sink. The heat pump design will have a COP limit bounded by the reverse Carnot cycle. This efficiency can never be achieved due to practical limitations, but it still serves as a check to ensure nothing has been omitted in the estimates. Ultimately, the heat pump's COP is Qh/Win (heat out/ work in). If this number exceeds the value predicted by the reversed Carnot cycle, there is problem with the data. Please note that as the COP limits are developed, there will be a re-definition of "work in" which shifts it from mechanical work to electrical energy at the input of the motor, then the drive inverter. This is done because the figure of merit needed to make comparisons against other energy sources must align with the source of energy used in the design. The end user of this heat pump does not care how much mechanical shaft work there is, rather they are interested in the electricity costs and hence a COP figure referenced to electrical input. Other than in the initial parts of this chapter, COP will refer to Heat Out vs. Electrical Energy In.

First, the Carnot cycle assumes a reversible process, which would constrain the system capacity to zero in any realizable design[19]. Next, both heat sink and cold source temperatures are increased and decreased, respectively, in order to achieve any meaningful heat transfer. This is to account for the temperature differences between the working fluids and the environment needed to cause heat transfer. Finally, the work done

[19] Reality has not the patience of thought experiments and will not suffer infinitely slow process. We also lack the mythical frictionless surfaces, resistance free substances and mass-less strings which are so popular in textbooks. In essence mother-nature will not be cheated.

during expansion is mostly lost as friction in the design. There are, of course, ways of recovering this inefficiency, but not at the price point or scale of this design.

First consider the Carnot cycle efficiency[20].

$\eta = W/Q_H = 1 - T_C/T_H$

The efficiency figure listed above is for a regular heat engine. Heat is taken from the heat source at T_H and disposed of at a cool sink at T_C. Work W is extracted along the way. The cycle, being ideal and hence reversible can therefore be inverted to find the COP limit for a heat pump. Heat is now removed from a cold source and disposed of in a heat sink. Work becomes and input rather than an output, and the figure of merit, the COP, is Q_H/W or the inverse of efficiency.

Rearranging the equation gives $COP = T_H/(T_H-T_C)$

Pumping heat from –15C to 20C yields a COP limit of 8.37.

Our COP goal of 4 appears to be a modest goal, until practical factor are included. The thermal reservoir temperatures should not be used for the estimate as the condensation and evaporation temperatures govern the pressures the compressor will be subject to. Condensation cannot occur at the target room temperature, likewise evaporation will not occur at the outdoor temperature. Condensation must occur slightly above room temperature and evaporation must occur below the outdoor ambient temperature. Most budget heat pumps are sized such that the outdoor evaporation temperature is 10K below ambient at rated capacity. A 5 Ton heat pump heat exchanger was selected for this design, but run at 2-3 Ton capacity. This raises the evaporation temperature to roughly 5K below ambient. In practice, the figure is closer to ambient, due to other design modifications, but for the purpose of this analysis 5K is a conservative starting point. The indoor heat exchanger is similarly scaled and achieves a condensation temperature of

20 Image of equation from Wikipedia

10K above ambient. This is consistent with many commercial designs, and will provide a good figure of merit for comparison. It should be noted that this analysis omits the degradation to efficiency caused by the power consumption of the indoor fan motor. Increasing the indoor airflow improves the refrigeration cycle efficiency at the expense of fan power. There is actually an optimal fan power that trades off refrigeration cycle efficiency and fan power consumption. Because of this, there are conditions where the expected rise in efficiency at high outdoor temperatures does not occur. With the condensation temperature raised to 30C and evaporation decreased to –20C, the new machine COP limit drops to 6.06.

So for our COP figure has assumed that work input is at the compressor shaft using a perfect compressor. An electrical motor, fed by a drive inverter, drives the compressor. The selected compressor operates at 83% efficiency (as specified from electrical energy input to compressed R22) when optimally loaded. The figure provided by the manufacturer is a composite number that includes the inefficiencies in the compressor itself combined with the losses in the motor. The COP limit referenced at the motor's input drops to 5.03. Factoring in 95% efficiency for the drive inverter, the theoretical limit (now referenced at the power inlet) falls to 4.78.

The real refrigeration cycle is, of course, less efficient than this. This design does not recover the work done during expansion and there is friction in the pipes. On the hot side, frictional losses in the pipes can be ignored and the loss of work recovery during expansion has been factored in. The frictional losses in the pipes result in a decrease in the pressure drop across the TXV, thus keeping the total hot side to low side pressure the same. Some of that work, albeit a very small amount, is actually recovered as heat, conducted through the pipes, to the building interior. Post evaporation pressure drops are small enough to be ignored in this design as the pipes are oversized and short. This design conveniently provides significant sub-cooling before the TXV. Estimating the work loss to expansion in a normal heat pump is involved as one must also consider the volume change that happens in the expansion device. In the augmented heat pump there is little conversion to gas during expansion as the liquid has been cooled before the pressure drop. The work loss during expansion is decreased as there is substantially less volume change in the expansion device, limiting the work loss to the product of the pressure drop and liquid flow rate.

Calculation of the new efficiency based on the loss of expansion work is not trivial due to the effects of compressing and expanding a real working fluid. The simplest method to estimate the new efficiency is to use a Mollier chart for the selected refrigerant. This method allows us to directly consider the factors of interest: the desired heat output and work input. A summation of the inputs and outputs while making a round of the Mollier chart provides an initial estimate that can then be corrected to account for the modifications to the refrigeration cycle. The estimate is simplified by only considering work in for compression and useable heat out.

The augmented heat pump regulates the compressor inlet super heat to zero. At –20C the saturated vapour line places specific enthalpy at 550 kJ/kg. Perfect compression results in an enthalpy rise to 610 kJ/kg. De-superheating and condensation at 30C to the saturated liquid line drops the working fluid enthalpy to 275 kJ/kg. The estimated COP, factoring in the inefficiencies of the compressor and inverter is 4.4 = (610-275)/(610-550)*0.83*0.95 [21]. For a normal heat pump expansion would happen after slight sub-cooling, resulting in roughly 30% flash gas in the evaporator. In the modified design, this figure is reduced to about 5%. The main effect ends up being an increase in evaporator capacity along with more subtle increases in efficiency (due to raised evaporation pressure) and non-subtle effects in coil icing performance.

[21] Compressor efficiency as specified at the electrical input and inverter efficiency at full load

Chapter 5

EXPERIMENTAL VALIDATION

5.0 Test Environment

The heat pump performance tests were run over the winter and spring months using outdoor air as the heat source. An indoor facility would have been preferable, but an environmental chamber of sufficient capacity was not available for these purposes. As a result, testing had to be done at the available outdoor temperatures. During the testing period, the heat pump was still under construction. Because of this, not all of the tests were conducted at all of the temperatures. In addition, the total duration of each test was limited by the available indoor space to be heated. The indoor temperature rise limited the tests that could be conducted. The maximum experiment duration was limited to roughly $1/10^{th}$ of the estimated thermal time constant for the structure. This time constant was estimated roughly by cycling the building's heating system. Ho, Hayes and Wood [10] describe a model for a floor slab hydronic heating system. They show estimates for the transfer function from slab heating to air temperatures along with correction factors for thermal mass of structure not related to slab. While their system has more thermal mass than the test structure, it gives a rough estimate for the expected time constants and validation of the test area's time thermal constant. The process of a structure heating the enclosed air was assumed to be reciprocal (at least to first order). The table below outlines the sequence of construction along with the testing that was done.

Table 6 – Test Schedule Summary

Time Period	Construction Completion	Tests
Late Fall	Outdoor Unit Assembly	Compressor Run Test
Early Winter	Indoor Unit Connection + Indoor Wiring + Portable Manhole Ventilator Fan For	Leakage Tests, Initial Heat Pump Performance Tests Without Automatic Liquid Injection

	Indoor Coil	Data Collection Was Done Manually
Mid Winter	Indoor Heat Exchanger Shroud Built + Proper Duct Fan Installation	Multi Speed Tests, Liquid Injection Automation. Data Collection Was Done Manually
Late Winter	Defrost Controller Added	Multi Speed Tests, Liquid Injection Tests, Initial Defrost Controller Tuning Data Collection Was Automated Heat Output Measurement Calibrated
Early Spring		Defrost Performance Tested

The early tests were done with manual temperature rise measurements using a dual probe k-type thermocouple gauge. These were used to estimate relative heating outputs vs compressor speed. The data collected during the early measurements is difficult to correlate from day to day as the ducting available at the time was based on a flex hose shape that changed from day to day. When testing liquid injection manually, the injection on and injection off tests were done as close together as possible to minimize the effects other changes. The difficulty with this sort of testing is the sensitivity to the liquid volume of injected liquid. Small variations in the injected volume can cause substantial disturbances to the operating heat pump. Without automatic controls, the collected data was noisy. The early tests were useful to gain a sense of the effect. Controlling liquid injection manually, however, was quite a difficult task. Automation of this task, along with automated data logging, extracted much more useful data during later testing. Figure 25 shows the test setup used for early winter tests.

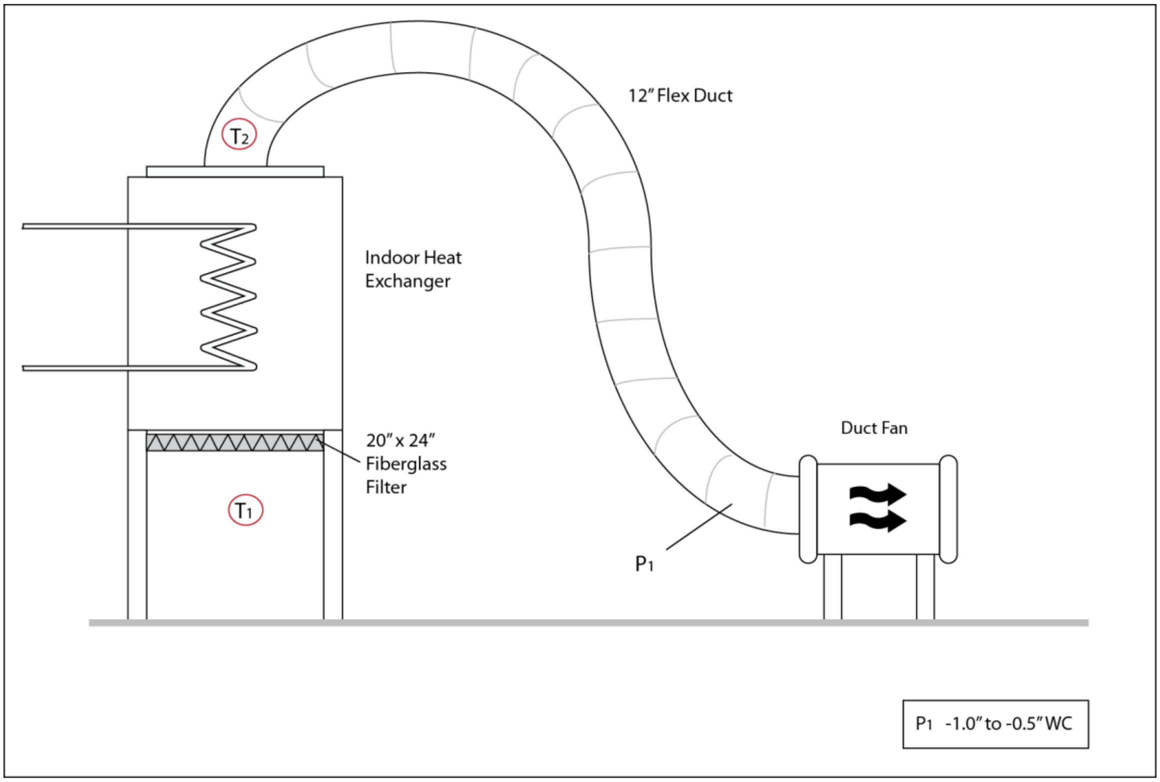


Figure 25 – First Winter Test Setup

Calibration of the heat output was not done for the earlier tests, as there was too much variation in airflow from day to day with the initial setup. Based on the pressure at P1, the flow rate was estimated at around 1000 CFM, but not measured. No flow meter was available for this purpose and the setup had not yet been fitted with a resistance heater for flow calibration purposes. After the new ducting and fan were installed, airflow was much more stable. The flat plate used, below measurement point T2, on the old setup was replaced with a better diffuser that decreased the variation in airflow caused by the initial bend in the ducting. Please see Figure 26

