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Superheat and Subcooled Levels for Water Source Heat Pumps across Varying Loads and Water Temperatures*

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Introduction

This paper evaluates and analyzes the superheat and subcooled levels for water source heat pumps. As per the non-ideal refrigeration cycle, each cycle has a certain amount of superheat in the suction line and a certain amount of subcooled liquid in the liquid line. By evaluating the level of measured superheat and subcooled levels for a known correctly charged system, we can document and publish the results for internal and external (field technicians') use. By connecting temperature gauges to the suction and liquid lines of a system, the superheat and subcooled levels of the system can be found relatively quickly. Charge can be declared to be correct if the values of the superheat and subcooled levels match those of the lab tested values (which are recorded from known optimally charged systems).

AAON began manufacturing of its water source heat pump product in late 2016. There is a range of capacities that the system can be configured to (as specified by the customer). The overarching goal of this paper is to find trends and consequently develop a set of equations that can be used to predict the superheat and subcooled values of the water source heat pump system given the entering water temperature (for an ideally charged system). This procedure will be repeated across several tonnages of the AAON water source heat pump product line.

Background

The simple refrigeration cycle has 4 processes – evaporation, compression, condensation and expansion. As the refrigerant (which in this case is R-410A) goes through these 4 processes, heat is absorbed from the room in the DX (Direct Expansion or Evaporator) coil and transferred to the atmosphere (or water in the case of a water source heat pump) through the condenser coil. In the liquid line (which is during the expansion or "post-condenser" process), the refrigerant turns into a subcooled liquid (this means the refrigerant liquid has a thermodynamic quality of less than 0). The subcooled value to be studied is the temperature difference between the actual state of the refrigerant and the temperature where the thermodynamic quality of the refrigerant is exactly 0 at that pressure (the saturated liquid temperature).

Similarly, in the suction line (which is during the compression or "post-evaporator" process), the refrigerant turns into a saturated vapor (this means that the refrigerant vapor has a thermodynamic quality of greater than 1). The refrigerant is then released to the discharge line. The superheat value to be studied is the temperature difference between the actual state of the refrigerant and the temperature where the quality is exactly 1 at that pressure (the saturated vapor temperature).

In the water source heat pump, the two coils that run the evaporation and condenser processes are called the evaporator coil and the coaxial coil respectively. However, as a water source heat pump can be used as a cooling or a heating system, the two coils become interchangeable in function. The constant denominator between the heating and cooling runs is that the coaxial coil is always the water coil and the evaporator coil is always the air coil.

The amount of refrigerant that flows through the HVAC system is critical because either undercharge or overcharge can reduce cooling equipment longevity, capacity and efficiency. As a competitive firm in the market, AAON must ensure that its products maximize positive factors (such as longevity and efficiency) and minimize negative factors (such as component damage and noise). The level of charge assigned to a system must be of a level that optimizes the overall engineering merits of the system.

During any system errors, field technicians often study the level of charge as an initial effort to perform a diagnosis. Currently, field technicians often use their own approaches or other cumbersome methods such as evacuating the system of charge, weighing the system and comparing it to previously measured or manufacturing nameplate data. Any disparity will allow the technician to determine that there is a charge level issue within the system (for example, a leak in one of the refrigerant lines). This process can be greatly simplified by adding a manufacturer published superheat and subcooled level of the system – the field technician would simply have to connect temperature gauges and compare the current system levels against the published values. Discrepancies beyond a specified tolerance would allow diagnosis of a charge level issue.

Methodology

In order to find the actual superheat and subcooled values across many different capacities of the water source heat pump, this paper begins by focusing on only a 5 refrigerant ton (60000 BTU/hr) water source heat pump system followed by expanding the study to other systems.

The initial step to studying the 5 refrigerant ton water source heat pump is to determine the optimal level of charge on the 5 refrigerant ton system. The “optimal” level of charge is defined based on the factors of unit capacity and EER (Energy Efficiency Ratio which is the ratio of the cooling effect to the work inputted by a compressor) across varying charge levels. The results for heating and cooling are graphed. The charge level that best accommodates the most number of factors across heating and cooling will be selected.

Following the charge determination test, an optimally charged 5 refrigerant ton water source heat pump will be placed in AAON’s psychrometric chamber and the values of superheat and subcooled levels will be recorded and studied across various loads. A psychrometric chamber is a well-insulated room with controlled ambient conditions of humidity, temperature and airflow – this room allows a thorough study of an HVAC system’s performance under different ambient conditions (varying points on the psychrometric chart). The psychrometric chamber also allows for varying the ambient water temperatures (this is critical as this paper is a study on water source heat pump systems). The chamber is equipped with measuring sensors and devices that capture various data variables including superheat and subcooled levels.

The generated output of the study of the 5 refrigerant ton system is a matrix that records superheat and subcooled levels of the system at varying water temperatures and unit loads. The unit load is a measure of the extent the system is being utilized to – a 5 refrigerant ton system that is providing a cooling effect (the difference in enthalpy of the refrigerant between the entry to the compressor and the exit from the expansion valve) of close to its maximum capacity would be at “High Return conditions.” Likewise, the same system giving a cooling effect of around 2.5 refrigerant tons would be attributed to “Medium Return conditions.” A cooling effect of only 1 refrigerant ton (at 25% capacity) by this system would be categorized as a “Low Return conditions.” These High, Medium and Low conditions will be 3 predefined points on the psychrometric chart (their definitions in terms of their temperatures will be held constant as we test other different capacity systems).

The following is a template of the overall output matrix for the initial test of the 5 refrigerant ton refrigerant system:

Water Temperature (°F)	**High Return Conditions (*SH/SC in °F*)**	**Medium Return Conditions (*SH/SC in °F*)**	**Low Return Conditions (*SH/SC in °F*)**
XX	XX/XX	XX/XX	XX/XX
XX	XX/XX	XX/XX	XX/XX
XX	XX/XX	XX/XX	XX/XX
XX	XX/XX	XX/XX	XX/XX

After this table is generated, an initial study of trends can be implemented (for example, superheat trends by changing the return conditions while keeping the water temperature constant can be studied).

The next step to this test is to generate the same data matrix as above for an identically configured water source heat pump charged for other refrigerant tons. The conditions defining the High, Medium and Low conditions will be kept identical. The new data matrix will allow mathematical isolation of the effects of superheat and subcooled levels of the water source heat pump systems to differences in charge. Trends across each system (unit sizes) will be studied individually as there is not much business value in understanding how charge levels of different unit sizes are related to each other (a field technician working on a 3 refrigerant ton system will not care how its charge level is related to the charge level of a 2 refrigerant ton system).

Literature Review

1. *A Performance Based Method to Determine Refrigerant Charge Level in Unitary Air Conditioning and Heat Pump Systems* (Keith Temple, 2004)[1]

In this paper, the author delves into determining the refrigerant charge level in unitary AC or

[1] http://docs.lib.purdue.edu/cgi/viewcontent.cgi?article=1684&context=iracc

heat pump systems. The author utilizes a performance based method where he implements initial laboratory testing of a system to determine the system's performance characteristics as a function of charge level. This data is then used to determine coefficients in a correlation for a refrigerant charge level (as a percentage of base charge level). The effects that the author studies as factors affecting the charge level are subcooled temperature, superheat temperature, liquid line pressure and suction line pressure. Through lab testing, the author comes up with correlation coefficients to relate the charge level to the studied factors (subcooled temperature, superheat temperature, liquid line pressure and suction line pressure). The author graphs his correlation equation against the actual charge value in the system – the values are within 10% of each other across a variety of different charge levels.

2. *An Evaluation of Superheat-Based Refrigerant Charge Diagnostics for Residential Cooling Systems* (Jeffrey Siegel and Craig Wray, 2001)[2]

In this paper, the authors seek to find the effects of incorrect charge in residential cooling systems and look at methods to evaluate the charge level. The authors first underscore the importance of correctly charging a system and graph the unit's measured Normalized Capacity against a "Deviation from Full Charge" metric. The author graphs different curves (each representing a different ambient temperature) and it is seen that the capacity optimizes at the 0% point in the "Deviation from Full Charge" for each of the different temperature curves. This chart explains that charge level is a crucial factor in order to maximize the capacity from the unit. Next, the paper looks into 3 different superheat test methods as a means of evaluating the charge level of the unit. The paper does extensive performance testing of each of the three superheat methods and finds that only two of the three proposed methods produce good results of charge determination. The paper then delves into the importance of accurately charging a system and studies various ways that an incorrectly charged system results in negative effects.

Summary

The literature review provided special insight to the importance of correctly charged systems.

[2] http://epb.lbl.gov/publications/pdf/lbnl-47476.pdf

Both the papers underscore the importance of correctly charged systems. In addition, while the papers do connect various methods to determining charge, there is no fully accepted and correct method even for simplified systems due to the number of different factors that play a hand in determining this level. As this paper's first step is to determine the optimal charge level and then to study trends at the associated superheat and subcooled levels at this charge level, this paper has important differences and will be looking at a unique relationship that was not delved into in the above literature. Another important difference between this paper and the second paper studied is that they studied residential and not commercial systems. The difference in the magnitude of the unit capacities makes for an important distinction. However, the papers did provide me with an important understanding of the methodologies that the researchers employed and of the results they received.

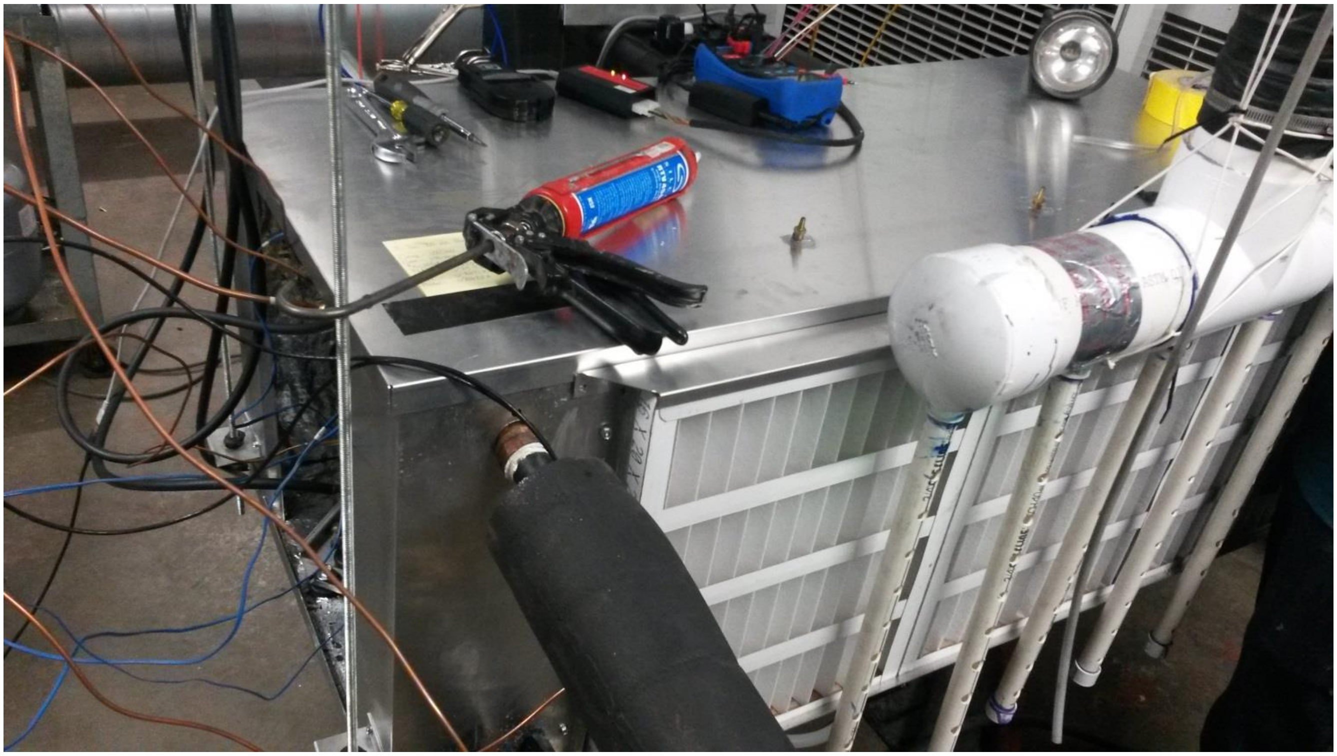
Charge Determination

Prio r to studying superheat and subcooled readings on various tonnages of water source heat pumps, it needed to be ensured that the charge level within the respective water source heat pump system was ideal. In order to determine the charge level, a performance based method was used. Using AAON's psychrometric chamber, different levels of charge were placed in the water source heat pump system and the unit capacity and Energy Efficiency Ratio (EER) were recorded. The test was repeated for both heating and cooling and the charge that was able to best meet or exceed the minimal required capacity and EER for both heating and cooling was selected. The minimal requirements are set by the regulatory body ASHRAE. These regulations are important so that customers in this industry are treated fairly – all manufacturers claiming to sell a 5 refrigerant ton water source heat pump have their units at least meeting or exceeding set capacities and Energy Efficiency Ratios. In addition, for each of the qualifying charge levels, any "red-flag" observations (loud noise, strong vibration, etc) were noted – if a particular charge was found to be responsible for such observations, that charge was "disqualified" regardless of EER and/or COP.

