THESIS

INVESTIGATION OF INDIRECT (SECONDARY LOOP) REFRIGERATION SYSTEMS IN COMMERCIAL FOOD SERVICE BUILDINGS

Submitted by

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ABSTRACT

INVESTIGATION OF INDIRECT (SECONDARY LOOP) REFRIGERATION SYSTEMS IN COMMERCIAL FOOD SERVICE BUILDINGS

Indirect (secondary loop) refrigeration systems have recently received increased attention due to their well-known effects on reducing refrigerant losses, particularly in commercial food sales buildings. Although their effects on operating costs, particularly in terms of energy efficiency, are less definitive, there is potential that indirect refrigeration systems might offer significant energy efficiency improvements in food service buildings. The aim of this thesis was to determine the feasibility of an indirect (secondary loop) refrigeration system for a food service building, specifically a Starbucks coffee shop. Six commercial refrigeration units were installed in a laboratory setting. The units were first tested with their air-cooled condensers to establish a baseline. Then, each unit was retrofitted with a water-cooled condenser, and all six water-cooled condensers were connected in series to form a secondary loop system and tested again. The results of this laboratory testing were used to create a predictive model to estimate the payback period for installing the system in different Starbucks coffee shop locations around the country. The model predicted the major requirements for a two-year payback period to be high energy costs (>$0.22/kWh), a warm to hot climate (AC runtime > 20 hours per day), and a sufficiently large store (containing multiple large food cases or ice machines).

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BACKGROUND

In March of 2016 the Energy Information Administration (EIA) published detailed tables of their 2012 Commercial Buildings Energy Consumption Survey (CBECS). In it, energy usage of the commercial sector is characterized in terms of fuel type (e.g., natural gas, electricity, etc.) and categorized into 16 principal building activities (education, food sales, office, etc.). Furthermore, each fuel type is further broken down into end-use categories. Figure 1 below illustrates electricity end-usage in the commercial sector:

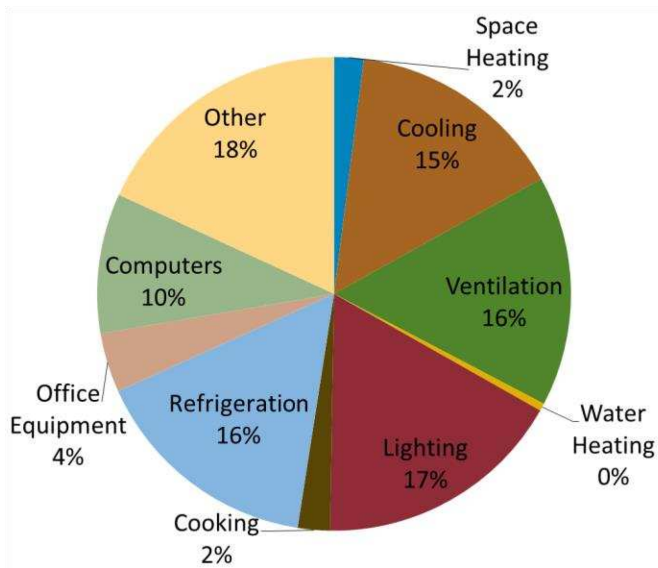


Figure 1: Categories of Energy Consumption in Commercial Sector (U.S. Energy Information Administration, 2016)

Total electricity consumption in the commercial sector in 2012 amounted to 4,241 trillion Btus, or 1.24 trillion kWh. Narrowing in on refrigeration, within commercial buildings, electricity consumption of refrigeration systems in 2012 was 670 trillion Btus (200 billion kWh). To better understand the types of buildings which use significant energy for refrigeration, the following figure shows a breakdown of refrigeration energy usage per square foot for each principal building category within the commercial sector:

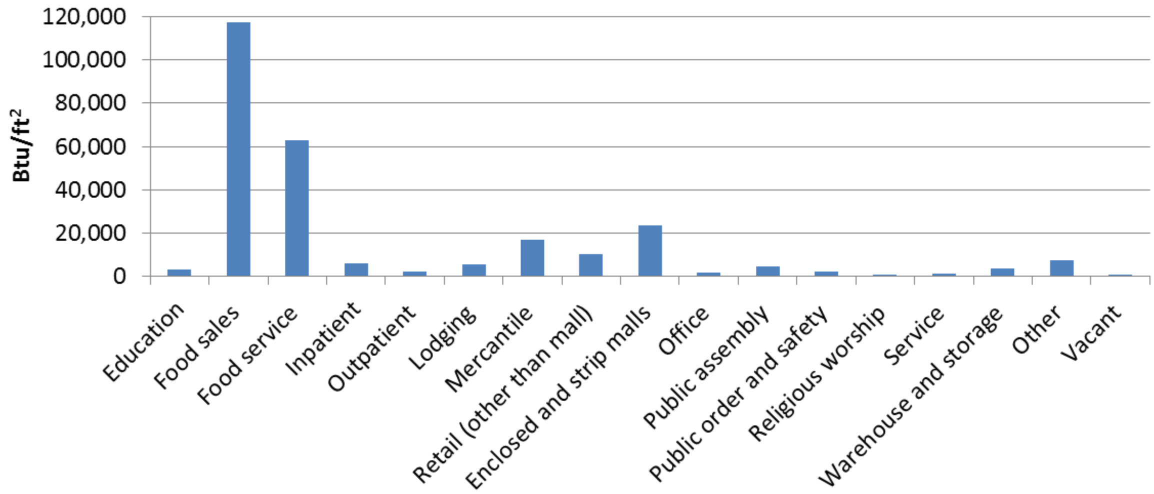


Figure 2: Refrigeration Energy Intensity by Commercial Building Type (U.S. Energy Information Administration, 2016)

It is evident from Figure 2 that both food sales and food service have exceptionally high levels of refrigeration energy usage relative to other commercial building types. This suggests that when considering energy efficiency improvements for these building types, refrigeration systems should receive particularly close consideration.

In addition to energy efficiency considerations for refrigeration systems, there is increasing concern over the negative effects of leaked refrigerant. The average U.S. supermarket uses 2,346,000 kWh annually, which equates to about 3,049,800 pounds of CO_2 emissions. (U.S. Environmental Protection Agency, 2012) By comparison, the average U.S. supermarket leaks about 875 pounds of refrigerant per year, which equates to 3,431,400 pounds of CO_2 emissions. (U.S. Environmental Protection Agency, 2012) According to Section 608 of the Clean Air Act, the EPA mandates that a refrigerator cannot legally operate with an annual leakage rate of greater than 35%. (Environmental Protection Agency, 1995)

The need for improved energy efficiency and reduced refrigerant losses has invited innovative ideas in terms of improving refrigeration systems. One of these ideas is the use of a secondary loop. This means that in addition to the fundamental components of a refrigeration cycle (the compressor, condenser, evaporator and expansion valve) there is also the addition of an entire secondary loop. This loop requires a piping network, a pump, and a heat exchanger. A diagram is shown below comparing a conventional, direct expansion loop where the refrigerant is used to directly transport heat from the cooled space to the heat rejection space, to a system containing a secondary coolant loop:

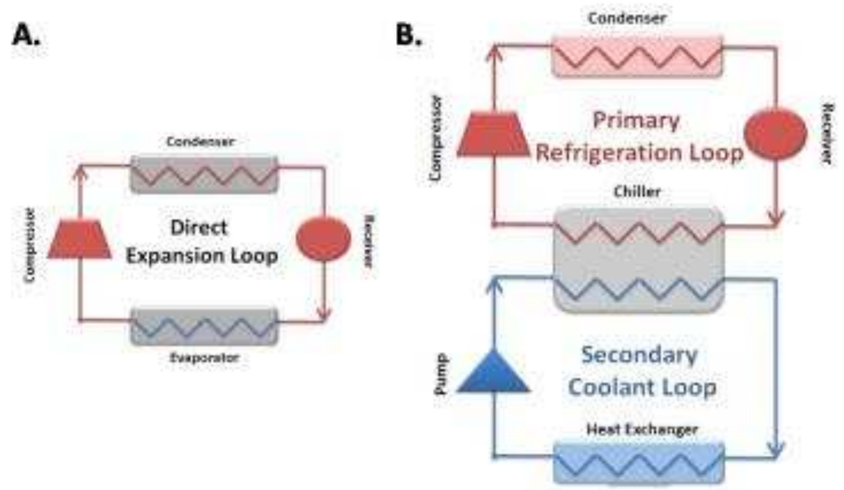


Figure 3: (A) Direct Expansion Loop (B) Secondary Refrigeration Loop (U.S. Department of Energy: Energy Efficiency and Renewable Building Technologies Program, 2009)