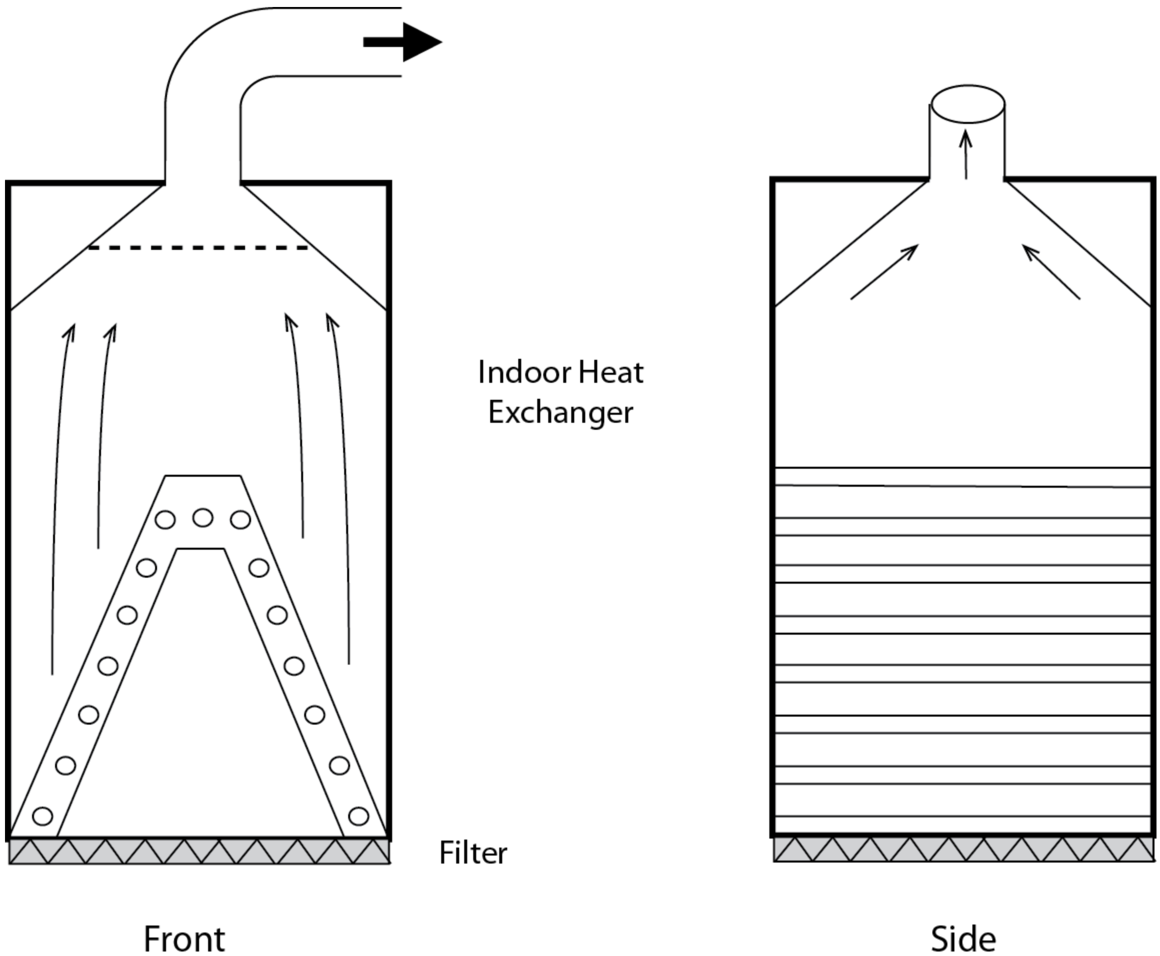


Figure 26 – New diffuser

With the new setup it was possible to calibrate the airflow by inserting a resistance heater in the air stream and measuring the air temperature rise. Temperatures were measured below the heater and at the centre of narrowest portion of the round duct. The resistance heater was the same width as the heat exchanger and it was hoped that there would be a sufficiently even heat distribution at the input to make the outlet temperature measurement insensitive to position. Re-measurement of the setup showed no measurable variation from the middle to the outer portion of the pipe. The author attributes this to either satisfactory mixing, or good spatial averaging of the observed temperature by the metallic thermocouple leads in the airflow. Automation of the liquid injection system and data collection also substantially decreased the noise present in the collected data.

Figure 27 shows the test setup used for the remaining tests.

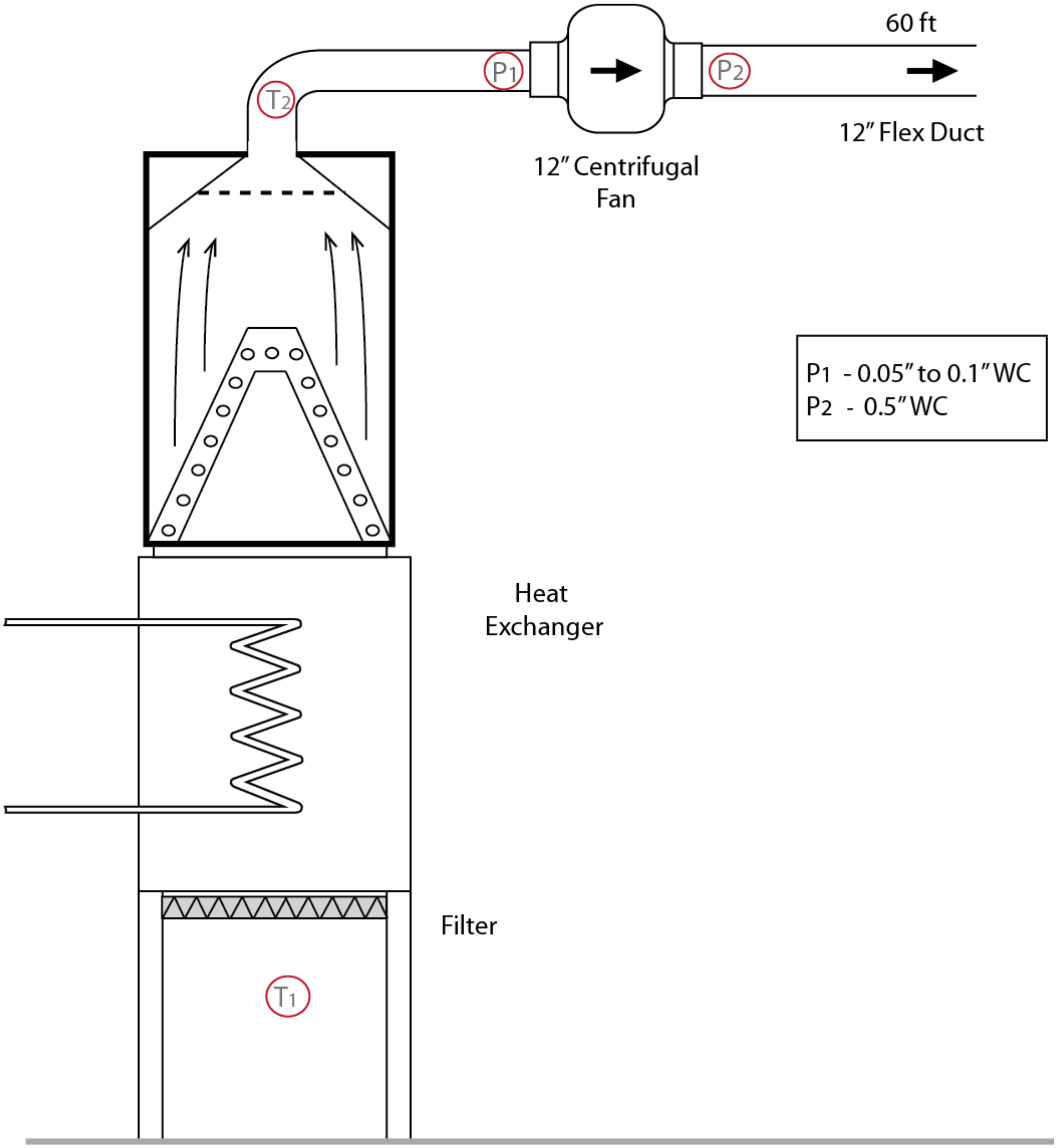


Figure 27 – Mid winter test setup improvement

5.1 Early Setup Results

The following series of graphs summarize data obtained on the first test setup. The X axis is the compressor drive frequency. Figure of merit is measured in an arbitrary unit of Temperature Rise / Input Power. For the reasons previously mentioned, the efficiency numbers were not converted to an absolute unit. There were a few measurements where it was suspected that the system had not settled. A second measurement was taken for these points to see if there was a shift in performance with additional dwell time. These points show up as a double measurement in the graphs.

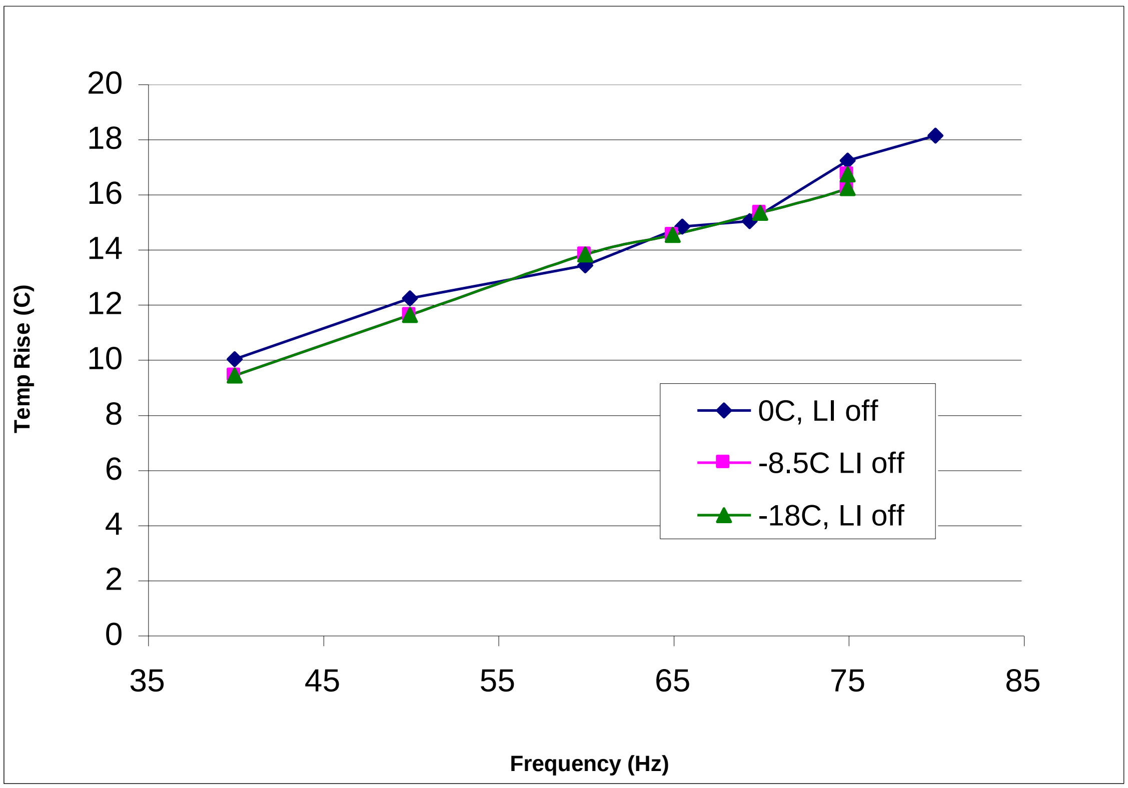


Figure 28 – Temperature Rise vs. Drive Frequency Curves for Various Ambient Temperatures (Liquid Injection Off)

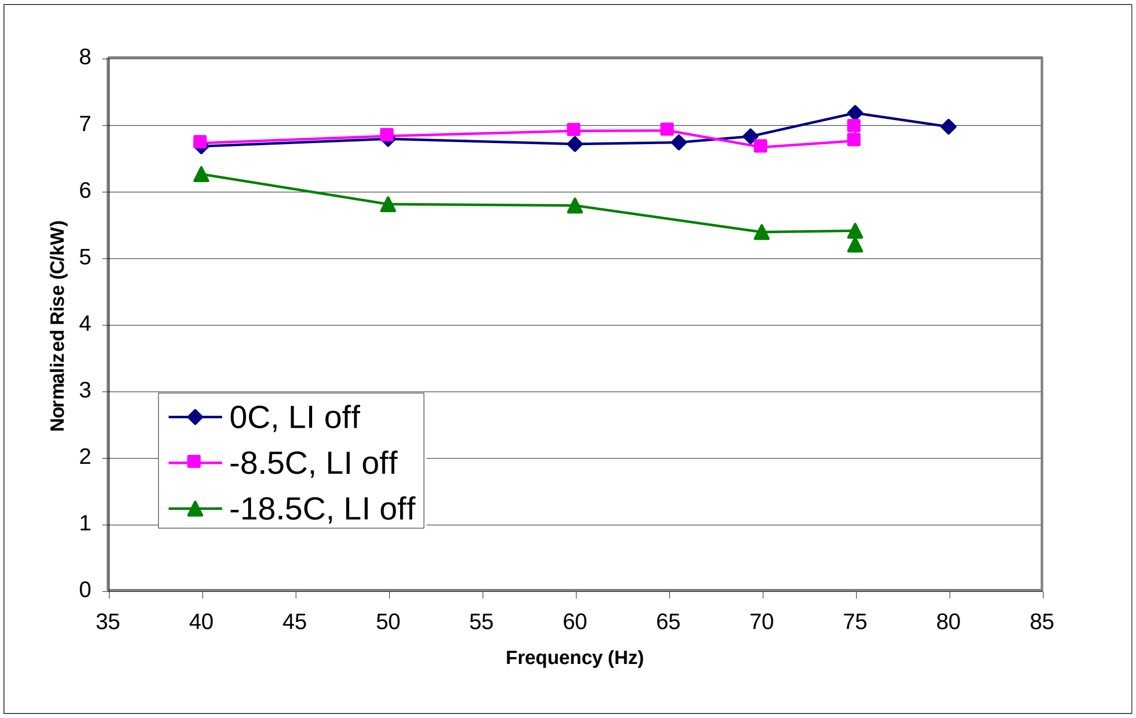


Figure 29 – Normalized Temperature Rise vs. Drive Frequency at Various Ambient Temperatures

As expected there was an increase in temperature rise as drive frequency was increased. The linearity of the curves suggests that flow and capacity were only being limited by compressor flow and not by the evaporator or condenser sizes.