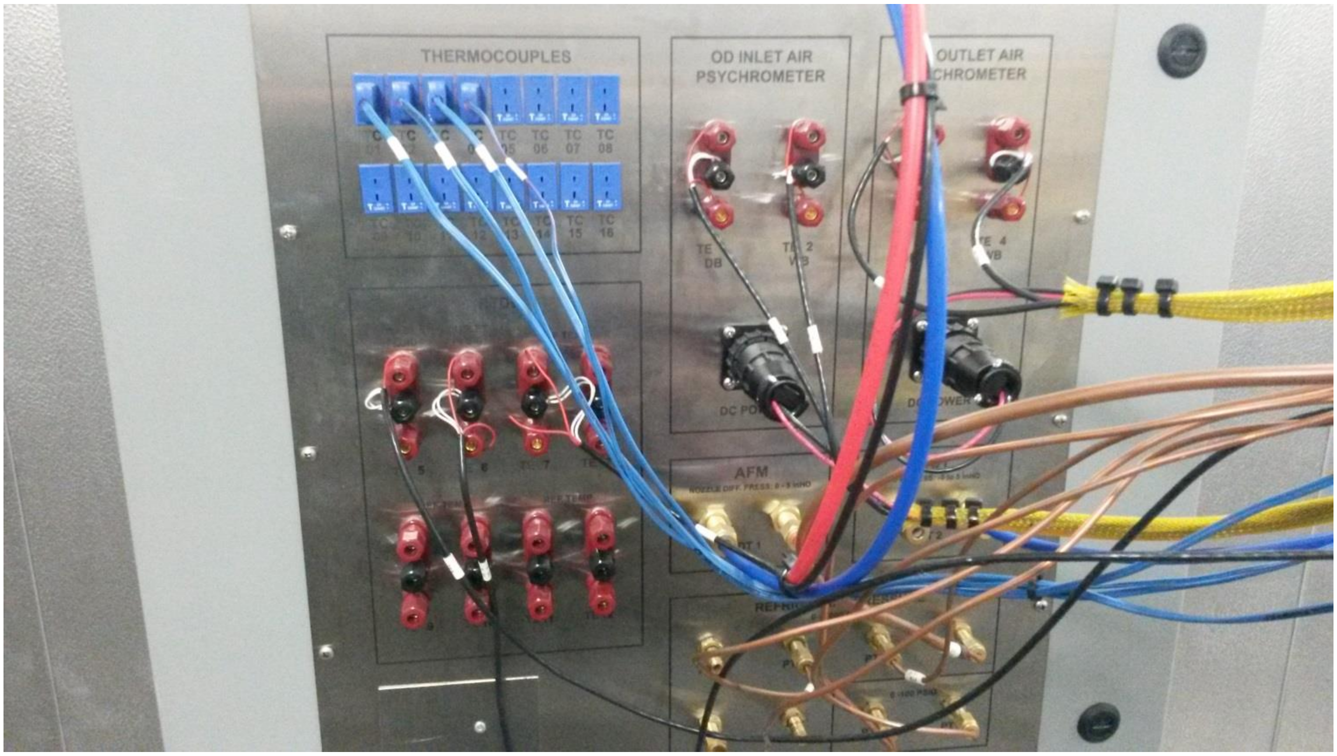
Test Setup

The following photos provide a walkthrough of the test setup:

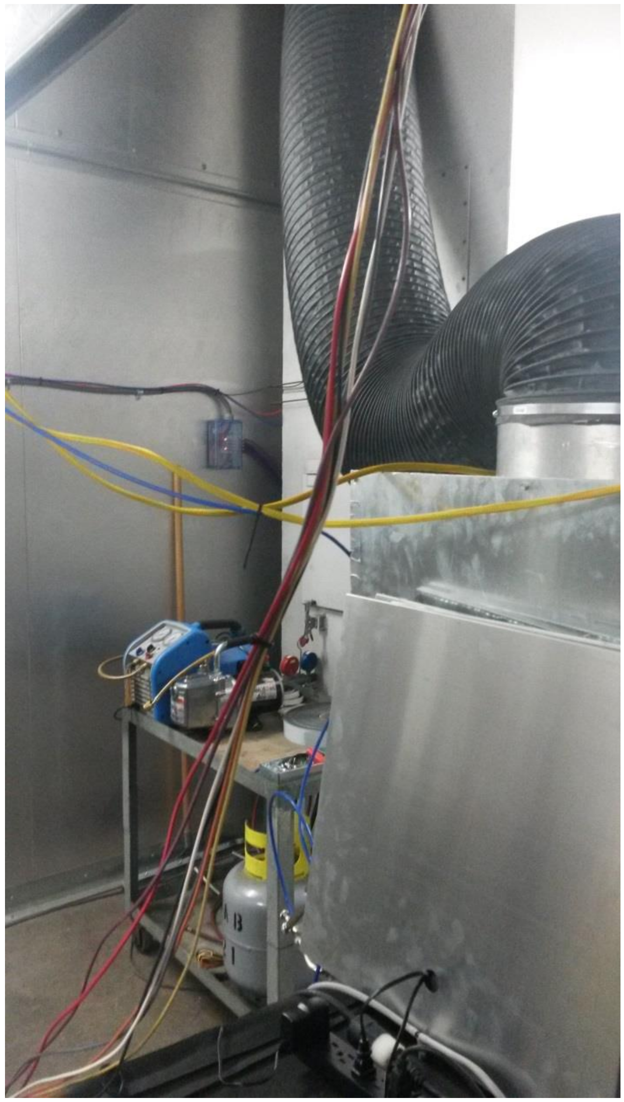
This is the AAON Water Source Heat Pump Psychrometric Chamber. The unit is placed inside the well-insulated chamber so as to prevent the ambient air conditions contaminating the inner controlled conditions of the unit. The psychrometric chamber is connected to a separate control room where the conditions can be managed and the test run performances can be recorded and studied.



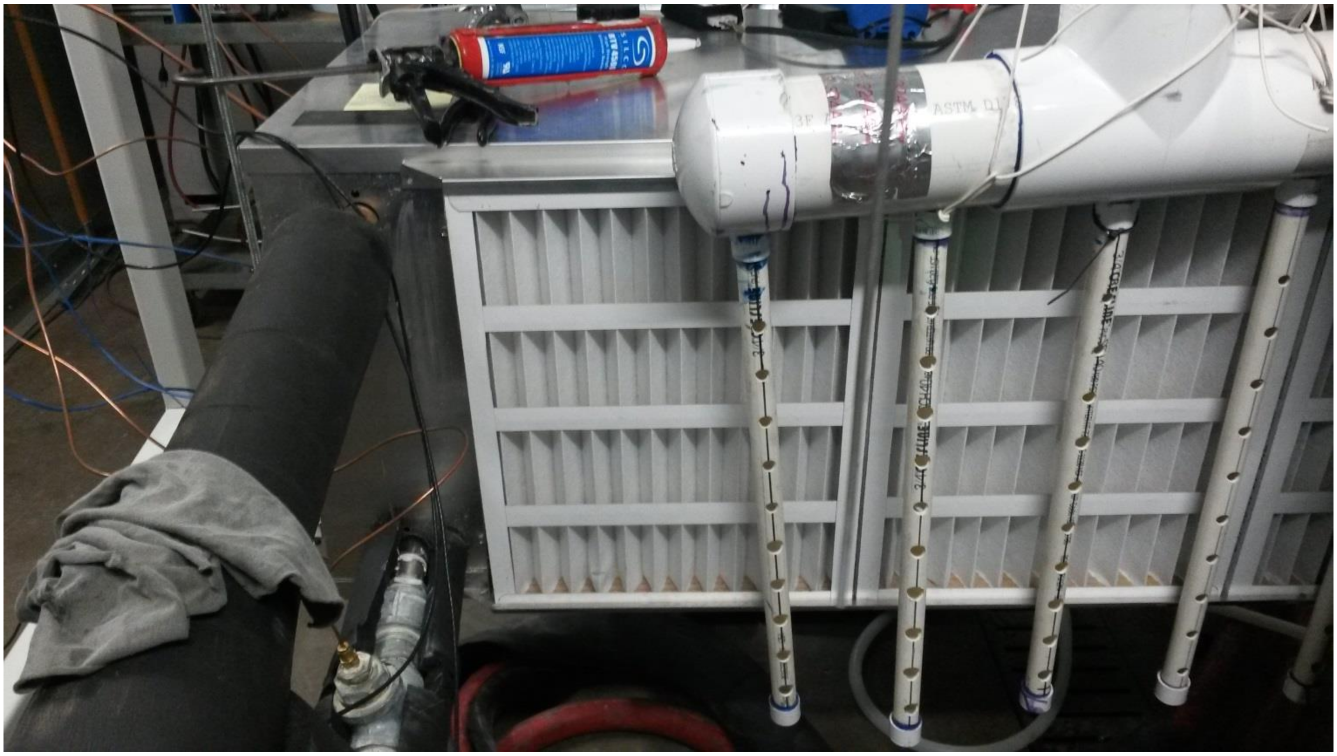
This is the unit placed within the chamber. There are several wires and probes to the left of the control panel – these are used to measure and record several different pieces of data – the data is stored in a computer in the adjoining control room.



This is a clear image of the data being transferred to the computer in the control panel (behind this connection board).



The ducting above feeds return air into the unit at a given preset value – this simulates the behavior of an outdoor space supplying air into the unit.



The white tubes are called a sampling tree - they are placed upstream of the filter to record the air conditions entering the WSHP.

This is a manometer that measures the pressure drop across a component. In this particular case, it is measuring the pressure drop across the evaporator coil during a cooling run.

Test Run Complications

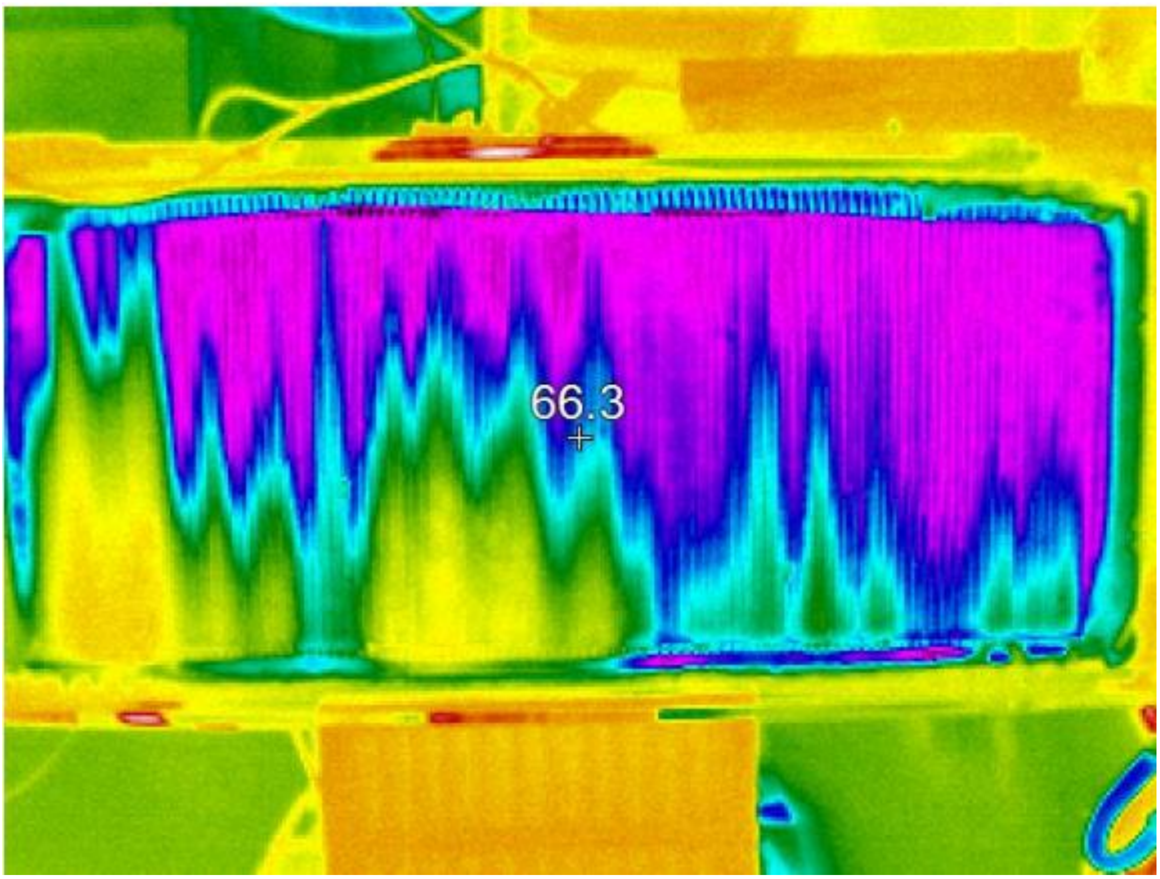
While it may seem that the test run procedure is a fairly straightforward task, there were several significant complications that came up during the psychrometric chamber test runs. It should be noted that while the water source heat pump product has been in the industry for several years, this is a completely new product for AAON. Thus, there were several iterations required for the product design that came up after the unit was in the test chamber.