Refrigeration systems designed for food sales building types have significant differences compared to systems designed for food service building types. For example, the following figures demonstrate the energy consumption of both building types:

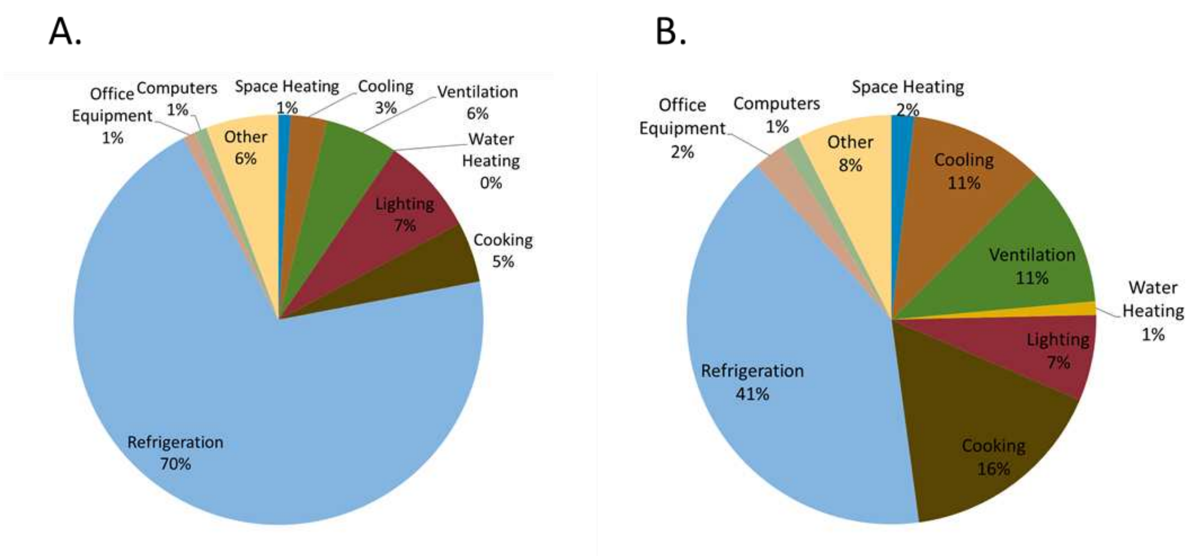


Figure 4: Energy Consumption in Commercial Buildings. (A) Food sales building types. (B) Food service building types. (U.S. Energy Information Administration, 2016)

For both building types, refrigeration comprises the largest single end-use energy consumption, but is notably higher for food sales commercial buildings. In addition to the relative energy consumption of the refrigeration systems for both building types, the design of the system itself has significant differences. For example, the major player in food sales is supermarkets. The typical design for a supermarket refrigeration system is a multiplex system, where refrigerators operate in the store area and refrigerant lines carry the refrigerant to a remote machine room which houses multiple parallel compressors. The condenser is typically located on the roof. By comparison, food service buildings include building types such as restaurants and coffee shops. These building types typically operate self-contained refrigerator units, which each house an entire refrigeration system. Considering the differences between the food service and food sales, the application of a secondary refrigeration loop in each is considered separately. In addition, indirect refrigeration systems have been explored and utilized in several applications relating to neither food service nor food sales.

Indirect (Secondary Loop) Refrigeration Systems in Dairies, Ice Rinks, and Climate-Controlled Transportation Vehicles

Indirect systems with secondary fluid circuits have long been used for systems with many units to be cooled. For large butcheries and dairies, direct systems have proven to be more expensive and complex, making indirect systems the convenient simple solution. (Effsys2 P2, 2010) Ice rinks have also been recognized as ideal candidates for indirect systems with their requirement of long lengths of tubing.