The next two graphs showing the difference between liquid injection and non-injected operation represent data taken at 0C ambient.

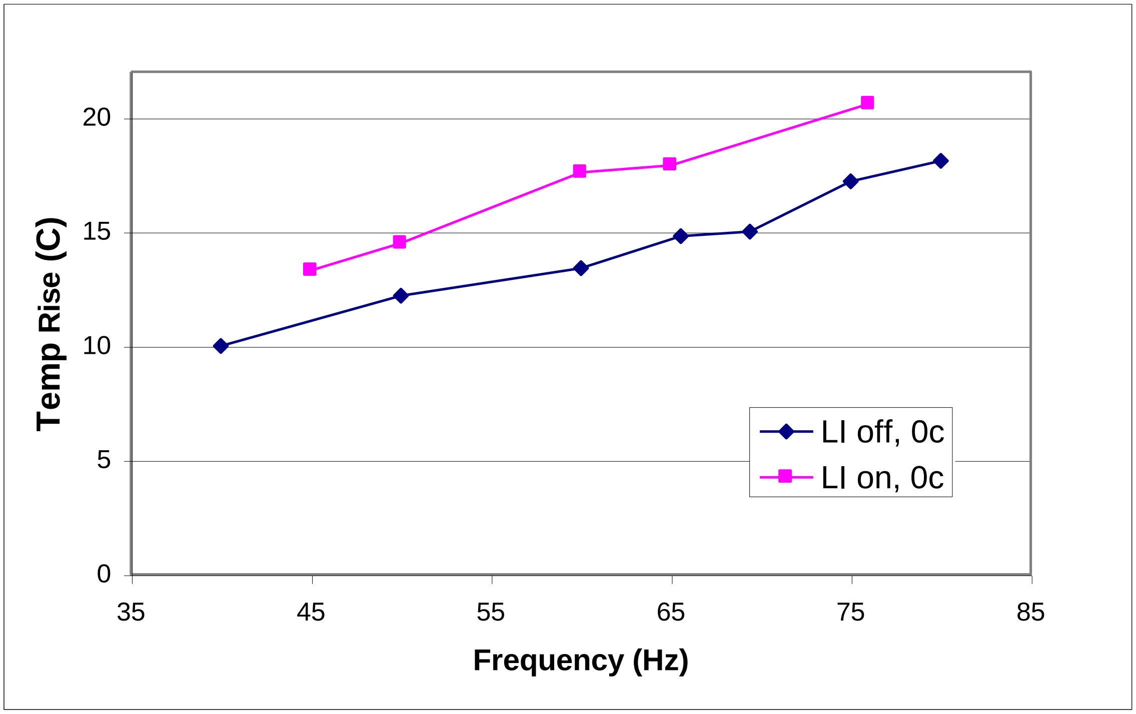


Figure 30 – Liquid Injection Effect on Temperature Rise-Frequency Curve at 0°C Ambient

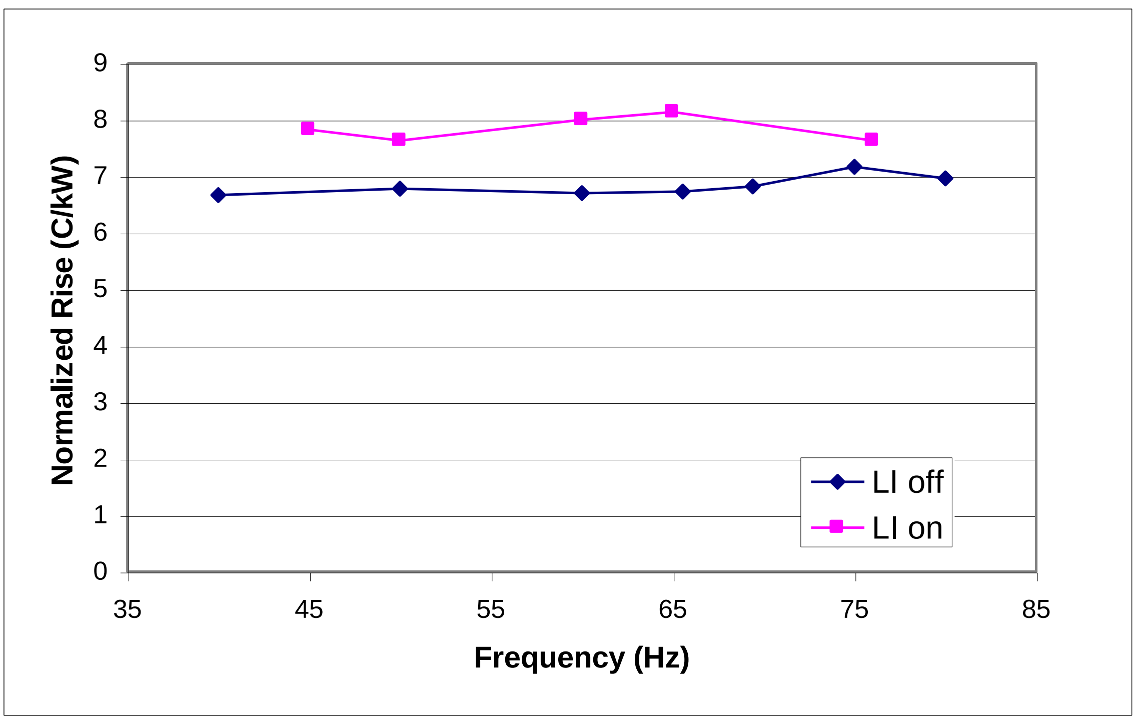


Figure 31 – Liquid Injection Effect – Normalized Temperature Rise

The aforementioned tests were run with fairly variable indoor and outdoor conditions. The manual logging also posed problems with the data collection.

The improvement in capacity and figure of merit can be seen. There was roughly a 15% increase in capacity, which is in line with the predicted (based on flow needed to sub-cool) 17% diversion of flow through the liquid injection system while injection is on.

An additional test was run using the same setup to determine if increased time averaging of the data would yield cleaner data. A day where the outdoor temperature was very stable at 0C was selected. The goal of the test was to obtain a more accurate estimate of the efficiency vs drive frequency curve. The intent was to use this data as an input to the next series of tests to be run on the newer setup being built.

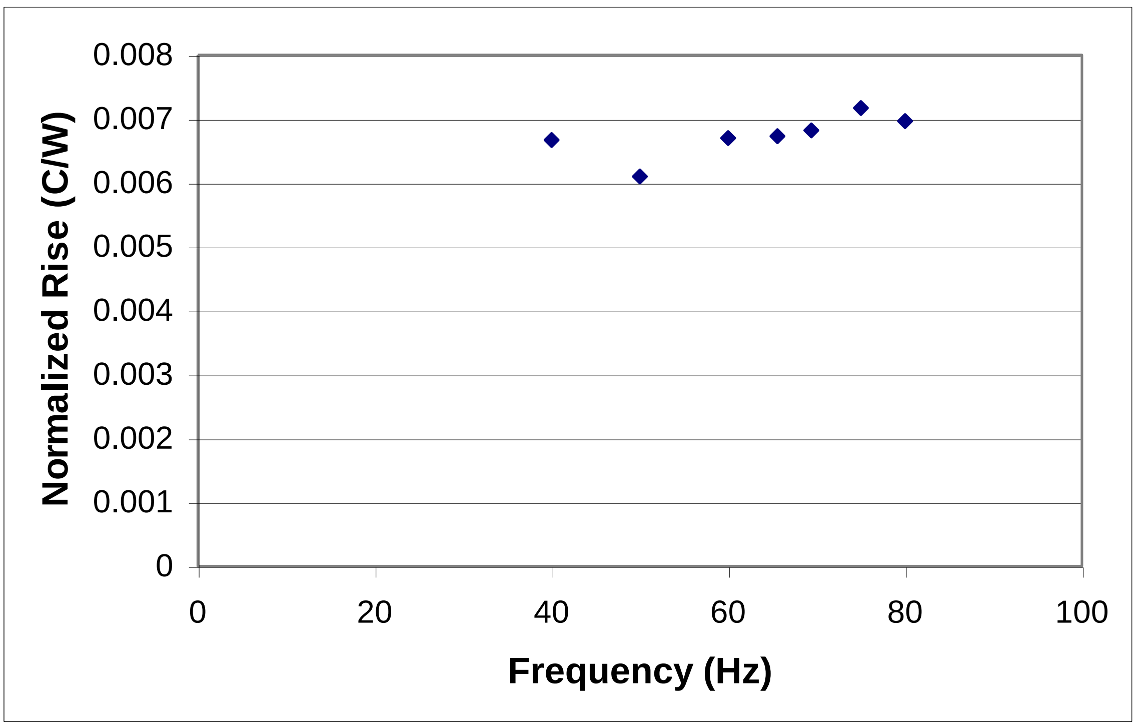


Figure 32 – Temperature Rise vs. Frequency Curve (0C Ambient)

The time-averaged data yielded only slightly cleaner numbers. The datum at 40Hz was assumed to be erroneous due to settling effects as the test started with a quick ramp to

40Hz followed by a series of steps till 80Hz was achieved. More time was not allocated to the first measurement, as the test would not have completed without significantly changing the indoor temperature before completing the last measurement. This was not considered to be a problem as the true peak efficiency point was expected to be located above 60 Hz for a R22 optimized compressor in a R290 system[22].

The first automated test using the new ducting setup was used to get an initial estimate of the system performance with liquid injection off, on and at several compressor speeds.

The new setup combined with automated data logging allowed meaningful error estimates.

5.2 Error Estimates

The estimation of COP was done by calibrating the temperature rise at a fixed air flow through the heat exchanger and by observing the reported input power from the drive inverter.

The system temperature rise calibration was done using an external resistive heater and two thermocouples. The thermocouples were re-used for normal measurements to offset any systemic errors introduced by the temperature measurements. Input power to the resistive heater was measured using a clamp on current meter and a voltmeter. The rated accuracy for both instruments is 1%, resulting in a 1.4% error for the observed power.

The reported power has 1% measurement accuracy, but this only applies to full-scale readings. The display quantization error introduces 2% error (half the reported least significant digit step size). The two error sources are assumed to be independent of each other as the former is a scale factor for the instrument while the latter is random quantization error. The two errors were combined in quadrature to give 2.2%.

[22] Due to the lower operating pressures for R290 vs R22

The temperature rise measurement error and the power reporting error were assumed to be independent. The resulting error in the COP estimate is 2.6% or 0.11 assuming a mean COP of 4.25

5.3 First Logged Test

The first tests were performed in the same location, using upgraded ducting and automated data logging. The newly added logging hardware allowed automated temperature and power consumption measurements.

The test sequence, whose results are plotted below, was as follows:

- 65Hz compressor speed with liquid injection off
- shutdown to allow pressure equalization
- 65Hz compressor speed with liquid injection on
- ramp to 70Hz
- ramp to 76Hz
- compressor off and resistance heater on

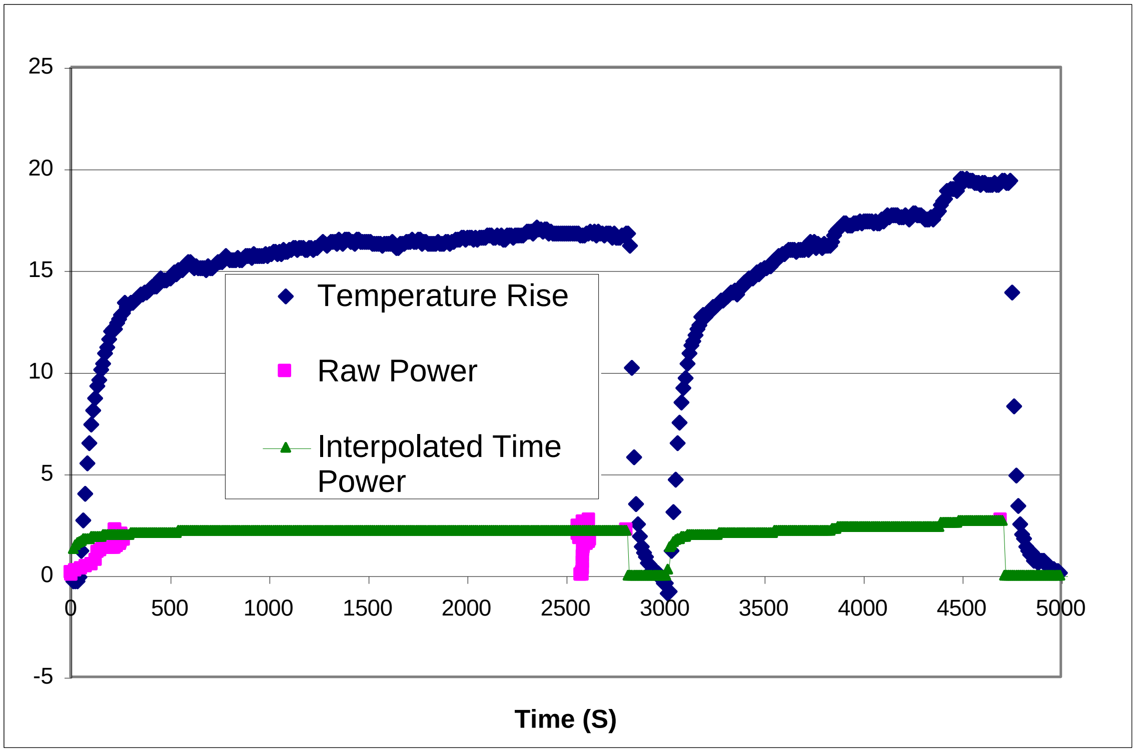


Figure 33 – Variable Speed Logged Test

The data corresponding to COP was extracted for the 70Hz and 76Hz portions of the test. The 65Hz data was not processed. An efficiency peak was not expected at that frequency (based on earlier data collection) and there was an upward trend in efficiency shifting from 70 to 76 Hz operation. The outdoor temperature was roughly 1.5C for these tests. The duration of the test was not sufficient to reach final steady state as the indoor temperature was climbing too rapidly to continue testing. The estimated steady state COP was 4.25 to 4.5, but this could not be verified due to excessive indoor heating.