There were 68 runs conducted before valid test runs to determine charge could be performed on the 5 refrigerant ton water source heat pump. As each run takes a minimum of 45 minutes and a maximum of several hours depending on the change needed, conducting 68 runs was a process that took 3 weeks.

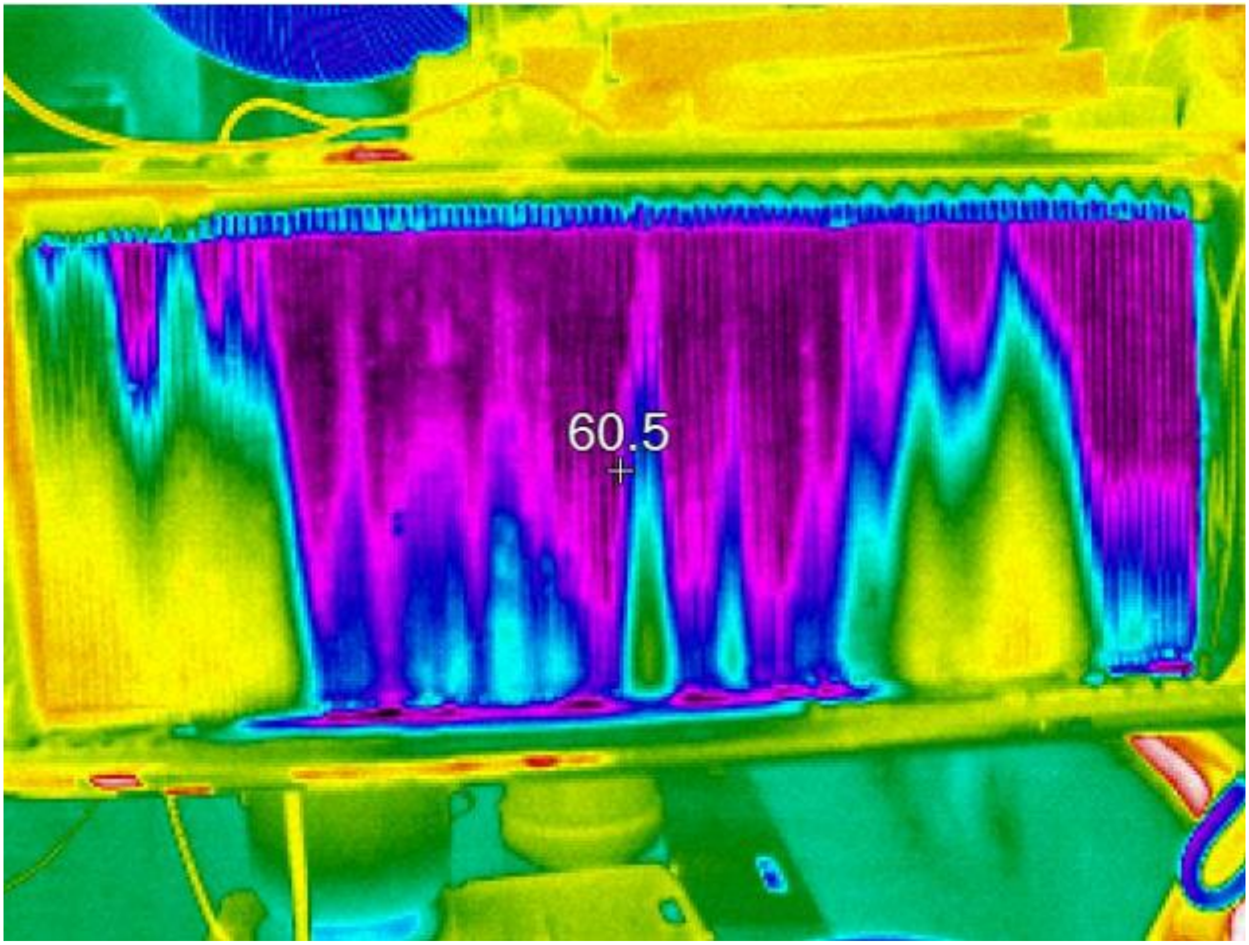
There were several different issues encountered during the test runs, but the following two highlighted issues were the most time-consuming:

Distributor

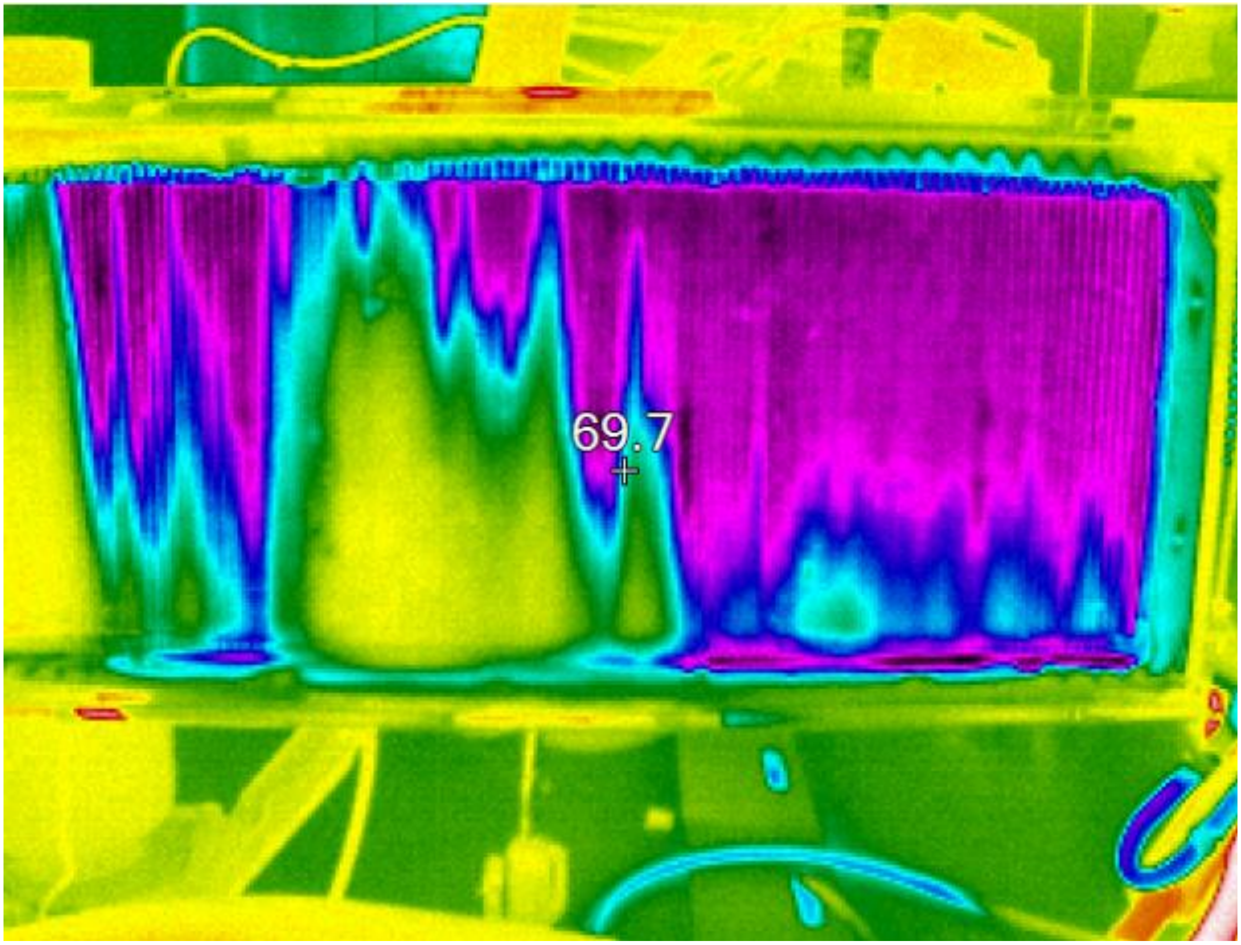
A distributor is used to uniformly flow refrigerant through the evaporator coil in the WSHP. This is important because we are trying to use as much of the coil's surface area as possible (in order to maximize heat transfer). It turned out to be a fairly cumbersome task to find the appropriate distributor for the 5 refrigerant ton WSHP. Thermal imagery was used in order to see the flow of refrigerant through the coil and after several iterations an appropriate distributor was selected. It took several runs of the unit in the chamber to be able to diagnose and solve this recurring issue because changing a distributor and reconditioning the psychrometric chamber can take up to three hours. The images below show thermal imagery of the coil with the various distributors tested. A FLUKE thermal imagery device was used for these images.



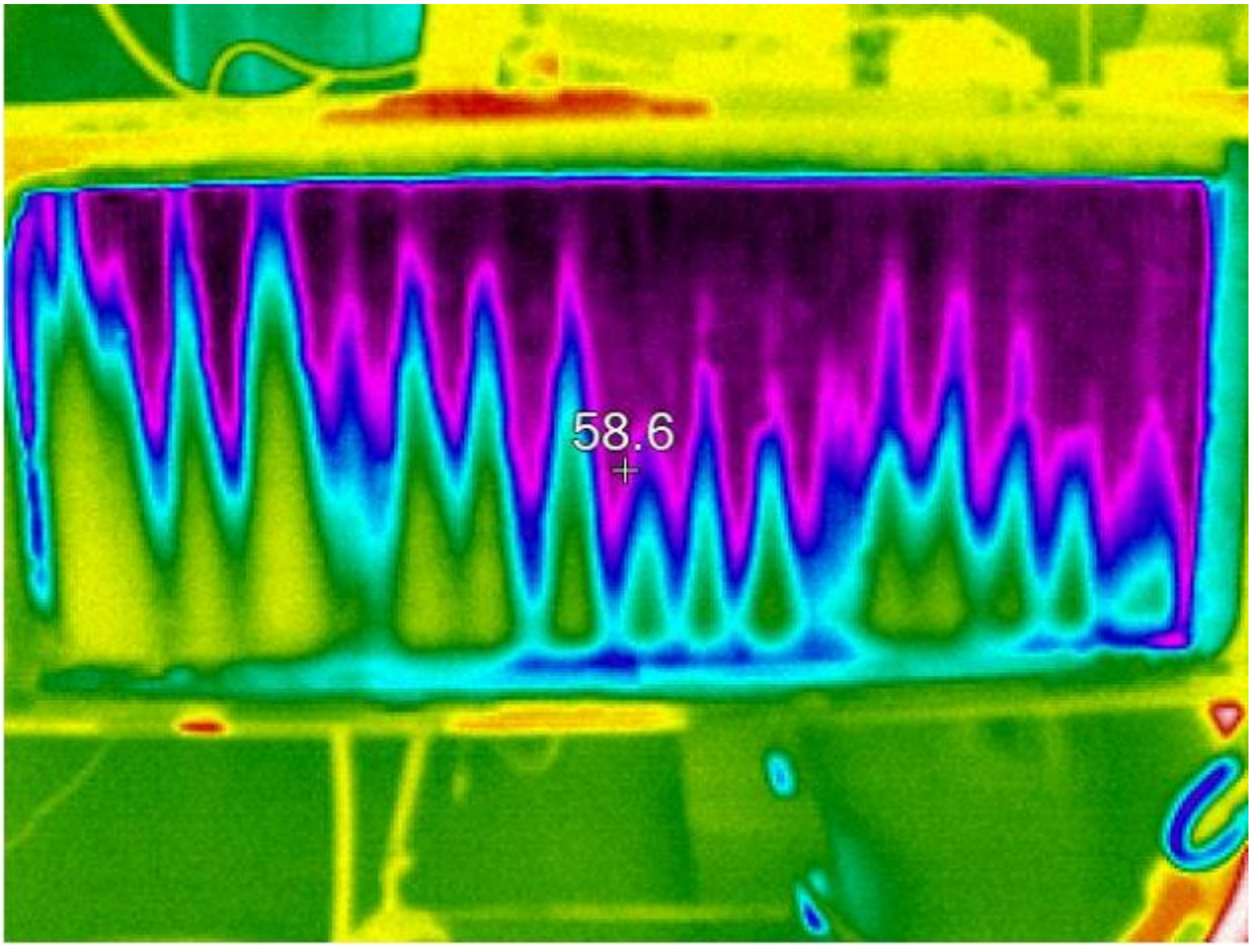
This is the E-1 distributor with 86 ounces in cooling – As we can see, the flow of refrigerant along the coil is not ideal.