Although there is often a stigma attached to the installation and operating costs of secondary systems, there are real, demonstrated examples suggesting the systems can perform well. In 2009, the City of Brooklyn Park renovated two ice rinks to utilize indirect refrigeration loops. Although installation costs were 3.5% higher than a conventional direct expansion system, the new system requires half the energy to perform at the same capacity as the previous system. (Stevens Engineers, 2009)

Another area under investigation as a potential candidate for indirect refrigeration systems is in transport refrigeration systems. Currently, multi-temperature transport refrigeration systems almost exclusively use direct expansion systems. (Finn, 2012) A study performed by the Institute of Refrigeration demonstrated that the performance of a secondary loop refrigeration system is highly related to the choice of secondary coolant. In this study, the power consumption of the direct expansion system was lower than that of the secondary system, suggesting careful consideration is required before deciding on a system. (Finn, 2012)

Indirect (Secondary Loop) Refrigeration Systems in Food Sales Buildings

The most commonly used refrigeration system for supermarkets today is a multiplex, direct expansion system. (Baxter, 2003)These systems consist of refrigeration units operating within the store, with refrigerant lines carrying the refrigerant to a remote machine room where multiple parallel compressors are located. In addition, heat rejection is typically performed by air-cooled condensers located on the rooftop. These types of systems require thousands of meters of refrigerant piping, and have historically been designed for ease of service instead of minimizing leakage. (Baxter, 2003) Figure 5 illustrates a representative multiplex system:

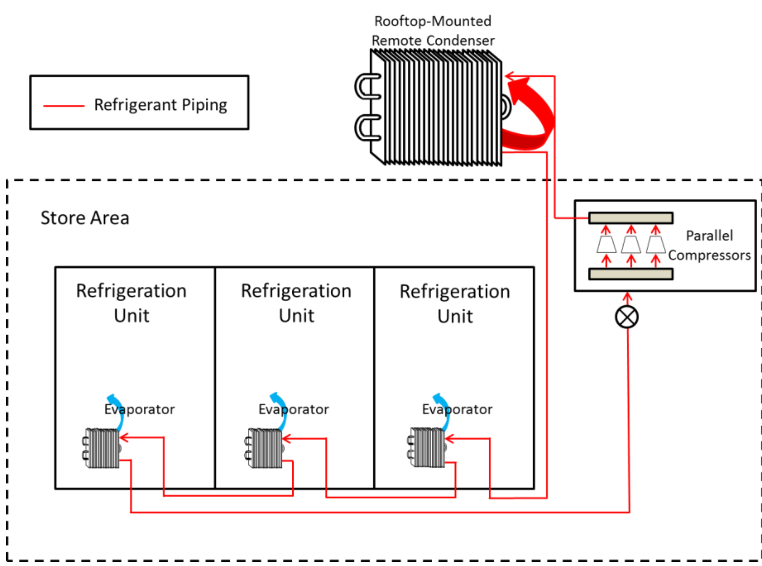


Figure 5: Multiplex Refrigeration System, Typical of Large-Scale Refrigeration Users

As previously described, growing concerns over the leakage of refrigerant is likely to shift the existing landscape of supermarket refrigeration. One alternative design is to use a secondary refrigeration loop. This would significantly reduce the length of the refrigerant line by using a heat exchanger to transfer the heat from the refrigerant to a secondary fluid. A possible configuration of a secondary refrigeration loop is shown below:

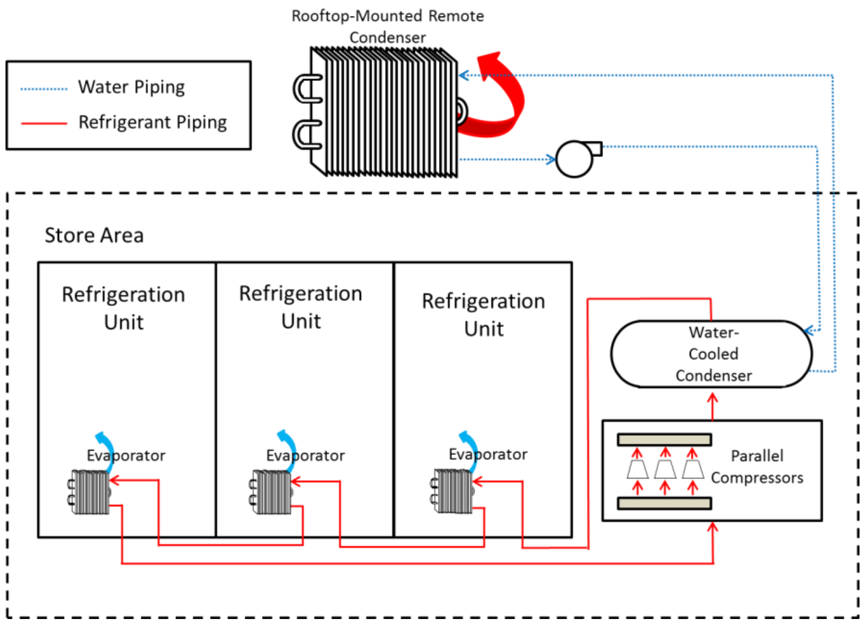


Figure 6: Secondary Loop Refrigeration System

Numerous studies have definitely demonstrated the reduction in refrigerant leakage associated with using a secondary loop instead of a conventional multiplex system. An analysis performed by Oak Ridge National Laboratory suggested a reduction in leakage of over 90%. (Baxter, 2003). Another study performed by Purdue University suggested a reduction in leakage of greater than 2/3. (Zhang, 2006) Finally, the California Energy Commission tested two similar facilities, one operating with a conventional multiplex system and the other with a secondary loop system. The results of this 9-month test demonstrated that the secondary loop system had a leakage rate that was ten times less than the multiplex system. (California Energy Commission, 2004)

Although their impacts on refrigeration leakage are well demonstrated, secondary refrigeration loops have not yet become mainstream because the operating and installation costs of these systems are commonly understood to be higher. As part of an effort to disprove this notion, there have been several studies aimed at comparing energy consumption between direct and indirect refrigeration systems.

In 1998, a study by Purdue University of two supermarkets located in North, France provided evidence that secondary loop systems are more expensive. The study compared a direct and an indirect system over the course of three weeks. The results showed significantly higher annual energy consumption for the secondary loop system than the multiplex system. (D. Clodic, 1998) Another study by Purdue University was a little more favorable to secondary loop systems, showing similar levels of operating costs between a secondary loop system and a multiplex system, but suggested that a state-of-the-are secondary loop system could outperform a multiplex system. (Zhang, 2006)

There have been a number of studies which would suggest that secondary loop systems can compete with and even outperform conventional multiplex systems. One study comparing two Canadian supermarkets demonstrated a specific energy consumption that was 8% lower for a secondary loop system versus a multiplex system. (Minea, 2007) Oak Ridge National Laboratory simulated several refrigeration systems, with the results showing greater than 10% reduction in energy consumption of the secondary loop system relative to the multiplex system. (Baxter, 2003) Hillphoenix performed a year-long study of their Second Nature refrigeration line. They demonstrated significant energy savings for a secondary loop system relative to a multiplex system. (Hill Phoenix Refrigeration Systems, 2011) Finally, the commonly referenced study performed by the California Energy Commission demonstrated savings of 4.9% for a secondary loop system. (California Energy Commission, 2004)

Despite more recent studies showing improved energy efficiency of secondary loop systems, the results are not necessarily conclusive. Because of the large number of store parameters, it's generally difficult, if not impossible, to compare the energy consumption of two or more stores. (Minea, 2007) Furthermore, many of the studies mentioned rely primarily on modeling work. For the field study performed by the California Energy Commission, the source of the savings is unclear, and potentially unrelated to the refrigeration differences. In general, due to the extra heat exchange process, all other items being equal, secondary loop systems are not expected to show substantial energy savings over other conventional systems. (U.S. Department of Energy: Energy Efficiency and Renewable Building Technologies Program, 2009) However, the Department of Energy has partnered with several major refrigeration companies to develop a secondary loop refrigeration system for supermarkets that will lower energy consumption by 25% and reduce greenhouse gas emissions by 75% by September of 2017. (Fricke, 2016)

Indirect (Secondary Loop) Refrigeration Systems in Food Service Buildings

Differing from food sales commercial buildings, in food service commercial buildings the incumbent refrigeration system is not a multiplex system, but exclusively self-contained refrigeration systems. Presently, there are no secondary loop systems employed in food service buildings. Although these building types have a substantial refrigeration load, as seen in Figure 2, the self-contained units do not pose the same risk in terms of refrigerant leakage as large multiplex systems with their extensive piping networks. As such, there has not been a substantial effort to improve these refrigeration systems to reduce the refrigerant leakage rate.