The graph for COP at 76 Hz operation shows a number of large discontinuities in the data in addition to a slight trend. The COP figures were processed from the datalogged power and temperature rise readings. Calibration of the heat output was done using a resistance heater as described previously. The large discontinuity around 100s corresponds to a slug of gas passing through the liquid line. This behaviour is associated with a transient shortage of refrigerant that can happen shortly after the compressor speed has been stepped. During the period immediately after a step in speed, the refrigerant in the liquid line can be depleted before increase flow from the condenser is established. The thermal mass of the condenser is responsible for this phenomenon. Immediately after a speed step, the pressure in the condenser does not rise as it is held constant by condensation. The rise in condensation temperature (and pressure) is delayed by the thermal mass of the condenser. Until the pressure rises, there can sometimes be a period when there is insufficient flow through the TXV to keep up with the increased evaporator demand. The downward trend of the graph is associated with the overall time constant of the system. The heat pump has a cyclic drift with a 5-10 min time constant. This is associated with the previously mentioned thermal mass and its effect on the system pressures. The behaviour does eventually settle if the compressor speed is held constant.

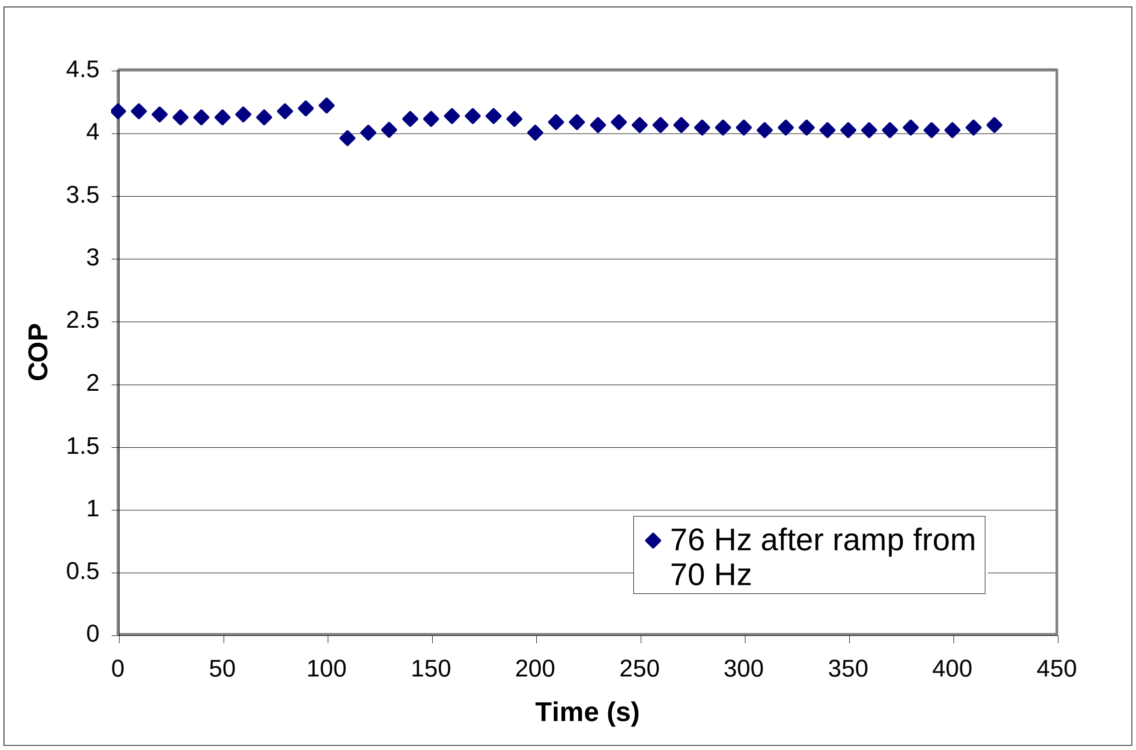


Figure 34 – COP at 76Hz

The restart mid tested provided a convenient means to compare the transient response with and without liquid injection enabled.

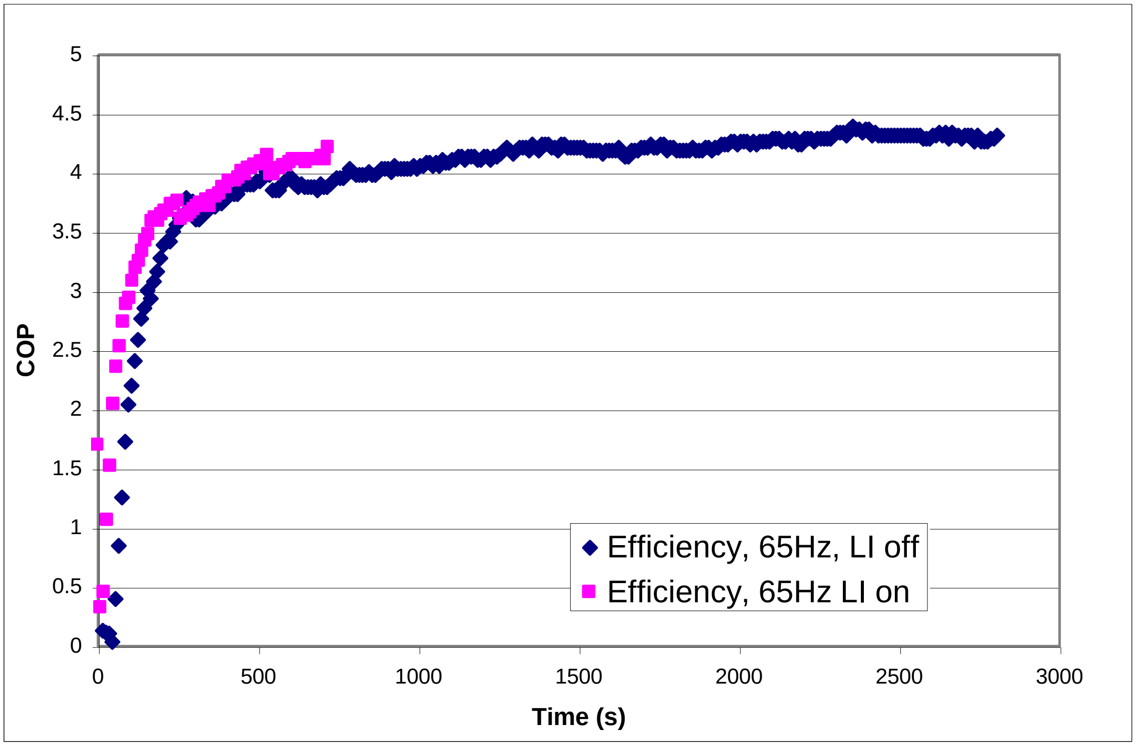


Figure 35 – Transient COP at 65Hz

The most evident improvement in performance with liquid injection active is the speedup of the system transient response. The gains in capacity are not visible as the data above is heating output divided by power input. The gains in capacity are accompanied by an increase in input power, albeit to a lesser extent.

The next test was performed at 10C outdoor temperature. At this temperature the compressor speed was limited as it was uncertain if there would be sufficient cooling to operate the compressor above its nameplate rating. The compressor would likely have survived, but it was too early in the test cycle to be abusing the compressor in this fashion.

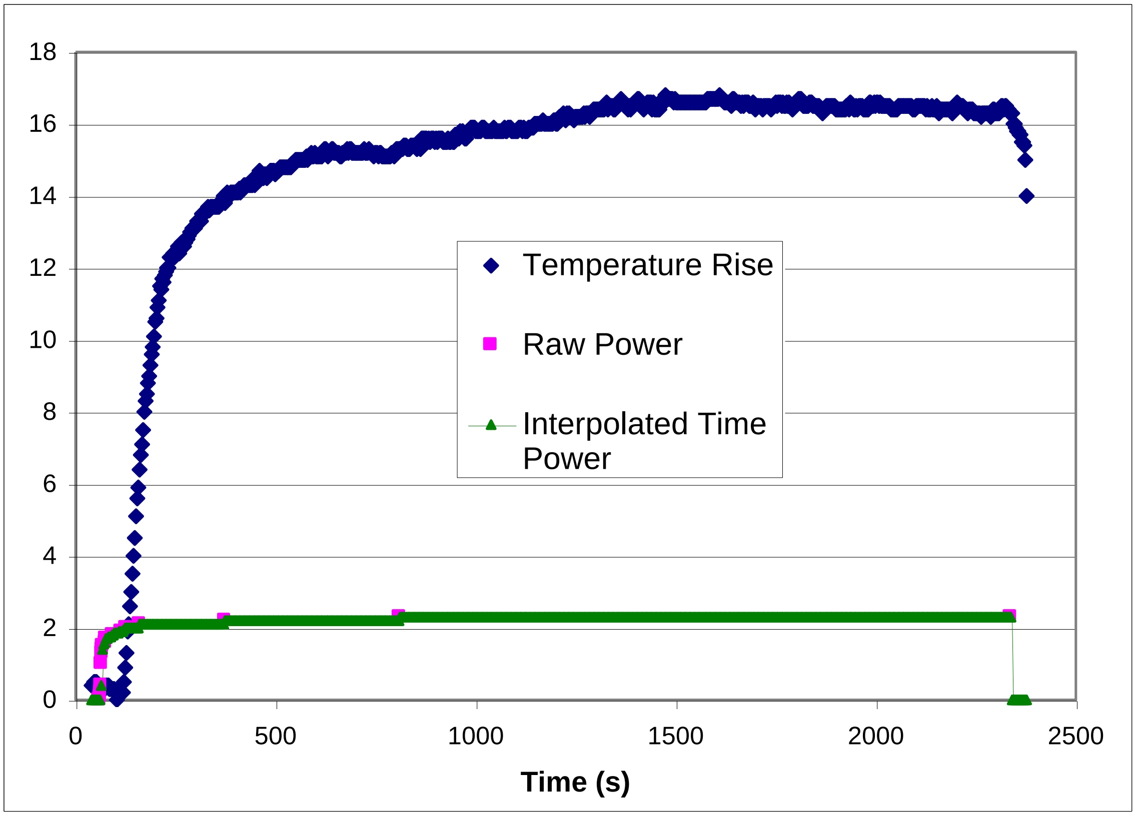


Figure 36 – Variable Speed Logged Test at High Ambient (10°C)

The heat output of the system was already excessive at 60Hz operation[23]. The compressor was definitely running below maximum efficiency speed. As expected the heat pump efficiency did improve upon the observed efficiencies at lower ambient temperatures. With additional safe guards it might have been possible to operate the heat pump at higher drive frequencies, but given the equipment and test setup limitation at the time, an over-speed test was judged too risky to perform.

[23] The building heating loads are quite modest at 10C outdoor temperatures. Even at 60Hz, the test could not be run for very long without overheating the building.

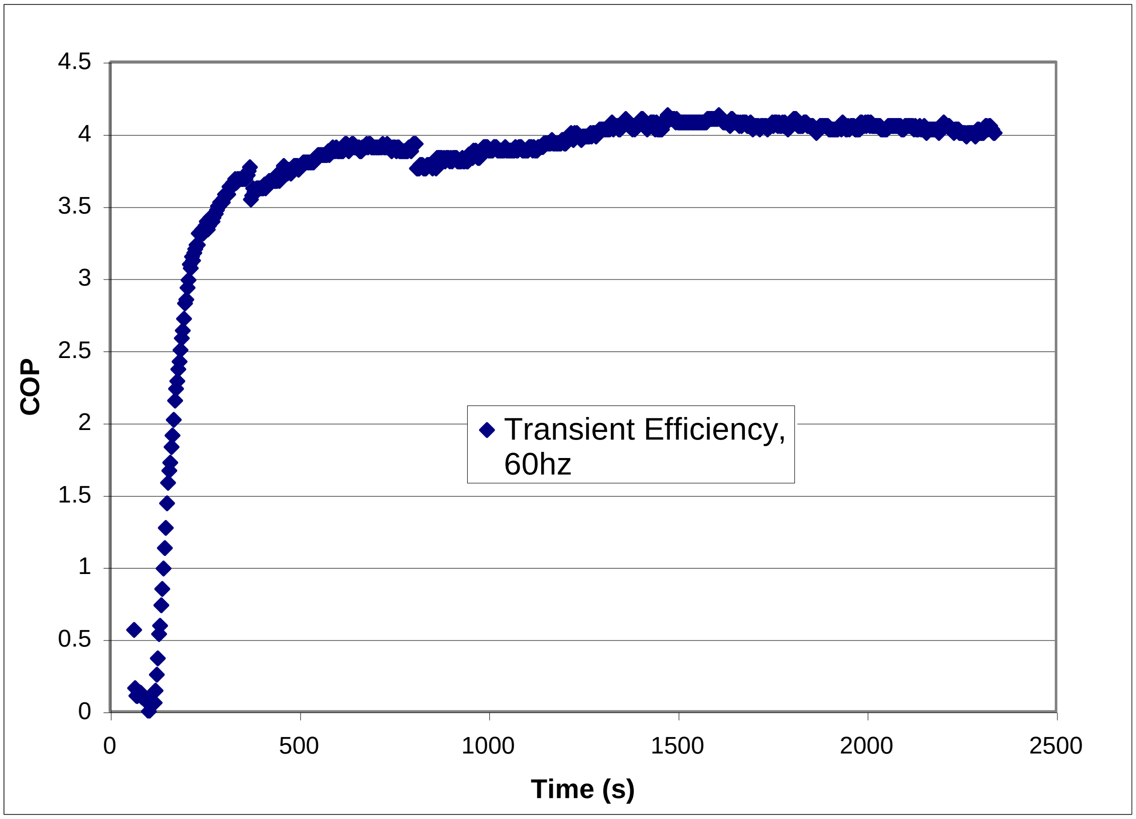


Figure 37 – Transient Efficiency at 60Hz at 10°C Ambient

Spikes in the COP curve can be observed. They correspond to the cycling of the injection solenoid. Control of the injection solenoid has significant hysteresis to avoid undue wear of the valve. The control of hysteresis was not fine enough due to limitations of the drive inverter's hardware. A future design would incorporate a higher resolution controller.