This is the E-2 distributor with 86 ounces in cooling – As we can see, the flow of refrigerant along the coil is not ideal.



This is the E-3 distributor with 84 ounces in cooling – As we can see, the flow of refrigerant along the coil is not ideal.



This is the E-4 distributor with 80 ounces in cooling.

As we can see from the graphics, the E-4 distributor provides the “best” flow of refrigerant through the coil. There are no blatant hotspots of refrigerant and the flow is the most evenly distributed out of all the 4 options. The E-4 distributor was selected for moving forward with the system.

Cooling and Heating Balance

In order to single in on a charge of a unit, we have to select a single charge that is optimal on both heating and cooling values (we cannot have a different charge for heating and cooling). In the initial psychrometric runs on the WSHP, it was found that selecting the optimal cooling charge was not meshing well with the optimal heating charge at all. Upon further engineering analysis, it was found that the reason for this was the delta between the volumes of the coaxial and evaporator coil. The evaporator coil had a volume of $110in^3$ while the coaxial coil had a volume of $92in^3$. When the system switches from cooling to heating (and the evaporator effectively becomes the coaxial coil and the coaxial coil becomes the evaporator coil), the charge was seen to be getting flooded due to the relatively small difference in the volumes of the coils. After several runs, the solution to this problem was found by placing a filter drier between the coaxial coil and evaporator coil (which effectively acts as a receiver tank in the system). Upon installing the filter drier, the optimal charges between cooling and heating began to somewhat converge and valid tests could now be done to determine the final charge to place in this unit.

Overall, it took 7 valid test runs to be able to single in on a charge for the 5 refrigerant ton water source heat pump.

The room conditions are controlled by ASHRAE Standards and are 80.6ºF dry bulb for cooling and 66.2ºF for heating, with water temperatures at 86ºF in both cases.

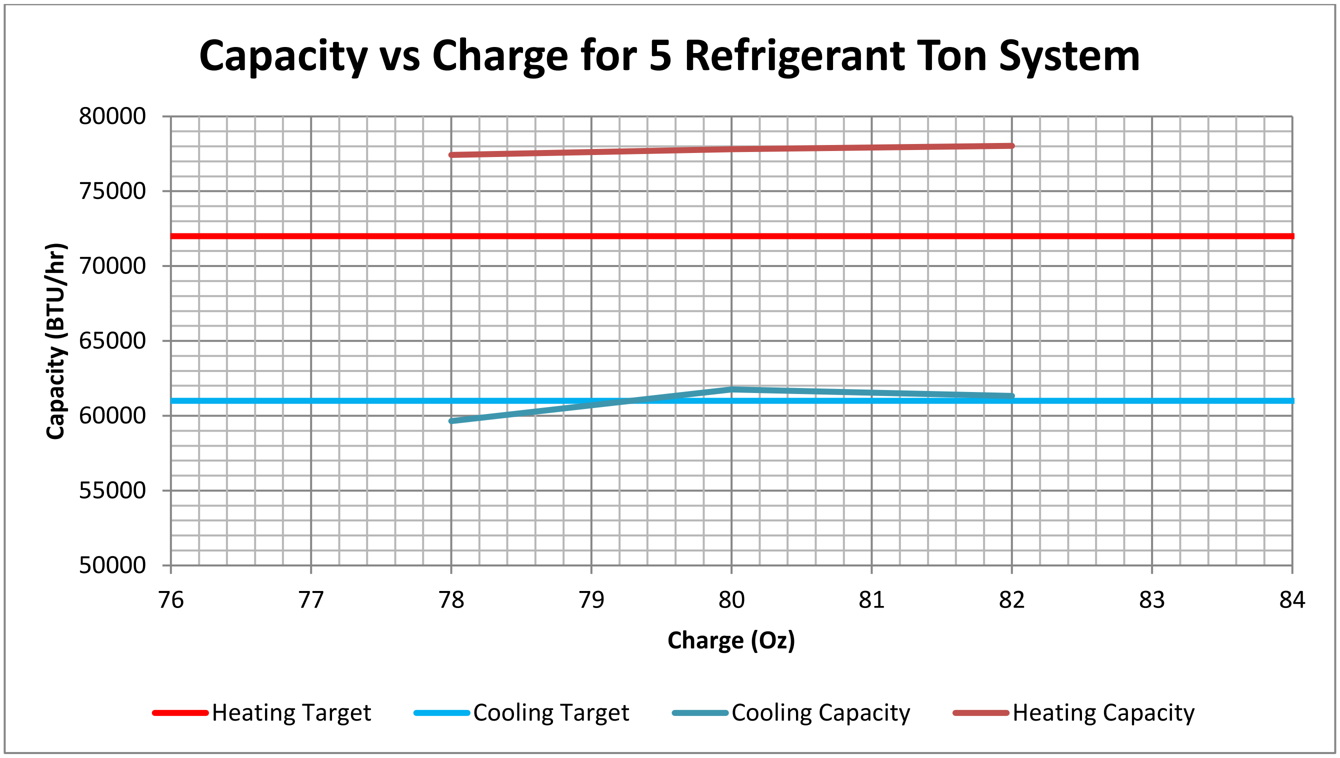
In the appendix, all the 7 Test Runs utilized to single in on a charge have been attached. All variables are directly inputted from the psychrometric chamber (that is, this is raw unprocessed data).

Charges of 78, 80 and 82 ounces of R-410A were used in different test runs. Based on the test runs, it is found that 80 ounces is the optimal charge in this scenario.

Run #	Charge (Oz)	Adjusted Air Side Capacity (cfm)	Adjusted EER (BTU/hrW)	Adjusted COP
69	78	59644.84	14.85	4.35
64	80	61758.59	15.09	4.42
68	82	61319.48	15.08	4.42
70	78	77428.34	17.59	5.15
71	80	77802.82	17.59	5.16
72	82	78034.95	17.55	5.14
73	80	59790.3	14.76	4.32

This table pulls all the relevant data from the complete test run reports for singling in on a charge from the 7 runs. Per industry standard, EER is used for evaluating cooling performance and

COP is used for evaluating heating performance.



The same tests and analysis were repeated for the other units and the following summarizes the optimal charges found across all the units studied –

Unit Capacity (Refrigerant Tons)	**Charge (Oz)**
1.5	36
2	36
3	56
3.5	56
4	54
5	80

The compressor and motor (along with the area of the outside air opening) changes with each of the unit capacities, therefore the relationship between unit capacity and charge is not direct (a higher unit capacity may require less charge as we see above).

Analysis of Superheat and Subcooled Levels

After charge was determined using the methods and procedures outlined above, the next step was to analyze the superheat and subcooled levels for each of the different tonnages of the water source heat pump studied. Ideally, we would be able to conduct runs that would go over High, Medium and Low conditions for all the tonnages being studied. However, due to testing time limitations, the High, Medium and Low conditions were only studied for the 5 refrigerant ton system. For the 1.5, 2, 3, 3.5 and 4 refrigerant ton water source heat pump systems, only runs across varying water temperatures were conducted (all of the runs were conducted at what is defined as "Medium Conditions" below).

The load conditions are summarized as follows –

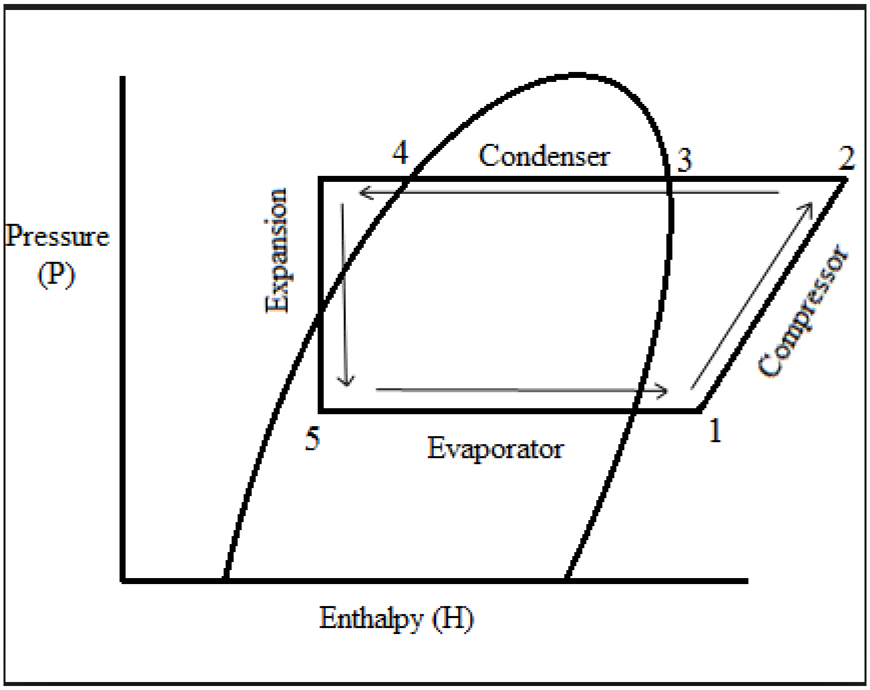
Conditions	**Ambient Temperatures**
High	85˚F DB/ 73˚F WB
Medium	80.6˚F DB/ 66.3˚F WB
Low	75˚F DB/ 60˚F WB

Hypothesis

Prior to generating test data, theoretical engineering knowledge was used to generate a hypothesis. As an expansion valve was used instead of a fixed orifice for the expansion process in this water source heat pump system (which is a metering device used to regulate the flow of refrigerant flowing through the system), it would be reasonable to assume to have the superheat stay fairly constant or slightly dip down as water temperatures are increased. As water temperatures are increased, for an otherwise identical setup (charge, compressor, coil, etc) the superheat value is expected to stay the same because the metering device (which in this case is an expansion valve) will attempt to regulate the refrigerant flowing. However, if the expansion valve does not work perfectly, for example, if it overcompensates for the higher ambient water temperature by restricting the refrigerant flowing (if the change is not directly proportional), the system head pressure will rise, and the superheat value will fall as the ambient water temperature rises. Thus, while there is a conjecture of the movement of the superheat across changing water temperatures, there is no set expectation to this relationship due to the presence of an expansion valve and not a fixed orifice.

It could be argued that liquid subcooled values will be expected to decrease (or in other words, liquid line temperatures can be expected to increase) as water temperatures are increased. From the non-ideal refrigeration cycle graphed below, we can deduce that as the water temperature is increased, the location of the “Evaporator” line will be higher (pressure and temperature are directly related). Consequently, for the same amount of work done by the compressor, one can expect the diagonal line on the pressure-enthalpy graph of the refrigeration cycle (from the suction to the discharge line through the compressor) to be “further reaching” (it reaches a higher enthalpy that is “more east” than previously at the lower water temperature). Next, for the same coaxial coil performance, we can expect the length of the “Coaxial” line on the refrigeration cycle below to be the same. Therefore, for the same compressor and condenser, we expect the subcooled value to be lesser as the water temperature is increased. Thus, the subcooled level of the system can be expected to fall (higher liquid line temperatures) with increasing water temperatures.

In addition, it is predicted that as the superheat temperature rises, the subcooled temperature falls. From the saturated vapor curve of the non-ideal refrigeration cycle below, we can deduce that as the suction superheat rises, for the same work done by the compressor and the condenser (that is the length of the respective lines on the graph stays the same), the subcooled level will not be as high (the lines will not be able to make it as far as they could with a lower discharge superheat level). Thus, an inverse relationship between the superheat and subcooled levels is predicted.



This pressure-enthalpy curve illustrates the non-ideal refrigeration cycle. In the case of the water source heat pump system, the "Condenser" line can be thought to be replaced by a "Coaxial" line. In addition, the evaporator and the coaxial coils can flip flop between each other depending upon whether the system is in heating or cooling mode.