However, self-contained refrigeration systems in food service buildings do offer a unique motivation towards improving energy efficiency that is not present in food sales buildings. Multiplex systems found in supermarkets reject the waste heat from the refrigerators to the outside. Self-contained systems reject heat to the inside, creating additional heat loads which must be managed by the building's AC system. From Figure 2, food service buildings have the second highest refrigeration load per square foot of floor space. Figure 7 below shows that they also have the second highest cooling and ventilation loads:

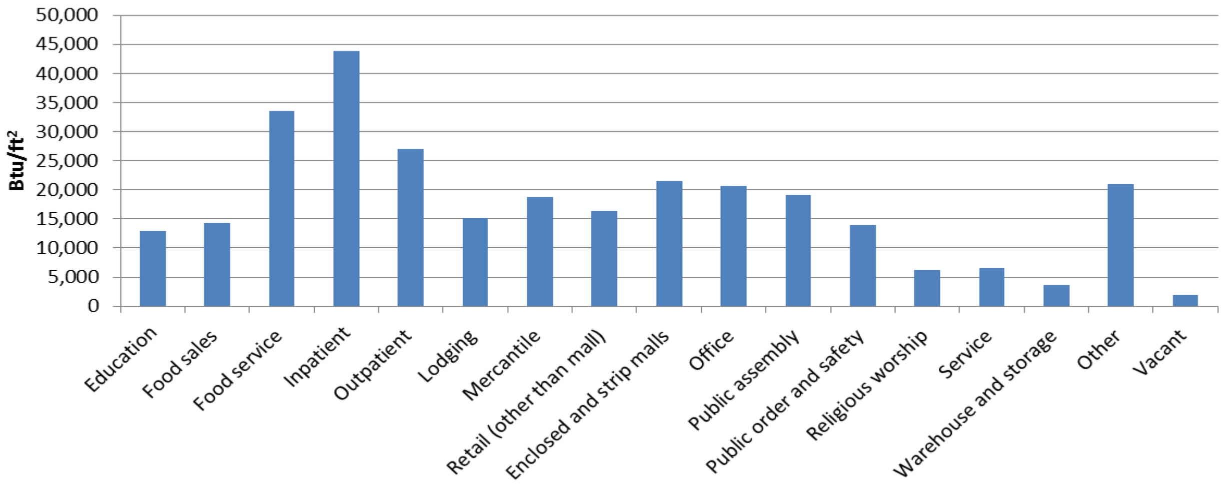


Figure 7: Cooling and Ventilation Energy Consumption by Commercial Building Type (U.S. Energy Information Administration, 2016)

Current Energy Efficiency Efforts in Food Service Buildings

There are numerous regional and national efforts towards improving energy efficiency within commercial buildings, and more specifically food service buildings. For example, as part of the Better Building Initiative, the Department of Energy partnered with companies such as Arby's restaurant Group, Inc., (Better Buildings: U.S. Department of Energy, 2015) Shari's Café & Pies, (Better Buildings: U.S. Department of Energy, 2016) and The Wendy's Company (Better Buildings: U.S. Department of Energy, 2016) to implement numerous energy efficiency measures. These measures included items such as lighting retrofits, high-efficiency HVAC retrofits, improved roof-top-unit controls, and more efficient refrigerators, with savings ranging from 25-50% of the store's annual energy bill.

In 2009 the Department of Energy published a report entitled "Energy Savings Potential and R&D Opportunities for Commercial Refrigeration." The report details commercial refrigeration energy consumption in terms of different refrigeration categories, including supermarkets refrigeration, food service equipment, beverage merchandisers, ice machines, vending machines, reach-in coolers and walk-in coolers, and describes the energy savings potential that exist for each type. The improvements described include items such as adding thicker insulation, using high-efficiency compressors, using high-efficiency fan blades, and improving refrigeration controls. (U.S. Department of Energy: Energy Efficiency and Renewable Building Technologies Program, 2009)