5.4 Defrost Performance

The next set of results shows the frost up performance of the heat pump. The test was run at 1.5C ambient during wet snow conditions. This was deemed to be one of the worst conditions for frosting. The evaporator would be below freezing and there would be an ample supply of moisture in the air. The compressor was run at 50, 60 then 70Hz. The goal of the test was to ramp up the compressor speed in the hopes that the outdoor coil temperature would dip below the dew point and accumulate frost. Light frost started accumulating at the end of the 70Hz run. With no real means to measure the amount of frost accumulated, a second conventional heat pump nearby was run to provide a qualitative visual reference.

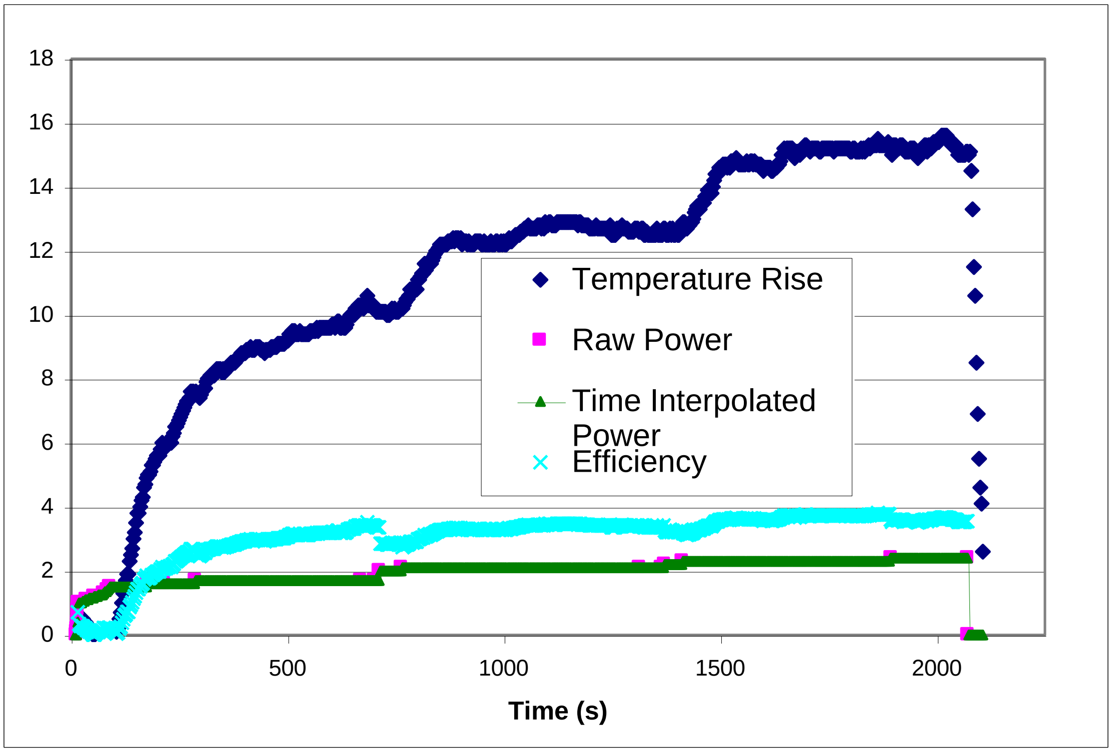


Figure 38 – Frost Up Performance Characteristics at 1.5°C Ambient

The next picture shows the modified heat pump running next to a conventional heat pump. Both pumps started frost-free and ran for 30 minutes.

Photo 1 - High efficiency heat pump (left). Commercial Carrier heat pump (right)

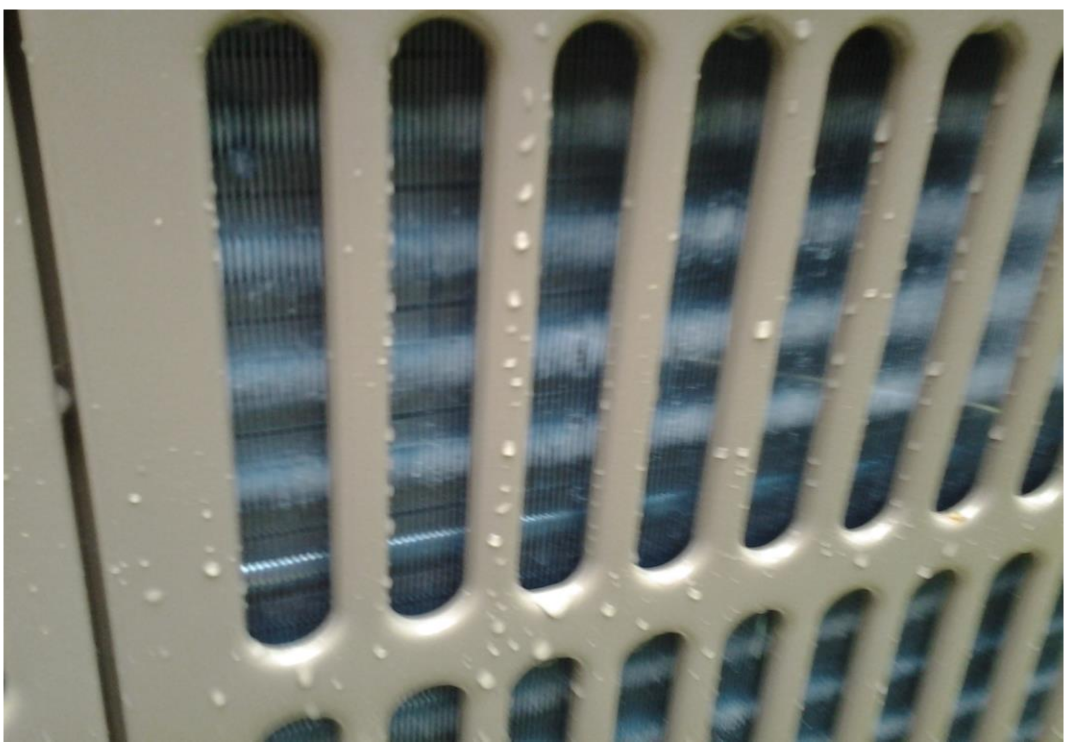


Photo 2 - High efficiency heat pump, frost accumulation at end of test.

The defrost controller had been tuned in the days earlier. With no real way to measure the volume of frost accumulation, time-to-defrost measurements were recorded. On days where little frosting would normally occur (cold and dry such that there was little moisture available) all defrost cycles were inhibited. During moderate days (RH 50-70% and just below freezing) defrost cycles would occur every 2-3 hours. On humid days just above freezing, defrost cycles were roughly 1 hour apart, but only occurred when the compressor was run at speeds in excess of what was needed to balance the building heat load.

The difference in defrost performance between the two pumps is most apparent on dry days. The augmented heat pump eliminates more than 90% of the defrost cycles as compared to a commercial timer based defrost algorithm.

The next test shows the ill effects of a refrigerant starved system. A small portion of the refrigerant was removed from the heat pump in an effort to see if a quicker transient response could be achieved. The decreased refrigerant charged caused the formation of bubbles in the liquid line that interfered with the metering device's operation. The compressor speed started at 70 Hz then ramped down to 65, 60 and finally 55Hz.

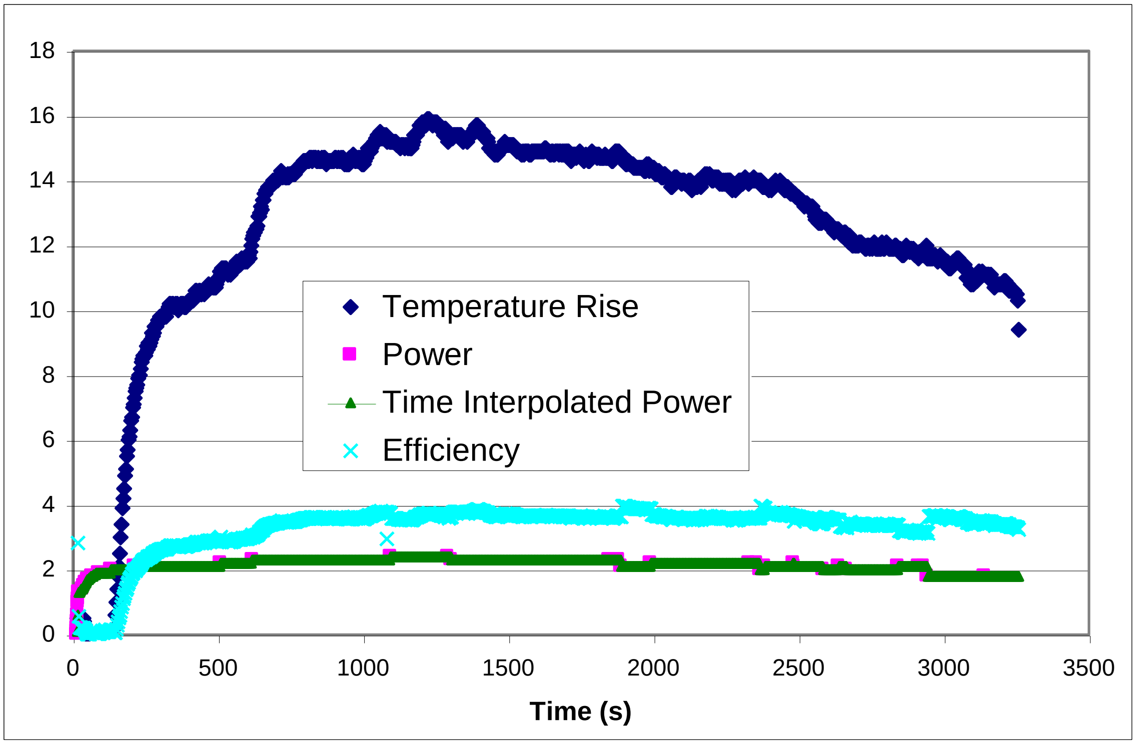


Figure 39 – Characteristics of a Refrigerant Starved System

The compressor was run at 70Hz till 1400 seconds. The lack of refrigerant prevented the liquid injection system and TXV from functioning properly. At about 700 seconds the liquid line was bubble free as sufficient refrigerant had been withdrawn from the evaporator and accumulators and moved to the high pressure side. From 1400 about 1800 seconds the compressor was run at 65 Hz. Between 1800 and 2300 seconds the compressor was run at 60 Hz, then slowed to 55 Hz after 2300 seconds. Operating the compressor at 55 Hz with proper refrigerant flow was nearly as effective as 70Hz operation with sub-optimal refrigerant metering. The sensitivity to correct refrigerant charge in the system is important to consider as the metering system becomes more complex. This design is more tolerant of overcharging than under-charging as there are free volumes where refrigerant can safely accumulate, but a metering system that can be upset by disruptions in the liquid refrigerant supply.

5.5 Continuous vs. Duty Cycle Operation

Heat pumps and air conditioners are less efficient when cycled. The loss of efficiency due to cycling can be quite dramatic if the control thermostat is programmed for short cycles, as is frequently done to improve occupant comfort. With a conventional design, cycling is inevitable as capacity is controlled in this fashion. To make a fair comparison against a system in cyclic duty, the delivered heat must be made equal and run times adjusted accordingly. The pairs of graphs below show the time integral of temperature rise (essentially a scaled cumulative capacity) and mean efficiency as a function of run time. Consider a 30 min cycle where the equivalent of 50% capacity is needed[24]. The continuous system would generate 15000 C*S in a 30 min period. This corresponds to roughly 1100 seconds run time for the cyclic system. The cumulative efficiency at this point is ~ 3.5 vs. 4.3. In both cases the same amount of heat has been transferred, but the COP of the cyclic system is substantially lower. The discrepancy gets even worse at higher ambient temperatures as the required heat transfer drops and results in shorter on times.