Regression Analysis

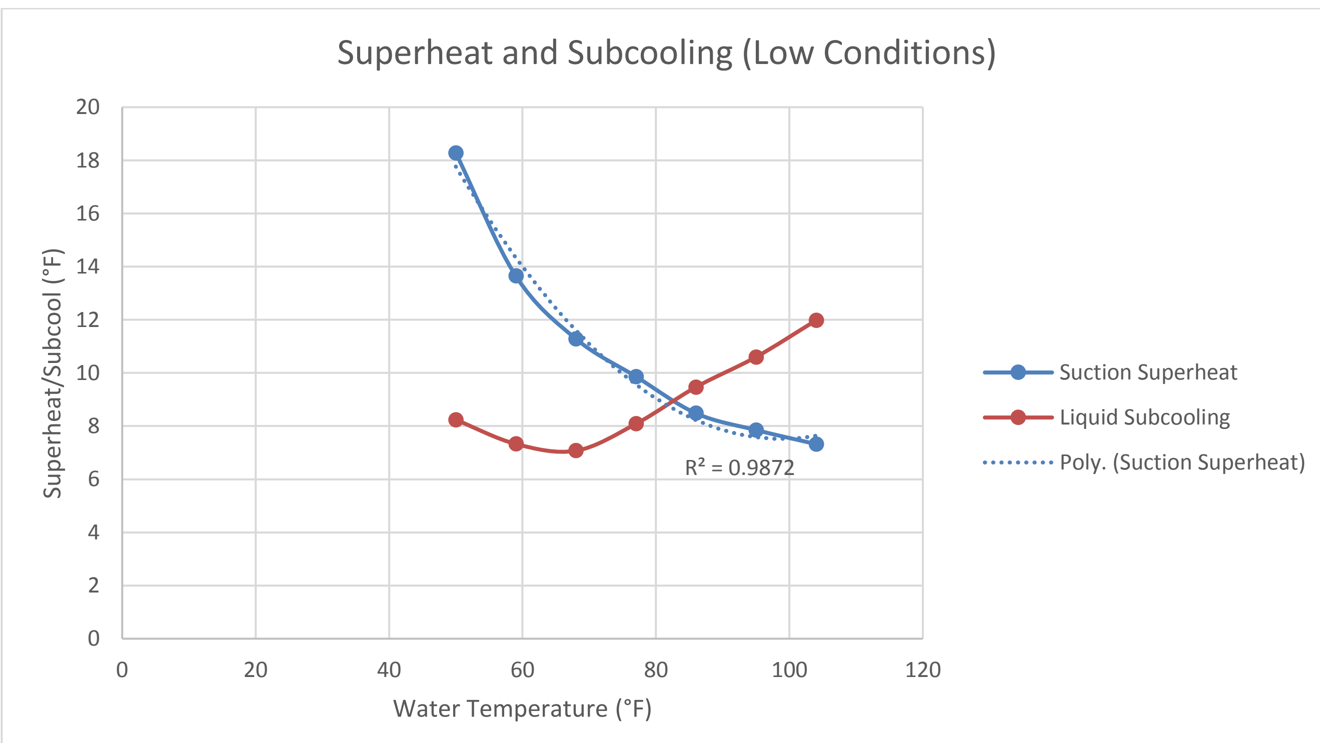
Regression analysis is a statistical process for estimating relationships among variables. Each of the tests below is followed by either a linear or polynomial regression analysis. In this study, the regression analysis was conducted by using the suction superheat and liquid subcooled values as the dependent variables and the water temperatures as the independent variable. The equations were found by generating polynomial and/or linear trendlines on the datasets using Microsoft Excel. For the polynomial equations, all of the R^2 values were over 0.96 and many of them were over 0.98 (an R^2 value of close to 1 signifies a strong and dependable relationship).

5 Refrigerant Ton Water Source Heat Pump

This is the most comprehensive test of all the systems conducted in this paper. Both water temperatures and load conditions were varied across different test runs.

Low Conditions

Water Temperature (°F)	Suction Superheat (°F)	Liquid Subcooled (°F)
50	18.28	8.24
59	13.64	7.33
68	11.28	7.07
77	9.87	8.08
86	8.48	9.46
95	7.85	10.60
104	7.32	11.98



There is a downward trend in suction superheat with increasing water temperatures. There is a downward trend initially in the subcooled levels, and after the 68°F water temperature test, the subcooled rises almost linearly from there.

Equation for the suction superheat level under Low Conditions

$h = 0.0043t^2 – 0.85t + 49.51$ where t = water temperature and h = superheat value

Equation for the liquid subcooled level under Low Conditions

For this situation, it is a parametric two-part equation –

If ambient water temperature is less than 68°F, use

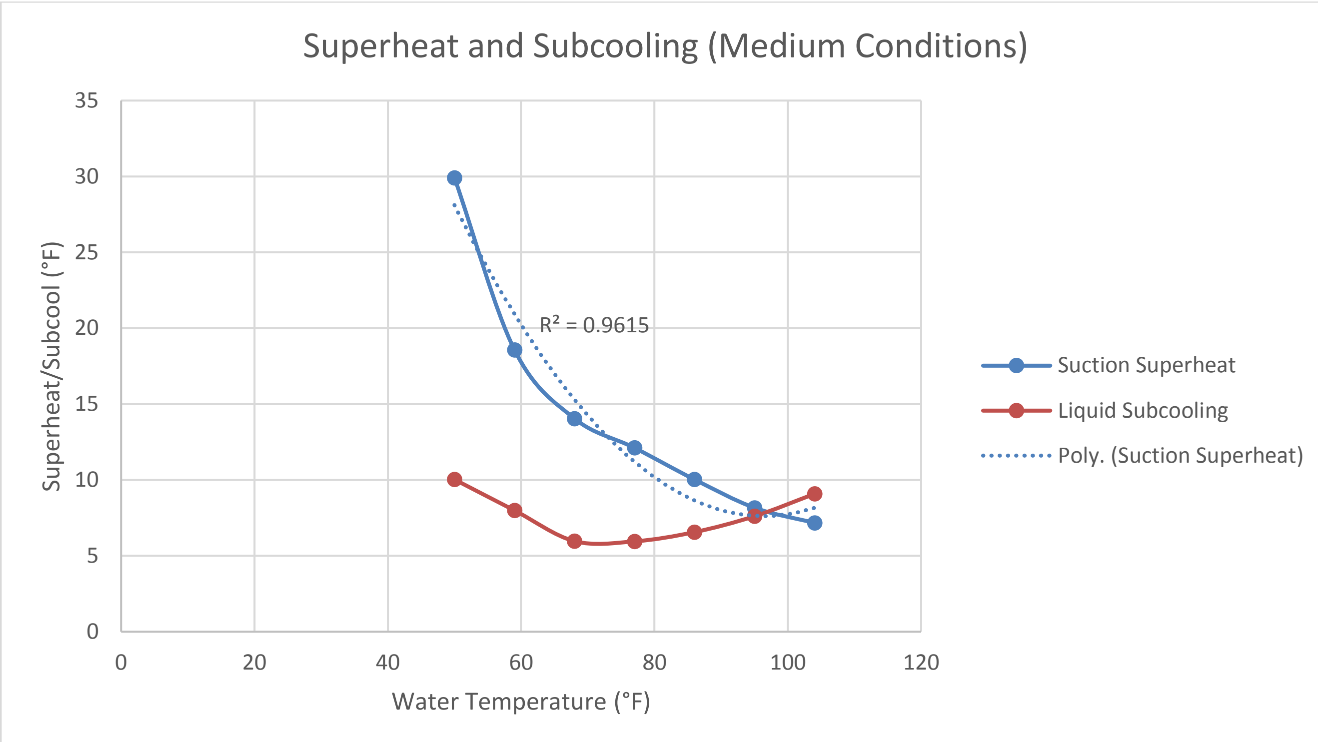
$s = 0.00404t^2 - 0.541t + 25.201$, where t = water temperature and s = subcooled value

If ambient water temperature is greater than 68°F, use

$s = 0.00043t^2 + 0.0637t + 0.743$, where t = water temperature and s = subcooled value

Medium Conditions

Water Temperature (°F)	Suction Superheat Level (°F)	Liquid Subcooled Level (°F)
50	29.91	10.03
59	18.58	7.99
68	14.05	5.97
77	12.12	5.95
86	10.03	6.56
95	8.17	7.61
104	7.17	9.08



There is a downward trend in suction superheat with increasing water temperatures. There is a downward trend initially in the subcooled levels, and after the 77°F water temperature test, the subcooled level rises almost linearly from there.

A regression analysis was conducted on all the data points. The following are the results. These equations can be used by field technicians to determine whether their system's superheat and subcooled values are matching correct values.

Equation for the suction superheat level under Medium Conditions

$h = 0.0095t^2 - 1.83t + 96.04$ where t = water temperature and h = superheat value

Equation for the liquid subcooled level under Medium Conditions

For this situation, it is a parametric two-part equation –

If ambient water temperature is less than 77°F, use

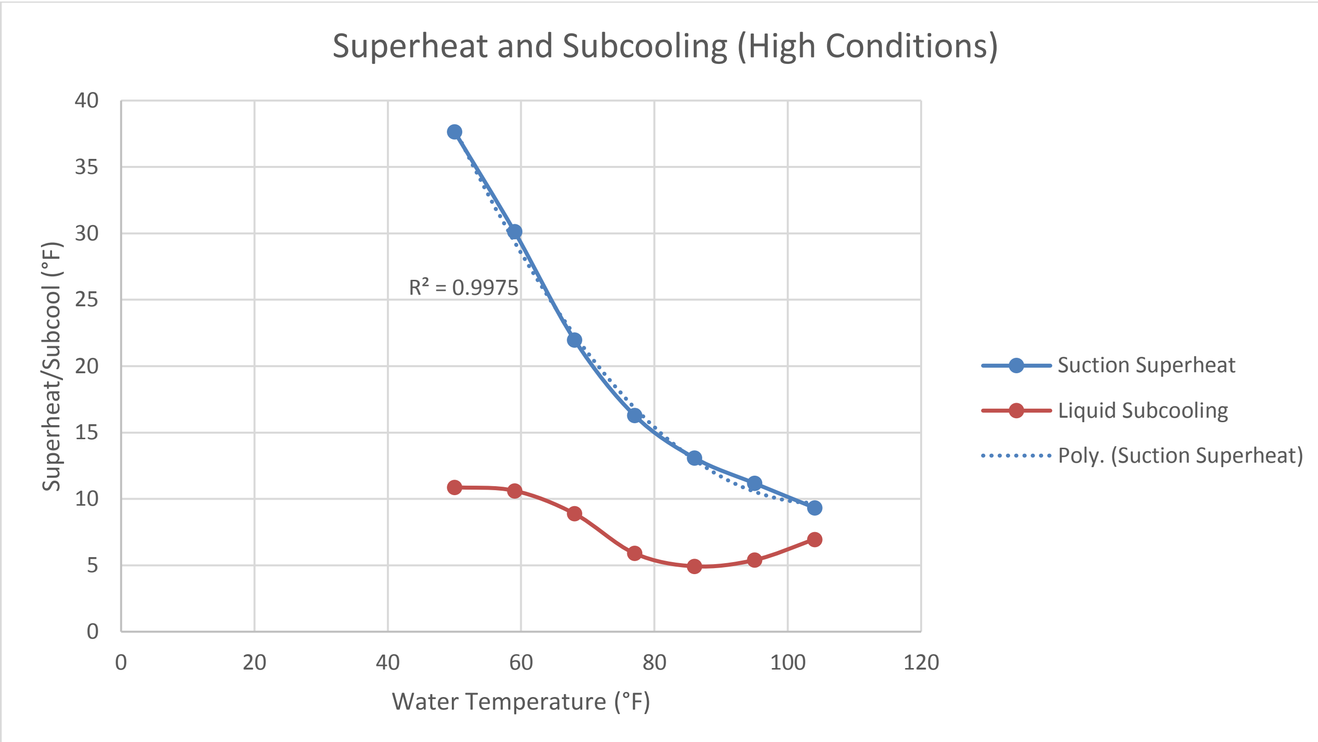
$s = 0.00624t^2 - 0.951t + 42.09$, where t = water temperature and s = subcooled value

If ambient water temperature is greater than 77°F, use

$s = 0.00267t^2 - 0.367t + 18.375$, where t = water temperature and s = subcooled value

High Conditions

Water Temperature (°F)	**Suction Superheat Level (°F)**	**Liquid Subcooled Level (°F)**
50	37.66	10.88
59	30.14	10.62
68	21.98	8.89
77	16.29	5.90
86	13.08	4.92
95	11.18	5.39
104	9.34	6.95



There is a downward trend in suction superheat with increasing water temperatures. There is a downward trend initially in the subcooled levels, and after the 86°F water temperature test, the subcooled level rises smoothly (albeit at a flatter slope than what was seen in the Low and Medium conditions).