In the series of graphs to follow, the Effective COP values were calculated using measured efficiencies vs. time. The times averaged COP, up to the time indicated in the X-axis, are graphed. The beginning of each graph inflects downwards as a short time must elapse before sufficient high side pressure is developed. The high side pressure is needed to operate the pilot mechanism in the 4 way valve. Without this pressure the valving sets the heat pump in cooling mode, degrading the COP figure. The purpose of the graphs is to estimate the COP of the heat pump given a run-time and show the degradation in performance due to cyclic operation.

[24] This is roughly representative of an appropriately sized system operating near freezing.

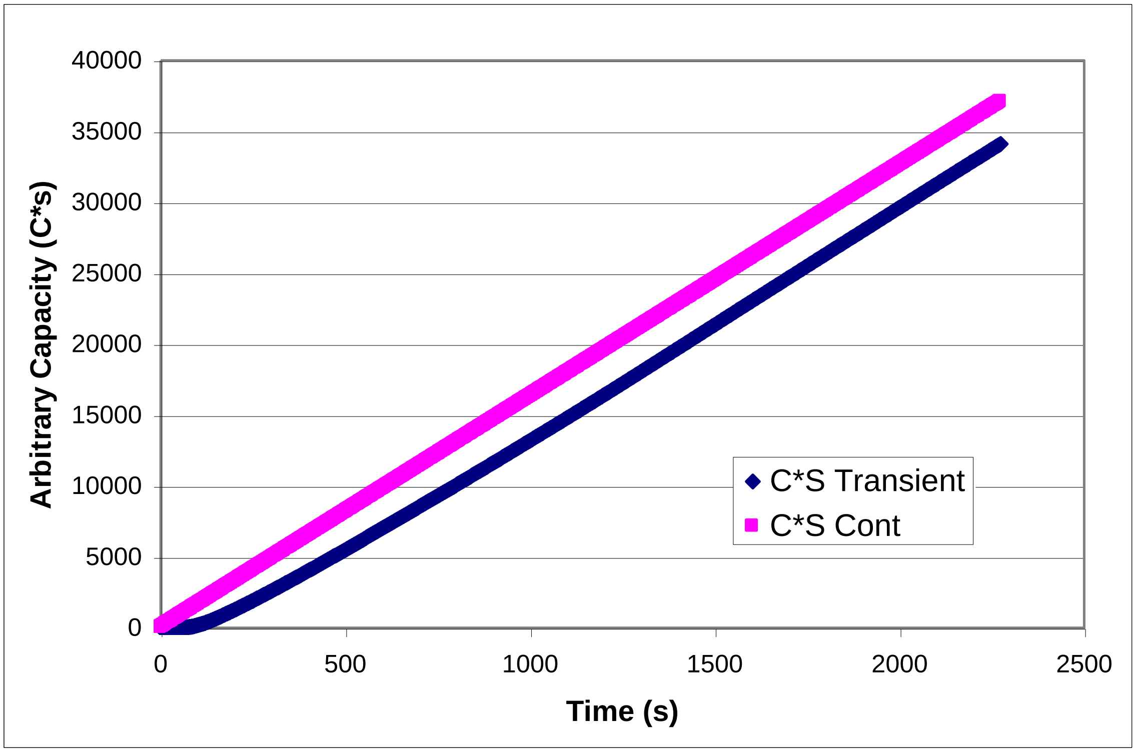
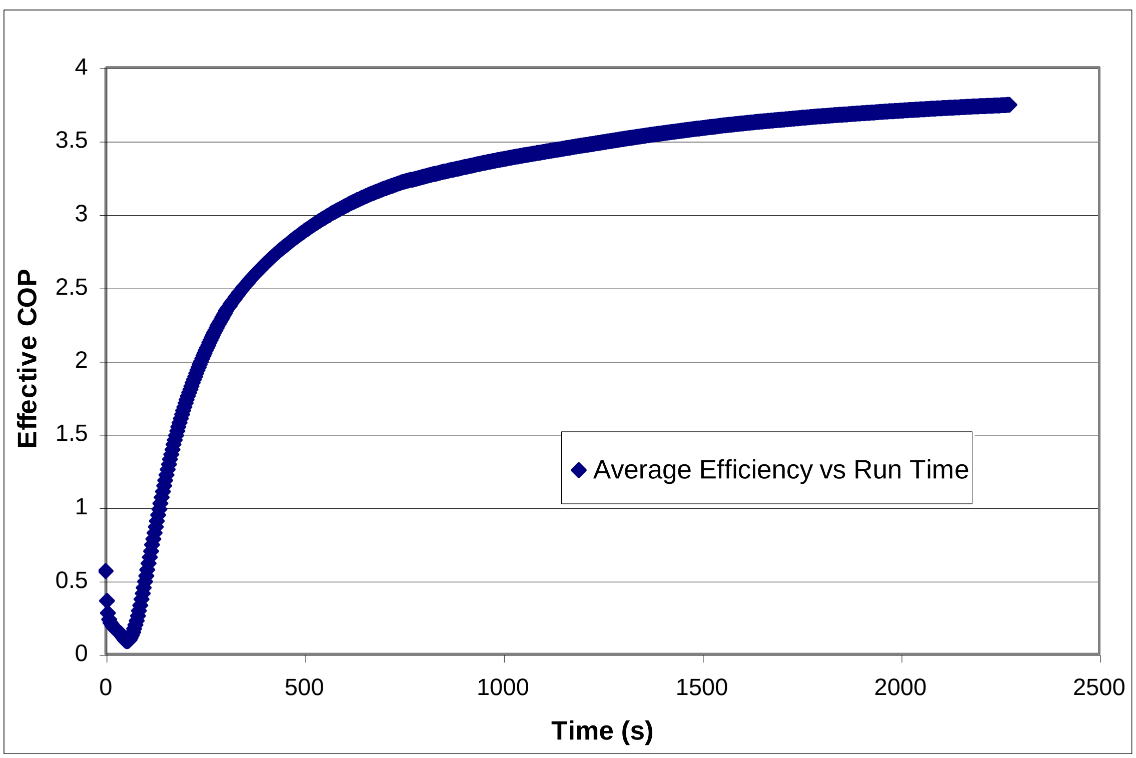


Figure40 – Cumulative Capacity as Function of Run Time at 60Hz, 10°C Ambient



Figures 41 –Mean COP as Function of Run Time at 60Hz, 10°C Ambient

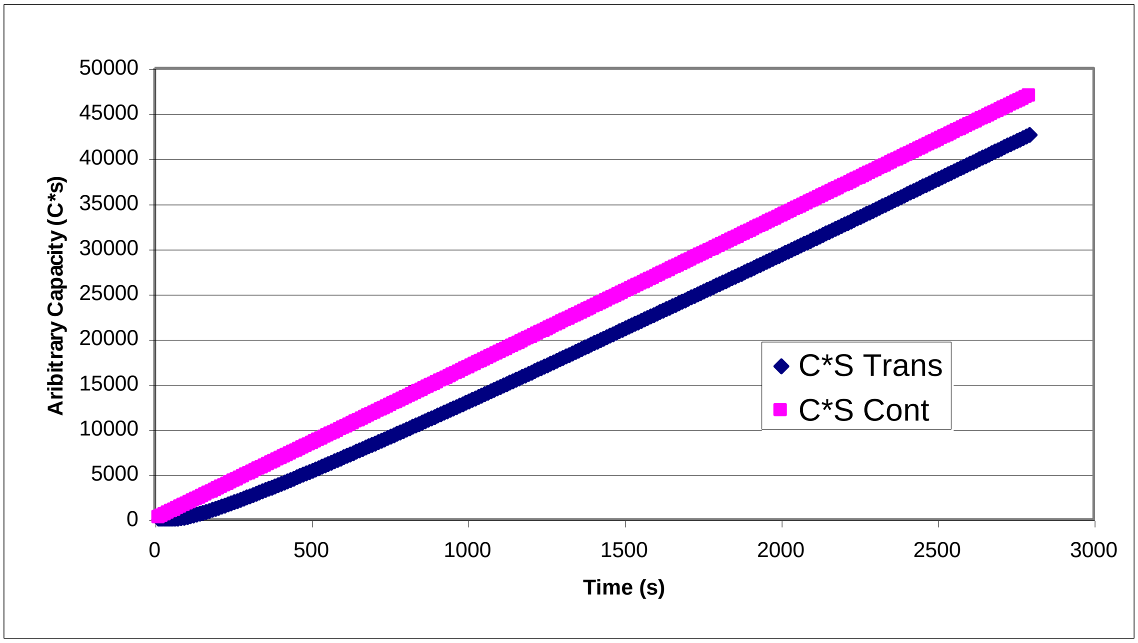
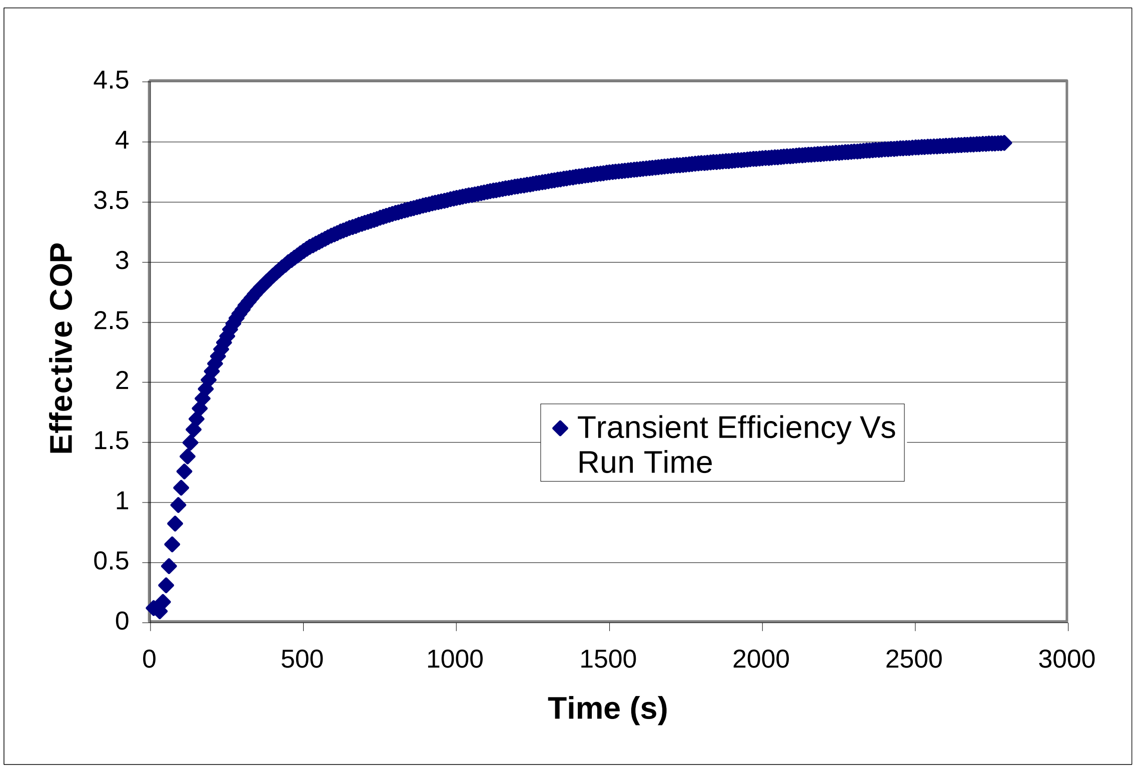


Figure 42 – Cumulative Capacity as Function of Run Time at 65Hz, 1.5°C Ambient



Figures 43 –Mean COP as Function of Run Time at 65Hz, 1.5°C Ambient

The loss of efficiency due to on-off cycling is even worse when defrost cycles are introduced. A defrost cycle disrupts the refrigerant flow much like an off period, but is worse as the indoor and outdoor side pressures swap instead of just equalizing.

5.6 Result Summary

The COP under most test conditions was in excess of 4.0. Capacity, as would be expected, was roughly linear with compressor speed. There was a slight ramping increase in heat output vs compressor speed at high drive frequencies, which can be attributed to increased losses in the compressor contributing to indoor heat output. The transient from poor efficiency to target efficiency was speed up close to a factor of 2 by liquid injection. A slight peaking of efficiency was observed at 76 Hz, but the effect is relatively mild.

Chapter 6

CONCLUSIONS AND FUTURE WORK

6.0 Conclusion

The measured COP performance was as expected, in the 4-4.5 range. Transient performance was also sped up, as expected, with liquid injection. The transient was nearly twice as fast as compared to both conventional systems and also against the same system with liquid injection disabled. While the steady state COP performance showed good improvements over a conventional HP, the gains in efficiency due to a quicker transient and/or through continuous operation can account for much larger gains in a practical system.

The improvement in defrost performance was also as good, though the predicted improvements were based on casual observation of a commercial heat pump's behaviour. While continuously variable compressor speeds can be used to prevent many transients (as compared to on-off control), the need to disrupt the refrigerant flow during defrost remains. The improvement in overall efficiency and capacity can amount to a substantial amount when one considers that the typical defrost cycle lasts several minutes and would normally be triggered every hour.

While this thesis considers the effects of steady state COP and transient COP on overall performance, it does not account for the interaction of very light thermal loads and low compressor speeds. Furthermore this thesis does not consider the effects of temperature regulation constraints and minimum compressor speed. The former occurs on warm days when the building's thermal load is minimal and the HP capacity is maximized. Under these conditions there are cases where simple on-off control is more efficient than a continuous low speed run because of the fixed power draw associated with the fans. The latter effect has to do with efficiency gains possible if larger temperature deviations are permitted in the heated area. When larger deviations are permitted and a structure is well insulated, it is possible to operate in a mode where a single heating cycle per day (lasting several hours, at a higher compressor speed) is more efficient than continuous control.

6.1 Future Work

There are future improvements to the heat pump efficiency that are achievable both through intrinsic improvements to the refrigeration cycle, but also because of the way electricity is sold. In most communities, electricity rates are variable based on time of use (TOU). This creates many opportunities for further cost of use improvements based on both the variable cost of electricity and the variable outdoor temperatures during a 24 hour period. Suggested extensions to this thesis would include the following:

> Feed-forward control based on weather forecast. Heat pump operation can be optimized based on the daily temperature cycle. Both historical data and weather predictions can be used to pick optimal run times for the heat pump assuming slight building temperature variations can be tolerated. During high outdoor ambient conditions, energy can be stored in the buildings thermal mass to allow a reduction in run time during less favourable ambient conditions.
>
> TOU optimization. The above mentioned run time optimization can be further enhanced when the variable price of electricity is considered. During periods of expensive electricity rates, heating can be restricted. A well-designed control algorithm will account for the cost of electricity vs time, the outdoor ambient conditions and the anticipated need for heat. By storing heat in anticipation of a future demand, running the heat pump during the high rate periods can be avoided.

Suggested improvements to the intrinsic cycle design would included the following:

Use of temperature glide to improve evaporator and condenser efficiency. This technique is actually one the author has used in some automotive applications. To improve the usage of the condenser, several refrigerants are blended such that the resulting mix condenses in different portions of the condenser (and evaporates in different portions of the evaporator). This enables more complete use of the heat exchangers as compared to a single component refrigerant system.

Optimized threshold for on-off vs continuous control. The optimal transition between continuous operation and on-off control is also a good study, perhaps for an undergraduate project supervised in part or as a portion of a graduate level design.

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APPENDIX A – INVERTER MANUFACTURERS

Table 7 – Table of Inverter Manufacturers

Brand	Designator
Hitachi	A
Polyspede	B
Hyundai	C
Danfoss	D

APPENDIX B – SUB-COOLER HEAT EXCHANGER DESIGN

Assumptions

Thermal conductivity of Cu[11] 400 W/mK
Thermal conductivity of Solder[11] 50 W/mK
Copper wall thickness 0.25mm
Solder Thickness 2mm
Exchanger width 29.8 mm
Cold side temperature –20c
Hot side inlet temperature 30c
Target Hot side outlet temperature –15c
Target heat transfer 0.65 Ton (this is in excess of what is needed since the accumulator heat exchanger will also aid in subcooling)

The effective thermal conductivity of the copper/solder/copper combo is

2.5/(0.5/400 + 2/50) = 60.6 (assumed 2mm solder thickness and 0.25*2 mm copper thickness)

This figure comes from a simplification where it was assumed that the thermal resistances of the individual materials would be additive and proportional to layer thickness. The different conductivity of the material forming the intermetalic layer was not considered given the large thickness of the materials. This assumption is based on more than a decade of experience in the electronics industry where personally observed copper-solder intermetalic layers seldom exceeded 10uM in thickness.

The heat transfer per unit length is 230 W/mK= 60.6 W/mK / 0.0025m * 0.0095m * deltaT

The target heat transfer is 3517 W/Ton * 0.65 Ton = 2286W

The heat capacity of R290 as inferred from the Mollier diagram (over the 30 to –15C temperature range) = (275-160) kJ/kg / 45c = 2.55 kJ/kgc

At 2 Ton capacity, there will be roughly 2.65 Ton of equivalent refrigerant flow. This assumes that the liquid line is tapped to feed the heat exchanger, after the intercooler. 2.65 * 3517 = 9320W. At 30c condensation, the heat pump extracts (610-275) kJ/kg of refrigerant flow. This gives us a refrigerant flow rate of roughly 28g/S (9320/335000). The density of liquid propane is approximately half a gram/cc yielding a flow rate of 56 cc/S. The cross section of the pipe is 0.78 square cm. 56/0.78= 71.8 cm/s

One side of the heat exchanger is assumed to be constant at –20c due to the phase change and presence of both phases on the cold side. The heat exchanger problem becomes a simple first-order cooling problem where the speed of the flow on the hot side translates the solution from time to position.

$$T(t) = e^{\frac{-t}{Tau}} * A$$

$$Tau = \frac{230(W/(m*K))}{0.000078(M^2)*500(kg/M^3)*2550(J/(kg*K))} = 0.432\,S$$

A = delta T initial

Solving for T(t)=5 gives t = 0.99 seconds or 71.3 cm.
Doubling the length and rounding up to the next integer number of turns around a 8” form gives 6 turns of tubing (150cm).

APPENDIX C – TXV SIZING

The TXV sizing was based on a combination of the Emmerson and Sporlan valve sizing guides[7,8]. A R22 TXV was selected as the base as the pressure curves for R22 and R290 are fairly close. The bulb gas charge was selected to have a broad operating temperature range and no liquid fill to avoid charge migration.

R290 is both less viscous and less dense than R22, but a larger volume of R290 is needed to meet the capacity requirements. There are also differences in inlet subcooling and differential pressure in the modified design. The first step in resizing the valve was to determine the correction factor needed for the different specific gravity and viscosity of R290. Reynolds number for the flow in the valve was estimated using Emmerson's guidelines. The Cv factor of the valve was estimated by converting the nominal tonnage rating to a refrigerant flow spec at the nominal valve differential pressure. This was then used to find the operating Reynolds number. While this method of estimating flows is not the most accurate, it does give a good prediction of the dominant effects in the valve assuming the Reynolds number turns out to be at one of the extremes of the scale.

Cv estimated = 0.55gal/min/(100/1.23)^0.5 = 0.049
Nr = 17250 (0.55 gal/min)/0.1643 / 0.061^0.5 = 235742

Based on the estimated Reynold's number, viscosity effects were ignored in the valve resizing. At 235K, the inertial effects are dominant. Given that the viscosities of R22 and R290 and close while their SG are very different, only the SG correction was used. Again, using Emmerson's guide for valve sizing, a correction factor of 1.55 was arrived at for the change in SG.

Next the Sporlan guide was used to apply correction factors for inlet sub-cooling and reduced differential pressure for R290.

Table 7- From Sporlan valve tables

	R290	R22
Cold side	28psi	42.3psi
Hot side	170psi	140psi
Line drops	6psi	6psi
Liquid distributor	35psi	35psi
TXV delta p	91psi	71psi

From the Sporlan valve tables, this results in a 0.88 correction factor

The R22 valve flows 0.88*1.55 times more when used with R290 at the design system pressures. However, 11% more volume low is needed with R290 to meet capacity.

The resulting correction factor is 1.36.

A 2.5 Ton valve was selected, resulting in a max capacity of 3.96 Ton and minimum 1.36 (Sporlan valves will run down to 40% nominal and at 110% max).

References :

Emmerson design guide d351798
Sporlan TXV Bulletin 10-10 April 2011

APPENDIX D – SNUBBER DESIGN NOTES

The solenoid drivers use an optically isolated triac module. A snubber is necessary to ensure proper commutation of the triac during the turn-off cycle. The purpose of the snubber is to limit both di/dt and dv/dt during the turn-off event such that there is sufficient time for the charge carriers within the device to have been swept away or recombined without re-triggering the device by charge injection through the parasitic capacitances of the device.

The solenoid being driven was measured for inductance and resistance and these values were used determine the phase shift of the current flowing through the solenoid. Because commutation can only occur at zero current, the phase shift is important as it determines the size of the voltage step an non-snubbered triac will experience during attempted commutation. This value was used in simulation to calculate the required snubber values for RC snubber. The design was constrained to have minimum 4x margin for dv/dt and di/dt and 10x margin for transient over-voltage across the triac. The high transient specification was selected to ensure survival in the presence of a 2kv line spike, which can happen during hard commutation of a compressor or fan motor in the same circuit. Finally the design was constrained to use a standard value capacitor (there are more value choices for resistors).

Below is the simulation result and model used for simulation. The simulation was done using the Linear Technologies' version of SPICE, a piecewise linear electrical simulator. The solenoid's inductance and DC resistance are modeled using L1 and R1. C2 is the parasitic capacitance associated with the TRIAC module and the circuit wiring. R2 and C1 are the snubber elements while L2 represents an estimate for the parasitic wiring inductance associated with about an inch of wire. Of the parasitic elements, L2 is the most important to include as it has a profound effect on real world snubber performance. Many snubber circuit fail in the field because the classical method for selecting R2 and C1 omit the inductance from wiring and C1's construction. The reader would be well advised to add 15nH per inch of wiring in the simulation model AND verify that measurements of the real circuit agree with the simulation. Do not take this warning lightly as the failure mode associated with incorrect tuning of the snubber is insidious, often causing a premature field failure several years after commissioning a design.

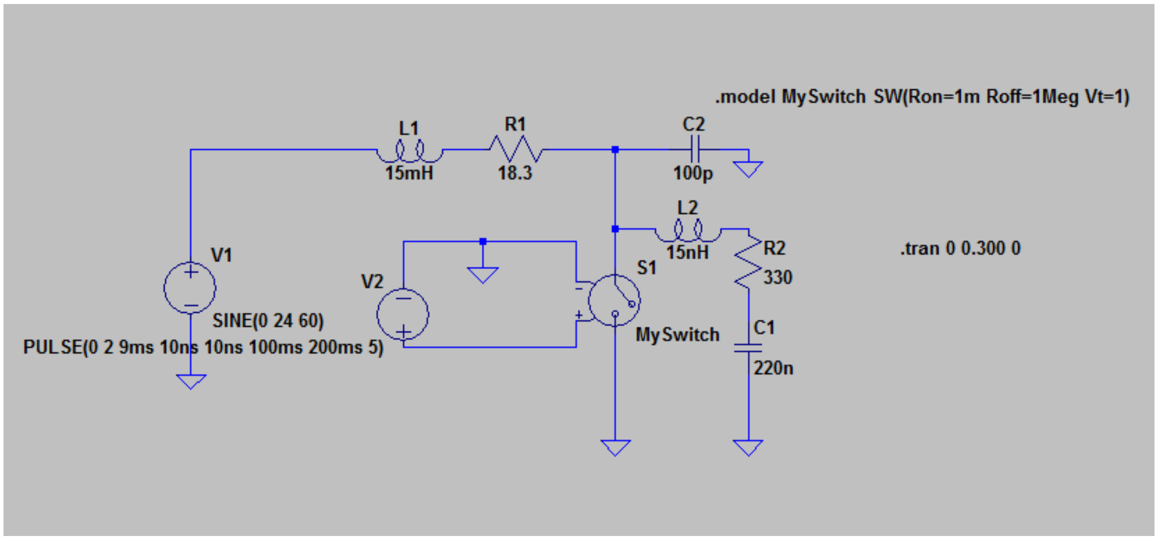

Figure 44 - Snubber optimization simulation circuit

Figure 45 shows the switch voltage and current (during commutation) with the snubber present. The switch voltage is in green while the switch current is in blue.