Equation for the suction superheat level under High Conditions

$h = 0.0095t^2 - 1.99t + 113.46$, where t = water temperature and h = superheat value

Equation for the liquid subcooled level under High Conditions

For this situation, it is a parametric two-part equation –

If ambient water temperature is less than 86°F, use

$s = -0.0024t^2 + 0.14t + 10.18$, where t = water temperature and s = subcooled value

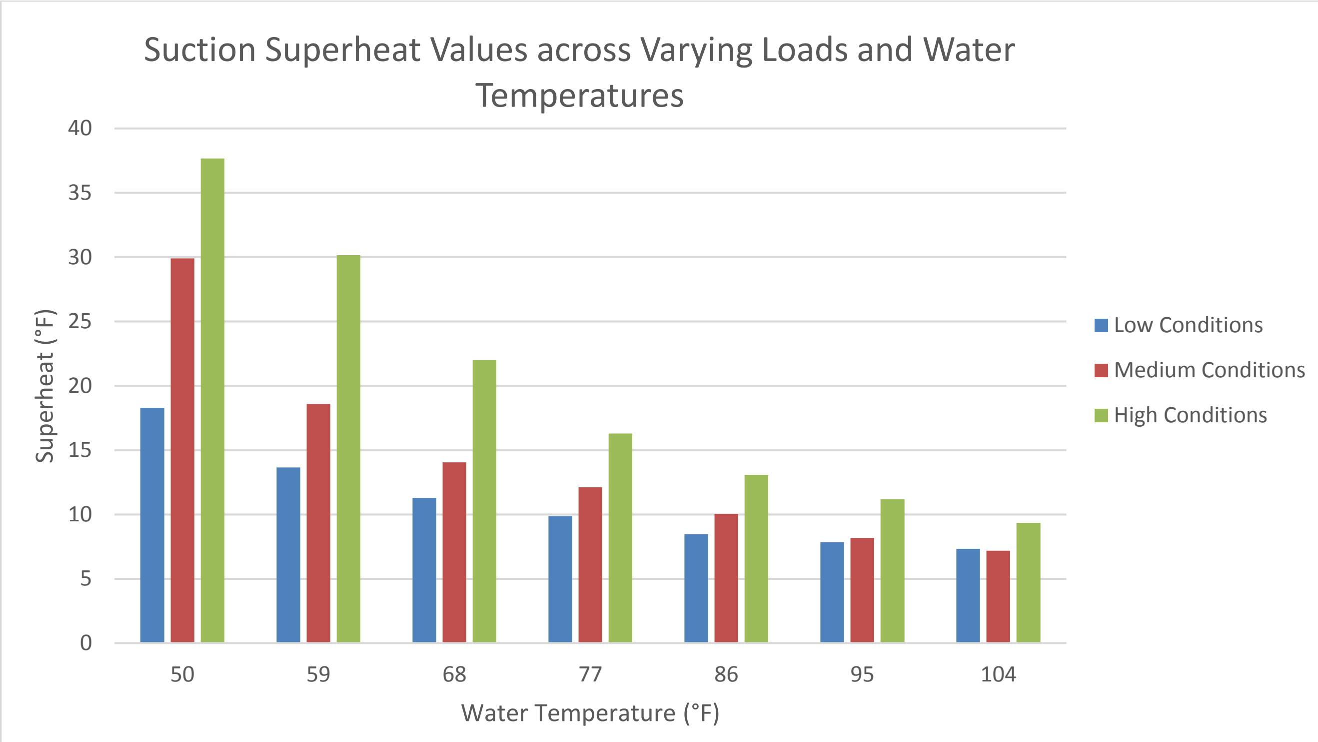
If ambient water temperature is greater than 86°F, use

$s = -0.0024t^2 + 0.14t + 10.18$, where t = water temperature and s = subcooled value

Superheat Trends across Varying Loads

For the 5 refrigerant ton water source heat pump, we have the luxury of studying the variations in data across varying loads in addition to changing water temperatures.

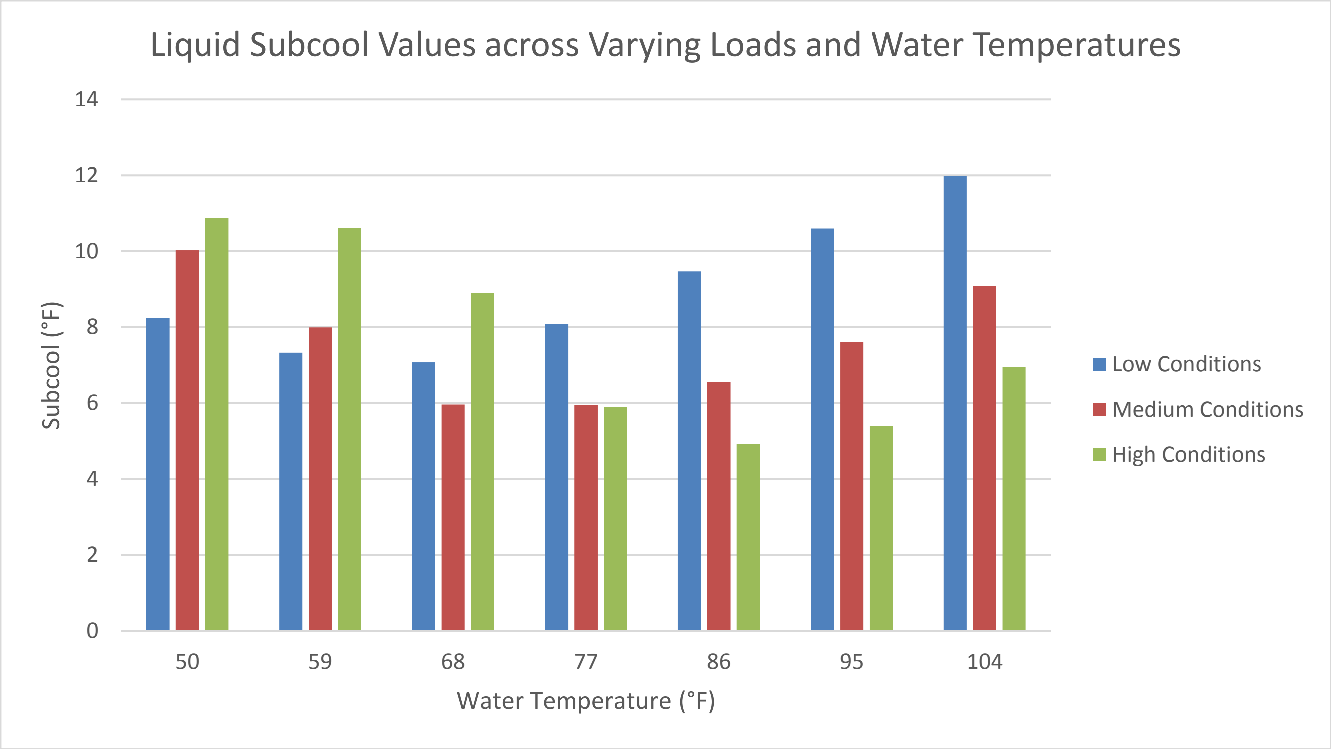
Water Temperature (°F)	**Low Conditions Superheat Level (°F)**	**Medium Conditions Superheat Level (°F)**	**High Conditions Superheat Level (°F)**
50	18.28	29.91	37.66
59	13.65	18.58	30.14
68	11.28	14.05	21.98
77	9.87	12.12	16.29
86	8.48	10.03	13.08
95	7.85	8.17	11.18
104	7.32	7.17	9.34



Looking at the graph's trends, we see a definitive downward movement in superheat values across increasing water temperatures and an increasing trend among increasing load at a given water temperature (overlooking the slight outlier result at the 104°F test).

Subcooled Trends across Varying Loads

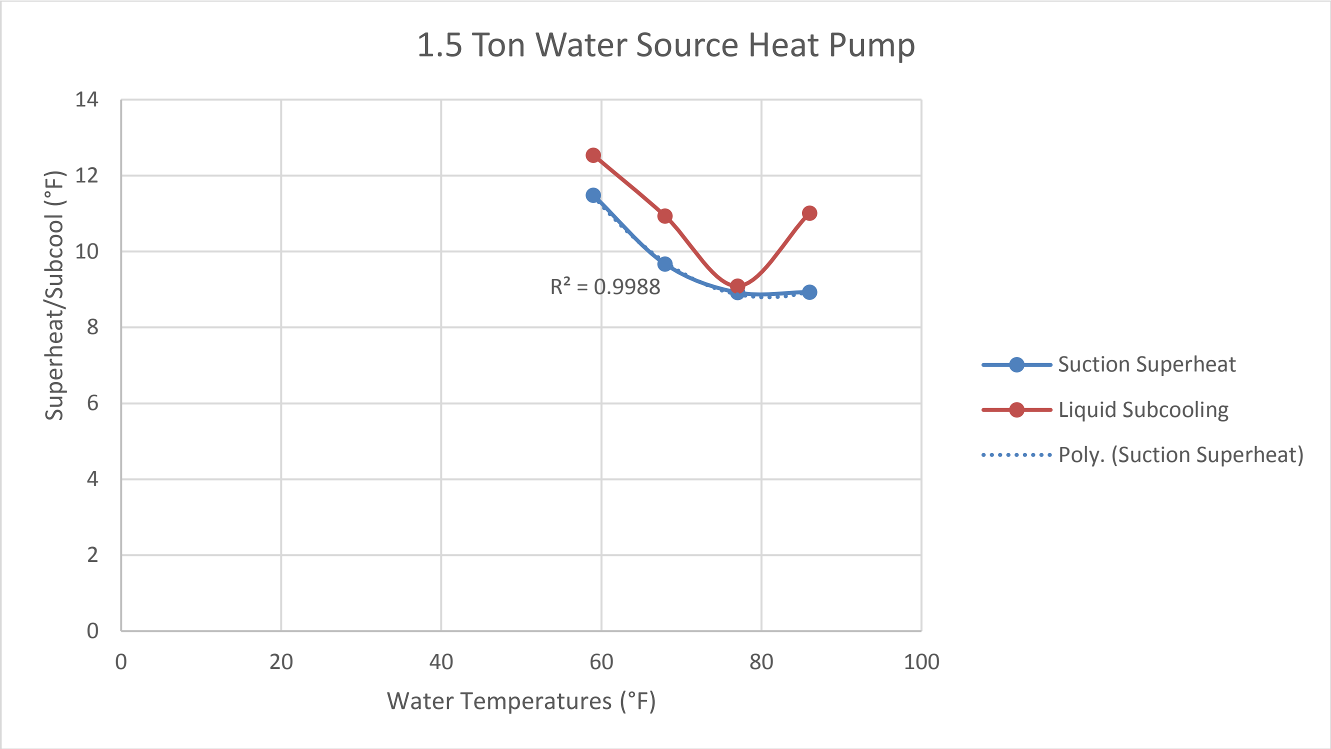
Water Temperature (°F)	**Low Conditions Subcooled Level (°F)**	**Medium Conditions Subcooled Level (°F)**	**High Conditions Subcooled Level (°F)**
50	8.24	10.03	10.88
59	7.33	7.99	10.62
68	7.07	5.97	8.89
77	8.08	5.95	5.90
86	9.47	6.56	4.92
95	10.60	7.61	5.39
104	11.98	9.08	6.95



Looking at the graph's trends, we see a downward trend initially in the subcooled levels, and after the 77°F water temperature test, it rises almost linearly from there (for each of the loads). In addition, there is another trend of a higher subcooled level for increasing loads (at a given temperature) up to the 77°F water temperature test (overlooking the slight outlier result at the 68°F test) and then a lower subcooled level at increasing loads for tests conducted at a water temperature beyond 77°F.

1.5 Refrigerant Ton Water Source Heat Pump

Entering Water Temperature (°F)	Suction Superheat Level (°F)	Liquid Subcooled Level (°F)
58.99	11.49	12.54
67.93	9.67	10.94
76.99	8.92	9.09
86.00	8.93	11.01



The 1.5 refrigerant ton test was conducted across only 4 water temperatures, so the trends present themselves with a lower level of confidence when compared to the 5 refrigerant ton test. Similar to the 5 refrigerant ton test, the suction superheat values are seen to trend downwards with increasing water temperatures. Also similar to the 5 refrigerant ton test, there is a downward trend initially in the subcooled levels, and after the 77°F water temperature test, the subcooled rises almost linearly from there.

<u>Equation for the suction superheat level</u>

$h = 0.00565t^2 - 0.913t + 45.66$, where t = water temperature and h = superheat value

<u>Equation for the liquid subcooled level</u>

For this situation, it is a parametric two-part equation –

If ambient water temperature is less than 77°F, use

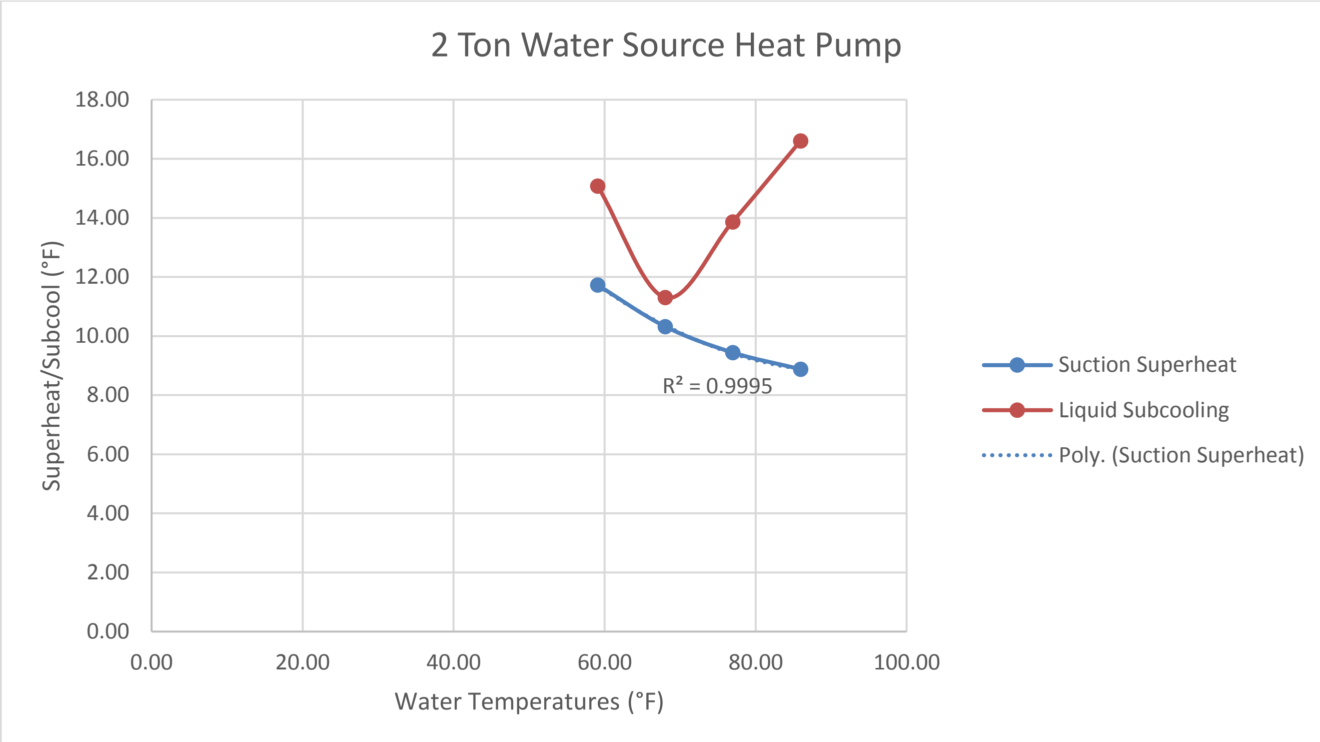
$s = -0.00139t^2 - 0.0024t + 17.53$, where t = water temperature and s = subcooled value

If ambient water temperature is greater than 77°F, use

$s = 0.21t - 7.32$, where t = water temperature and s = subcooled value [For this portion of the two-part parametric equation, a linear regression model was used as there are only two data points]

2 Refrigerant Ton Water Source Heat Pump

Entering Water Temperature (°F)	Suction Superheat Level (°F)	Liquid Subcooled Level (°F)
59.07	11.73	15.08
68.02	10.32	11.30
76.95	9.44	13.87
85.96	8.88	16.61



The 2 refrigerant ton test was conducted across only 4 water temperatures, so the trends present themselves with a lower level of confidence when compared to the 5 refrigerant ton test. Similar to the 5 refrigerant ton test, the suction superheat values are seen to trend downwards with increasing water temperatures. Also similar to the 5 refrigerant ton test, there is a downward trend initially in the subcooled levels, and after the 68°F water temperature test, the subcooled rises almost linearly from there.

Equation for the suction superheat level

$h = 0.00266t^2 - 0.491t + 31.45$, where t = water temperature and h = superheat value

Equation for the liquid subcooled level

For this situation, it is a parametric two-part equation –

If ambient water temperature is less than 68°F, use

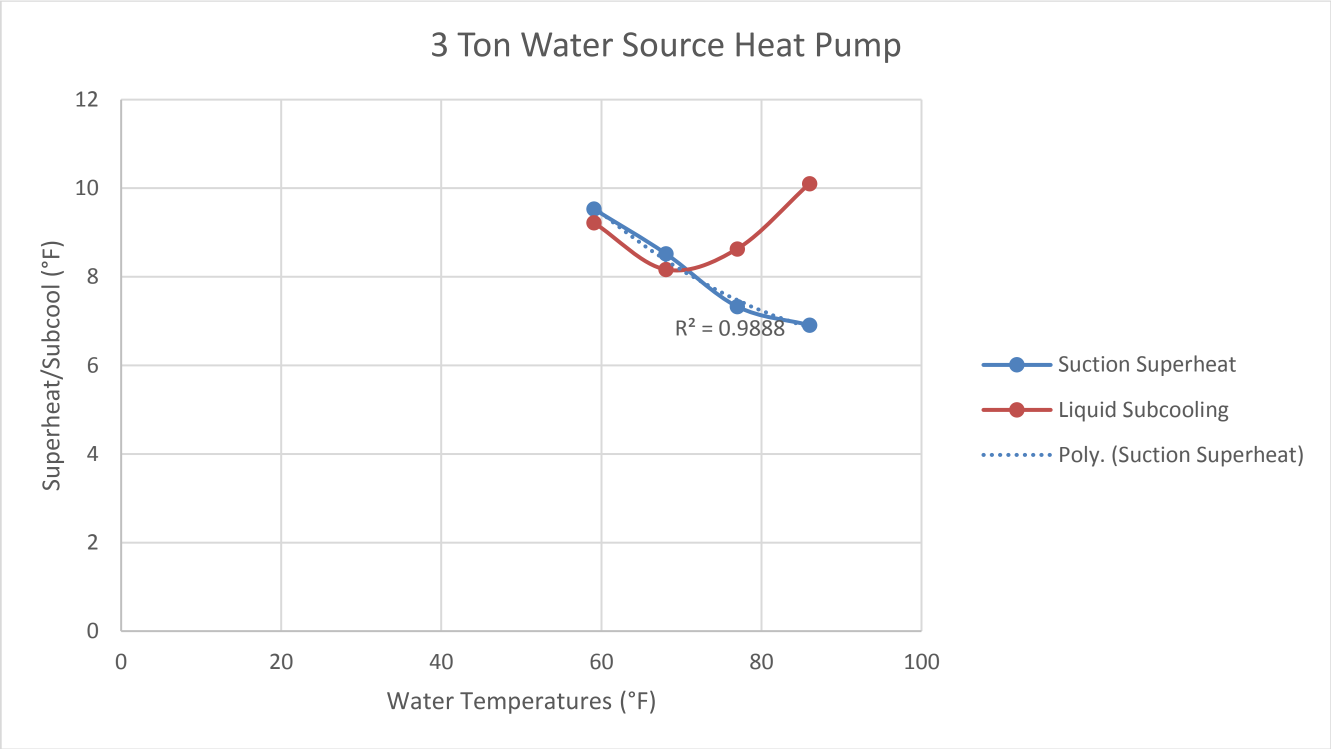
$s = -0.422t + 40.03$, where t = water temperature and s = subcooled value [For this portion of the two-part parametric equation, a linear regression model was used as there are only two data points available]

If ambient water temperature is greater than 68°F, use

$s = 0.00203t^2 - 0.39t + 27.64$, where t = water temperature and s = subcooled value

3 Refrigerant Ton Water Source Heat Pump

Entering Water Temperature (°F)	**Suction Superheat Level (°F)**	**Liquid Subcooled Level (°F)**
59.07	9.53	9.22
68.04	8.52	8.17
76.99	7.33	8.63
86.01	6.91	10.11



The 3 refrigerant ton test was conducted only across 4 water temperatures, so the trends present themselves with a lower level of confidence when compared to the 5 refrigerant ton test. Similar to the 5 refrigerant ton test, the suction superheat values are seen to trend downwards with increasing water temperatures. Also similar to the 5 refrigerant ton test, there is a downward trend initially in the subcooled levels, and after the 68°F water temperature test, the subcooled rises almost linearly from there.

<u>*Equation for the suction superheat level*</u>

$h = 0.00183t^2 - 0.367t + 24.85$, where t = water temperature and h = superheat value

<u>*Equation for the liquid subcooled level*</u>

For this situation, it is a parametric equation –

If ambient water temperature is less than 68°F, use

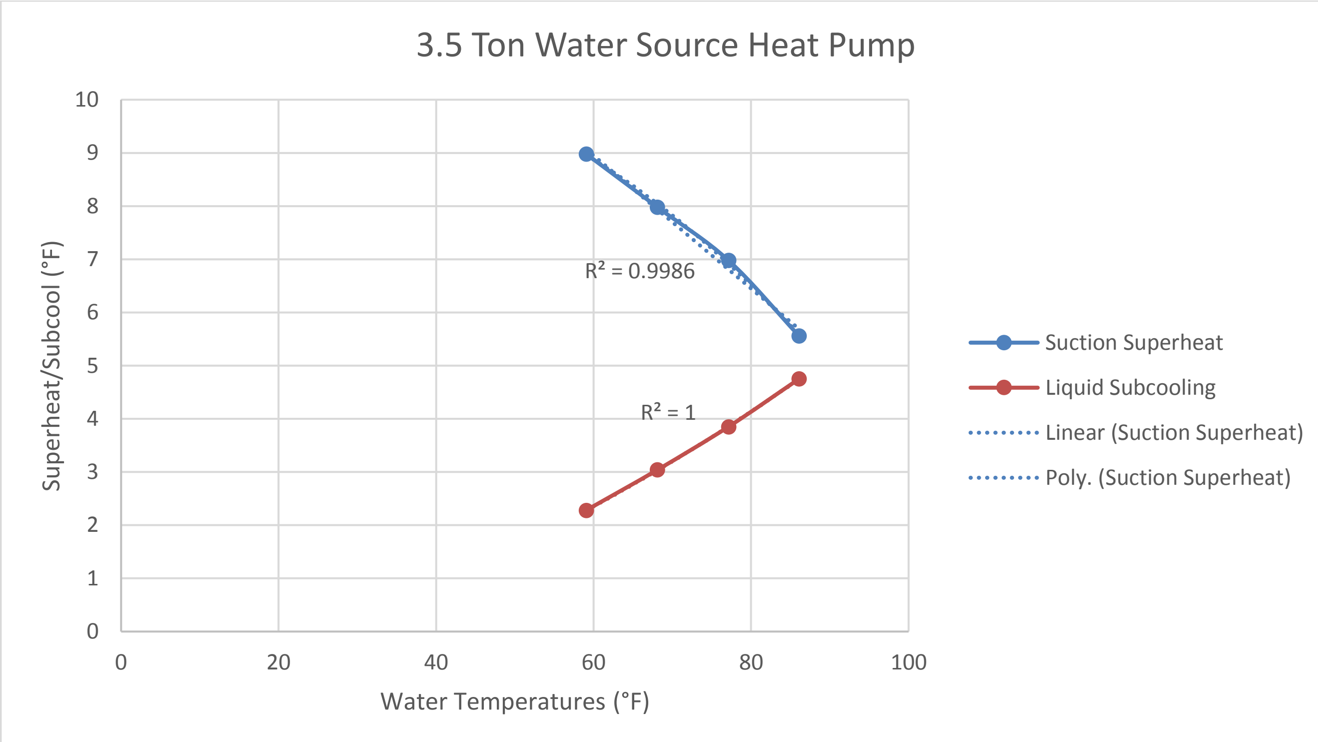
$s = -0.117t + 16.135$, where t = water temperature and s = subcooled value [For this portion of the two-part parametric equation, a linear regression model was used as there are only two data points available]

If ambient water temperature is greater than 68°F, use

$s = 0.00618t^2 – 0.844t + 36.99$, where t = water temperature and s = subcooled value

3.5 Refrigerant Ton Water Source Heat Pump

Entering Water Temperature (°F)	**Suction Superheat Level (°F)**	**Liquid Subcooled Level (°F)**
59.06	8.98	2.28
68.1	7.98	3.04
77.12	6.98	3.85
86.08	5.56	4.75



The 3.5 refrigerant ton test was conducted only across 4 water temperatures, so the trends present themselves with a lower level of confidence when compared to the 5 refrigerant ton test. Similar to the 5 refrigerant ton test, the suction superheat values are seen to trend downwards with increasing water temperatures. Also similar to the 5 refrigerant ton test, there is an upwards trend in the subcooled levels over increasing water temperatures. However, unlike the 5 refrigerant ton test, there was no downwards trend observed in the subcooled levels within this data range. This is the only test conducted that did not have both an upwards and downwards trend in the subcooled levels. This could be due to the test being conducted at a temperature range that was simply zoomed in on the upwards trend (that is, there may be a downwards trend at temperatures below the temperature range that was studied).