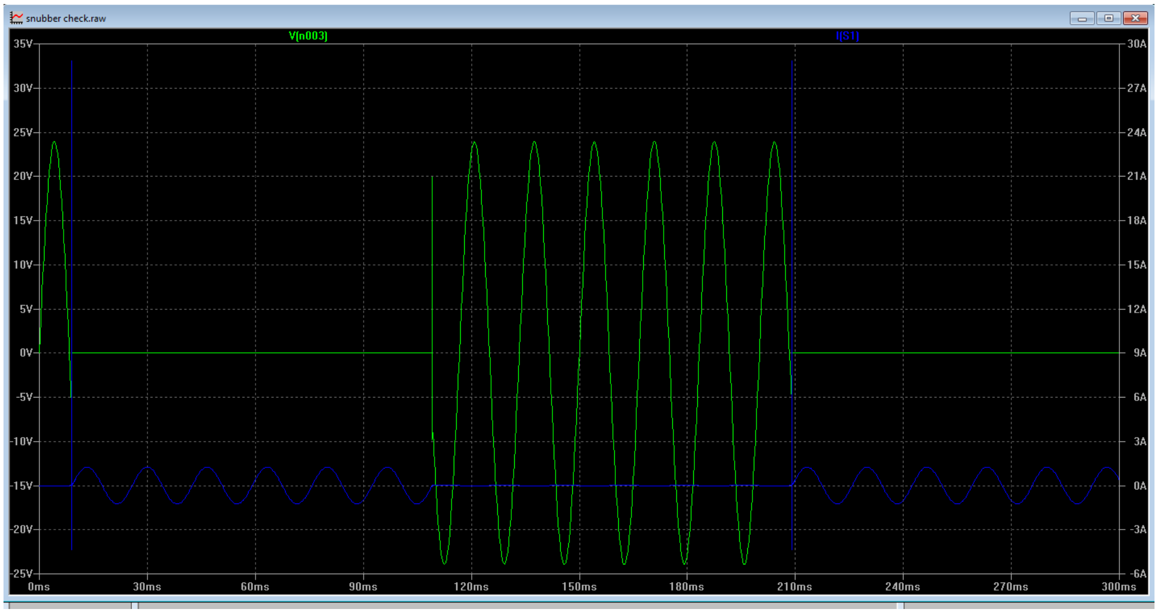

Figure 45 - Snubber optimization simulation result

APPENDIX E - PICTURES OF PROTOTYPE DURING ASSEMBLY AND DURING TEST

Photo 3 - Finished Unit During Charging

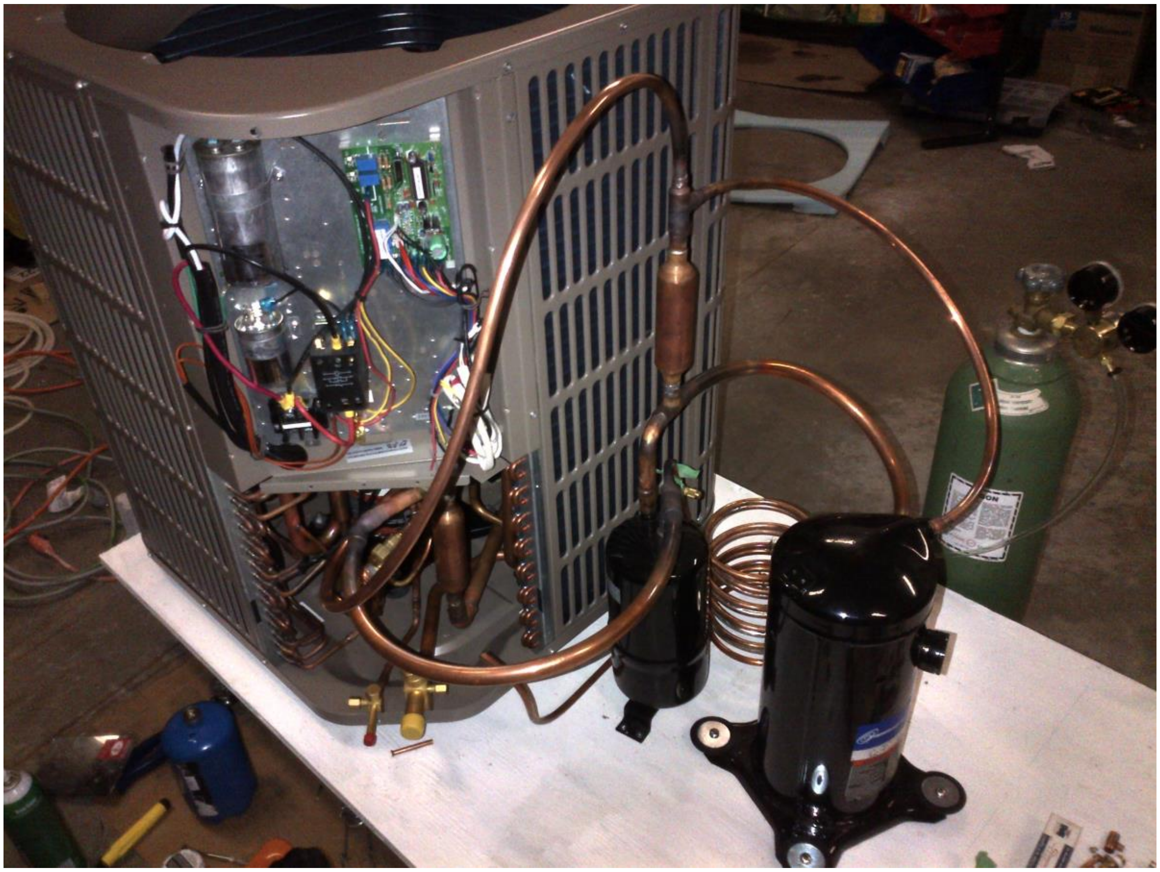

Photo 4 – External Component Assembly

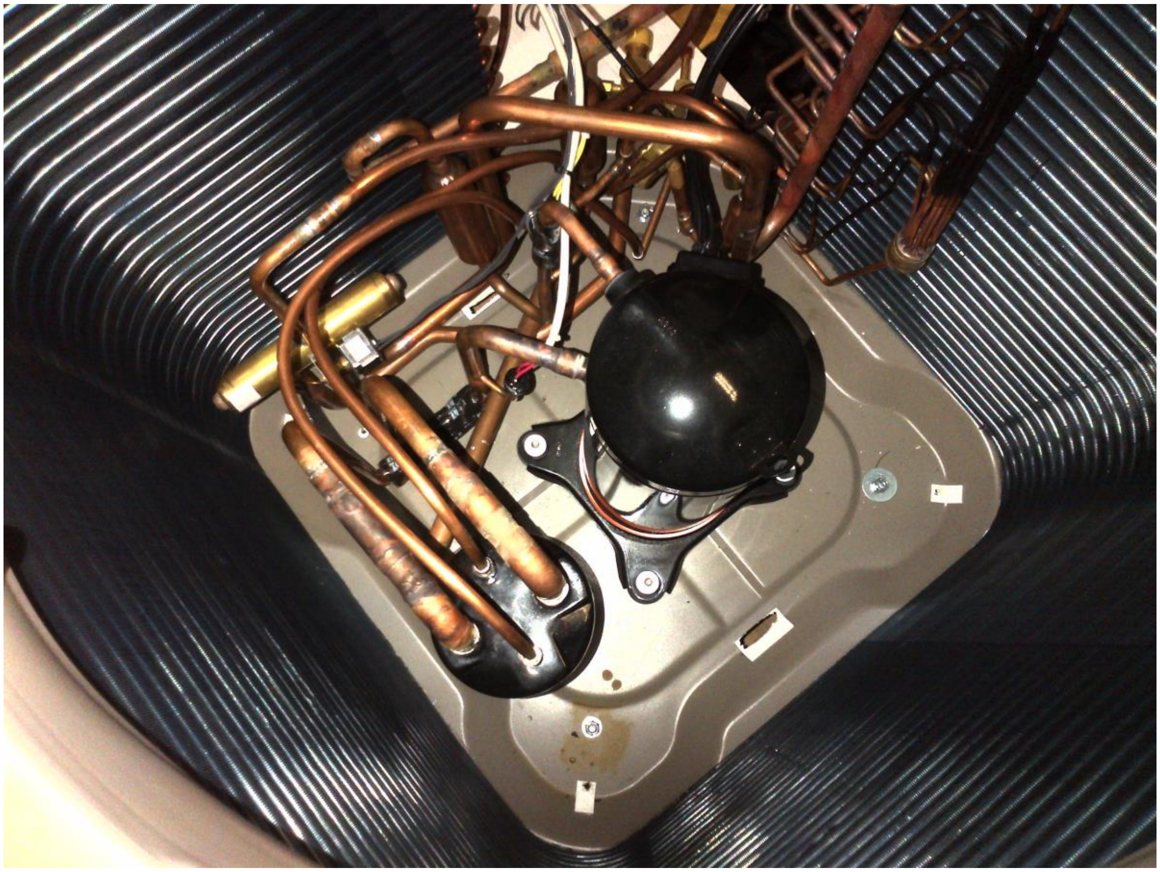

Photo 5 – Internal Modifications

Only the compressor and 4-way valve were retained. The rest of the internal plumbing and components were added in the modification.

APPENDIX F – DESIGN INSPIRATION

This design was inspired by necessity. Two years ago I was faced with the issue of heating a building serviced only with electricity. While there was an existing heat pump in place, its efficiency was wanting. After a little research it was determined that the best efficiency commercial unit was a Mitsubishi HP, which used an economizer style refrigeration cycle. Unfortunately, none of the local Mitsubishi dealers were willing to install a unit, which after some research was attributed to the general level of competence and workmanship within the HVAC industry. It was then that I decided to design and build my own heat pump.

Vapour phase injection compressors were not available in sizes that were appropriate to my application. It was thus necessary to design a refrigeration cycle that would achieve my efficiency targets while using a conventional style compressor. The economizer cycle functions by making the compression process multi-stage. Interstage intercooling is achieved by injecting cold vapour. This has the effect of improving compression efficiency by removing the compression superheat at an intermediate pressure, such that each compression stage becomes more efficient. In addition, a large portion of the refrigerant flow skips one of the compression stages, improving efficiency.

My first design aimed to mimic the two-stage compression method with one compressor by having it pump heat from the outside temperature source to an intermediate heat sink, then from the heat sink to the final indoor temperature. A single compressor would be used, but alternate between extracting outdoor heat and improving the heat previously stored in the intermediate reservoir. The sketch below shows the rough configuration of the element used to do this. I named the element, after its main feature, the stratified sink intercooler. The principal of operation is quite simple. The intercooler removes heat from the liquid refrigerant and transfers it to the water bath. Because the flow of the refrigerant is from the top down, the resulting density stratification in the water bath allows the exit temperature of the coolant to remain very close to the coldest portion of the water bath. This continues to happen even as heat is stored in the bath as the active portion of the heat exchanger simply moves downwards until the pocket of cold water becomes quite small. In contrast to a stirred bath, or even a bottom fed unit, the refrigerant exit temperature remains more or less constant right until most of the water bath has been heated.

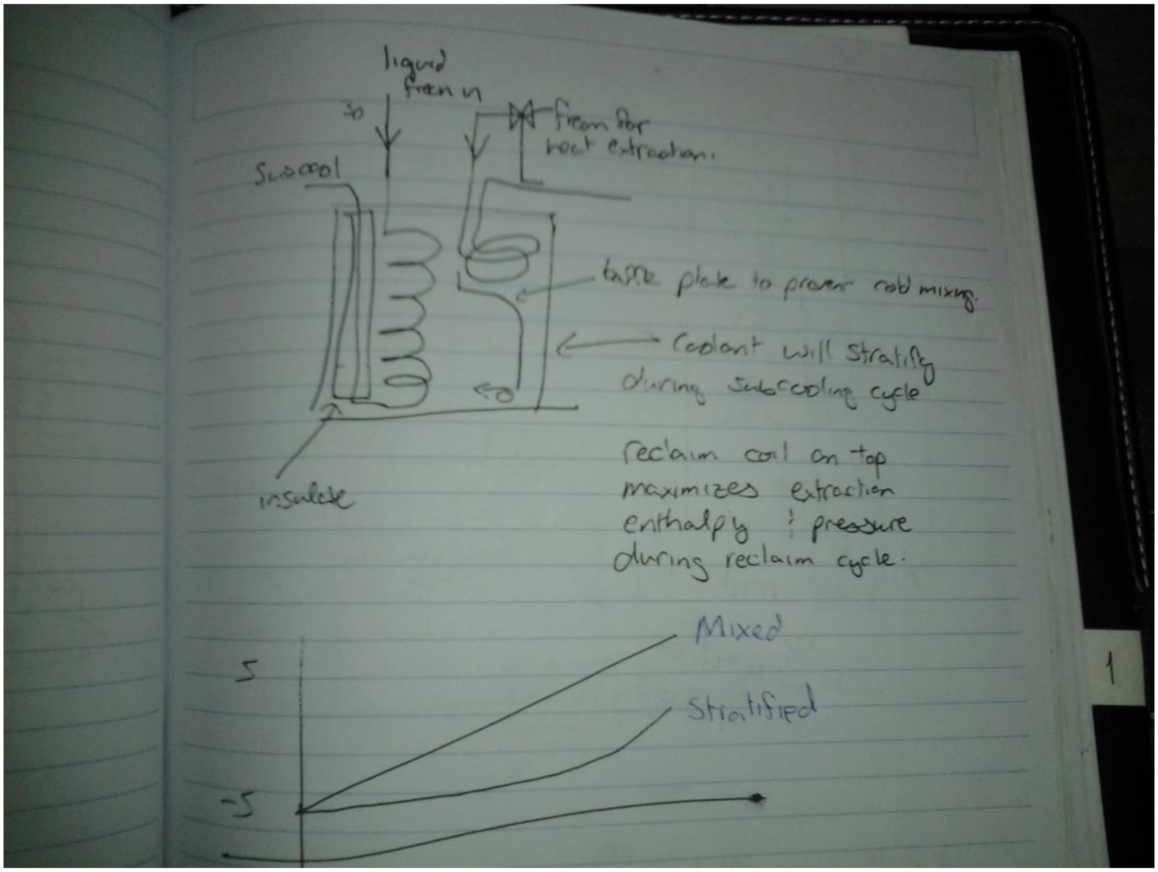

Photo 6 – Stratified Charge Intercooler

The first part of the cycle involves rejecting heat to the indoor coil, followed by subcooling refrigerant in the stratified intercooler. When the bath has been fully heated, the cycle is switched such that heat is now withdrawn from the bath and rejected to the building interior. This cycle provides an efficiency improvement, but also a capacity penalty. The capacity penalty along with the large coolant bath made this method impractical for the heating requirements I was designing for. I had considered allowing the coolant bath to freeze to reduce the size requirement per unit heat capacity, but I did not want to deal with the durability issues associated with a system where freezing is expected.

After a few hand calculations, it occurred to be that the complexity of the economizer style system was unnecessary except when dealing with climate extremes. I suspected that I could achieve most of the efficiency with enough subcooling and careful control of the compressor. This thesis was built to test if careful controls and slight changes to the refrigeration cycle would be sufficient to rival the efficiency of much more complex heat pumps.

APPENDIX G – A PARTIAL LIST OF HVAC SIGNAL DESIGNATORS AND FUNCTION

Signal Designator	Function
C	24Vac common
R	24Vac hot
G	Call For Fan
W	Call For Heat
Y	Call For Compressor
W2	Call For Second Stage Heat
Y2	Call For Second Stage Compressor
E	Emergency Heat
O	Reversing Valve
B	Varies by manufacturer. Alternate Reversing Valve
D	Call For Defrost Heat