Equation for the suction superheat level

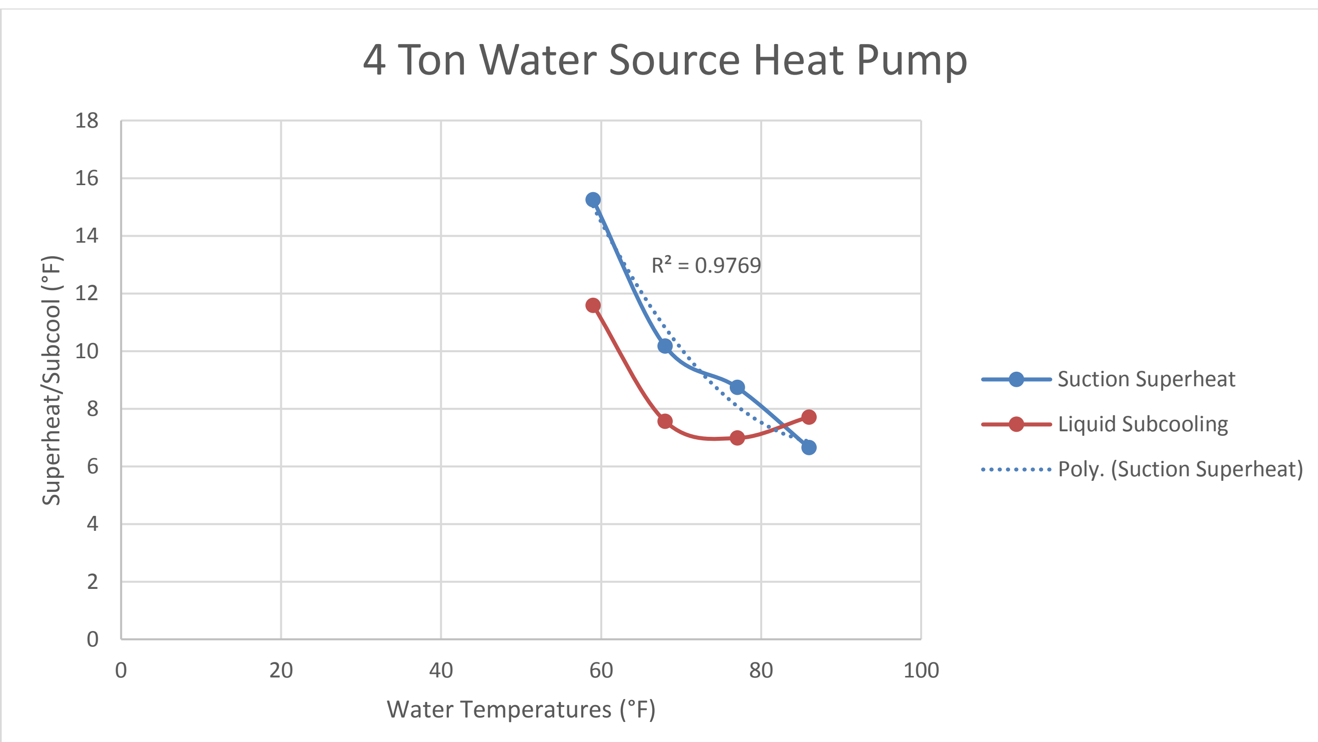
$h = -1.324t^2 + 0.0672t + 9.61$, where t = water temperature and h = superheat value

Equation for the liquid subcooled level

s = $0.000454t^2 + 0.0254t - 0.799$, where t = water temperature and s = subcooled value [This is the only test that did not result in a parametric equation for the subcooled region]

4 Refrigerant Ton Water Source Heat Pump

Entering Water Temperature (°F)	Suction Superheat Level (°F)	Liquid Subcooled Level (°F)
58.99	15.25	11.59
67.99	10.17	7.57
77.00	8.74	6.98
85.96	6.65	7.71



The 4 refrigerant ton test was conducted across only 4 water temperatures, so the trends present themselves with a lower level of confidence when compared to the 5 refrigerant ton test. Similar to the 5 refrigerant ton test, the suction superheat values are seen to trend downwards with increasing water temperatures. Also similar to the 5 refrigerant ton test, there is a downward trend initially in the subcooled levels, and after the 77°F water temperature test, the subcooled rises almost linearly from there.

Equation for the suction superheat level

$h = 0.00923t^2 - 1.641t + 79.69$, where t = water temperature and h = superheat value

Equation for the liquid subcooled level

For this situation, it is a parametric two-part equation –

If ambient water temperature is less than 77°F, use

$s = 0.0212t^2 - 3.141t + 123.05$, where t = water temperature and s = subcooled value [For this portion of the two-part parametric equation, a linear regression model was used as there are only two data points available]

If ambient water temperature is greater than 77°F, use

$s = 0.08t + 0.71$, where t = water temperature and s = subcooled value

Conclusion

By studying the results and trends as a whole, it can be concluded that the superheat values decrease over increasing water temperatures. It can also be said that the subcooled levels decrease up to a certain water temperature (this temperature depends upon the unit size), and then increase over increasing water temperatures. In addition, with all else equal, higher loads cause the superheat value to unanimously increase (regardless of water temperature or unit capacity). On the other hand, higher loads cause the subcooled values to decrease at lower water temperatures, and increase at higher water temperatures (the inflection point was seen to vary with unit size).

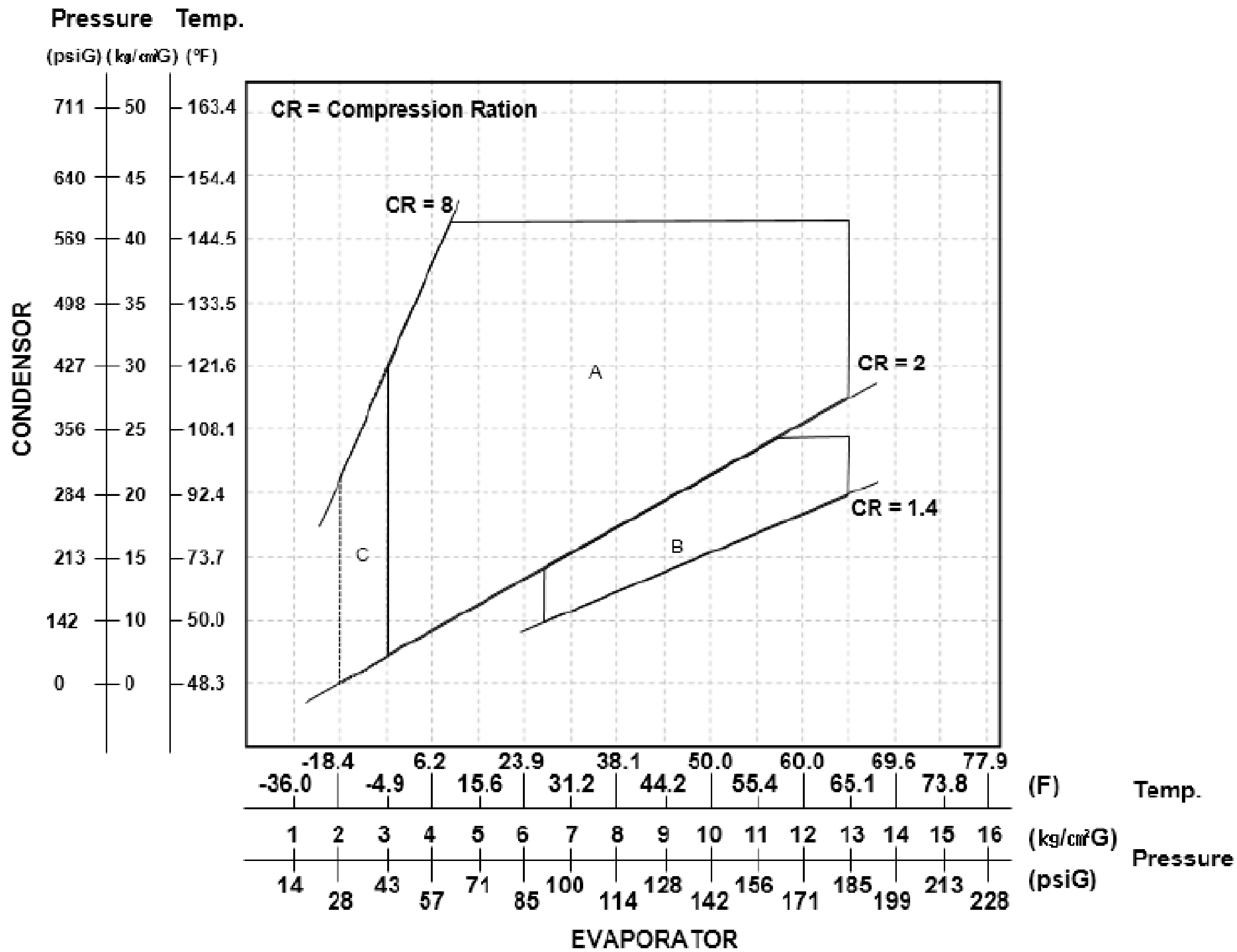
The hypothesis for the superheat values in the system (that the temperatures should stay constant due to the expansion valve) was not in concurrence with the results found. There was a definitive downwards trend in the superheat values across increasing ambient water temperatures. Upon reassessing the systems and data, this difference can be accounted for by the expansion valve "reacting" more than proportionally through the increasing water temperatures. When ambient temperature rises, the refrigerant liquid entering the expansion valve is at a higher temperature (and consequently higher pressure). Thus, the enthalpy of the refrigerant liquid entering the expansion valve is higher at higher ambient temperatures. This higher enthalpy results in more flash gas and a higher low-side pressure. This increased low-side pressure on the expansion valve will tend to result in lower superheat levels than previously due to the non-linear scaling of pressure to saturation temperature of refrigerants (from refrigerant saturation tables, a rise in pressure results in a less than proportional rise in saturation temperature). Furthermore, with each cycle, the expansion valve undergoes a process called hysteresis[3] each time it opens and closes - this can also decrease superheat values.

In addition, as compressors perform differently at different temperatures, it may have been an incorrect assumption that the work inputted by the compressor into the system is identical across the different test runs. This difference in compressor performance may have played a factor in lowering the superheat values over rising water temperatures. The polynomial equation associated with each of the units obtained through regression analysis captures the overall mathematical behavior of the decrease in superheat over rising water temperatures.

The hypothesis for the subcooled levels (that they should decrease due to the same work inputted by the compressor and enthalpy change by the condenser) was only partially in concurrence with the results found. It was found that up to a certain water temperature (which was found to vary with unit size) the subcooled levels decrease, and then begin to rise as the water temperatures are

[3] http://www.achrnews.com/articles/88190-tev-hysteresis-and-evaporator-characteristics

further increased. As the water temperature rises, there are two opposing effects on the subcooled level – firstly, the condenser will want to drive the subcooled levels down for an identical compressor work input (as discussed in the hypothesis) – secondly, a compressor's performance (compression ratio) changes with changing operating temperatures. Each compressor operates best at a certain temperature below or beyond which it does not have the same performance (in terms of compression ratio). Thus, the initial assumption that the work input by the compressor is the same throughout the tests may be flawed. At different water temperatures, the length of the diagonal line on the pressure-enthalpy curve (the compression line) may have a different length, which may result in a lower subcooled value at the end. The compressor operating map for the 1.5 refrigerant ton can be seen below – there is a definitive difference in performance between the operating points. The parametric equations associated with each of the units obtained through regression analysis capture the overall mathematical behavior of the initial decrease and the subsequent rise in subcooled levels over rising ambient water temperatures.



Area A : Normal Operating Zone

Area B : High Density Flow Zone
-During Starting within 3 minutes

Area C : During defrosting & re-starting
-Running time within 3 minutes

The compressor operating map for the compressor used in the 1.5 and 2 refrigerant ton water source heat pump units

Test Validity

From the data above, there were several points of data that presented themselves to be outliers. An analysis of several of the procedures revealed areas of the tests and analyses that may have resulted in a certain level of inaccuracy. Firstly, there may have been non-condensable fluid (such as dry air) in the system that raised head pressure of the system and made the system appear with very high subcooled and very high superheat simultaneously. It is difficult to know whether there is high superheat or subcooled due to the system naturally or whether it is being driven by impurities in the piping. Generally, high swings between runs would indicate that there may be an impurity and that the system needs to be flushed and cleaned. This was seen to occur several times, and the runs used for analysis were the ones where high swings were not observed. However, this precaution does not mean that the runs used were definitively void of non-condensable fluids in the system.

Another item that may have adversely affected the system is that of the gauges – like any experiment of this kind, miscalibrated gauges can cause severe discrepancies between the recorded results and the actual results. Similarly, if the experiment was set up wrong, or if there was any other type of user error that was not accounted for, the actual results would have a certain level of inaccuracy. Malfunctions such as reversing valves not properly sealed could also raise the suction pressure being read and lower the accuracy level of the tests. In addition, leaks anywhere in the system could negatively affect the correctness of the results. Whenever such factors were known to occur, steps were taken to mitigate the issue. However, it is not always possible to know whether such factors are occurring - thus, it is possible that there were unknown factors that were causing issues that the results did not account for.

In general, the sample size of the data sets was fairly small because conducting few test run points over many systems was given priority over many run points on a few systems. Having more data points would provide a higher level of confidence in the analysis. However, as almost identical trends were observed repeatedly across the different datasets, the tests and results may be regarded as fairly trustworthy. However, like any other analysis that relies on experimental data, more data points would always be welcome to boost the robustness of the analysis.

Overall, there are a variety of issues that could have caused the tests to have a certain level of inaccuracy. However, proper precautions were taken wherever possible and the results can be regarded as being as accurate as can be expected for a lab setting.

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