Also in 2009, Pacific Northwest National Laboratory published “Energy Efficiency Potential in Existing Commercial Buildings: Review of Selected Recent Studies.” This report suggested that refrigeration systems offer the second greatest potential in terms of energy savings (lighting being the first). It further detailed the types of improvements available for commercial refrigerators, such as high efficiency units and the use of variable speed drives. (Pacific Northwest Laboratories, 2009) Another DOE report published in April of 2016 entitled Energy Efficiency in Separate Tenant Spaces—A Feasibility Study discusses several technologies aimed at improving building efficiency. The technologies discussed that relate to HVAC include replacing HVAC units with higher efficiency models, improving building envelope performance, HVAC zoning and window attachments. (U.S. Department of Energy: Energy Efficiency & Renewable Energy, 2016)

Furthermore, a paper by Fisher and Karas on ice machines, one of the most significant producers of waste heat in smaller commercial buildings, focused only on the efficiency of the machines themselves, without mentioning the impacts on the AC system. (Fisher & Karas, 2012) Existing rebates are in place for upgrading refrigeration units to more efficient models, but again do not currently attempt to address the effect of the heat output of these units on the AC System. (City of Vancouver, 2012)

Potential for Secondary Loop Refrigeration

The current trend of energy efficiency efforts targeting commercial buildings typically focuses on energy equipment categories (e.g., lighting, HVAC, refrigeration, etc.) independently, without necessarily considering an integrated system design. Specifically, the interaction between a building’s refrigeration units and its HVAC system can often be such that they are operating directly against one another. The heat produced by the refrigerators has to be handled by the HVAC system. This creates a building level energy efficiency issue beyond one specific equipment category, making it a particularly difficult problem to address, and most often one that is not addressed at all. Figure 8 illustrates an example of a self-contained refrigeration system commonly used in food service:

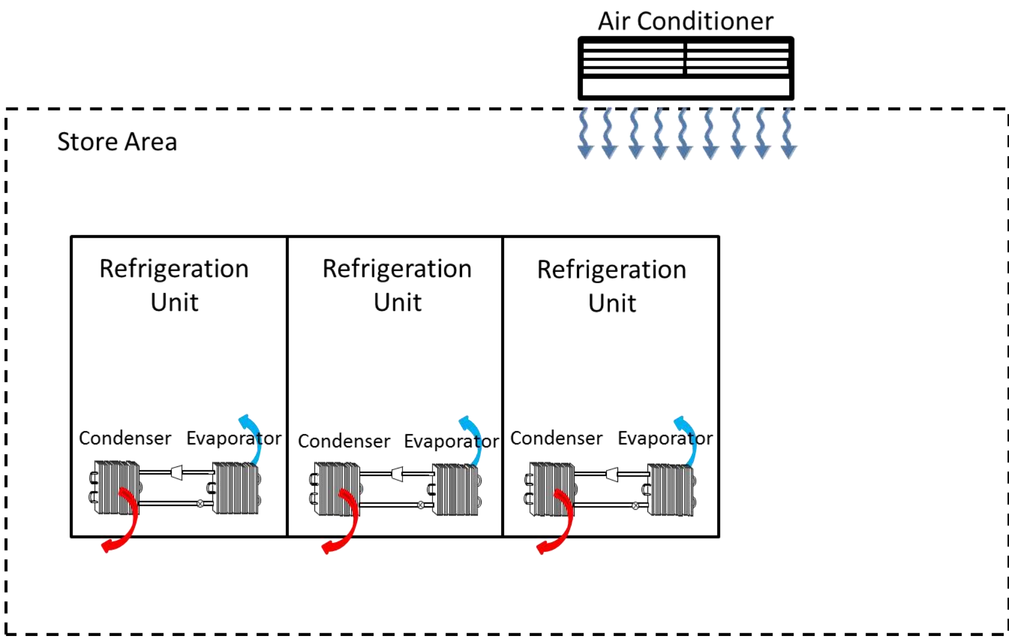


Figure 8: Self-Contained Refrigeration System, Typical of Food Service Building Types

In a published checklist describing energy saving items for commercial buildings, the National Renewable Energy Laboratory described exhaust air heat recovery as a possible option for buildings in the right climate zone and also at high utility rates. (National Renewable Energy Laboratory, 2011) One possible design to recover the exhaust heat off of the refrigerators is to replace each of the air-cooled condensers on the refrigeration units with water-cooled condensers, and connect all of the condensers in series. Using water as the secondary fluid, the heat can be taken from the machines and piped to any location, effectively making an indirect, or secondary loop, system. This design is illustrated in Figure 9.

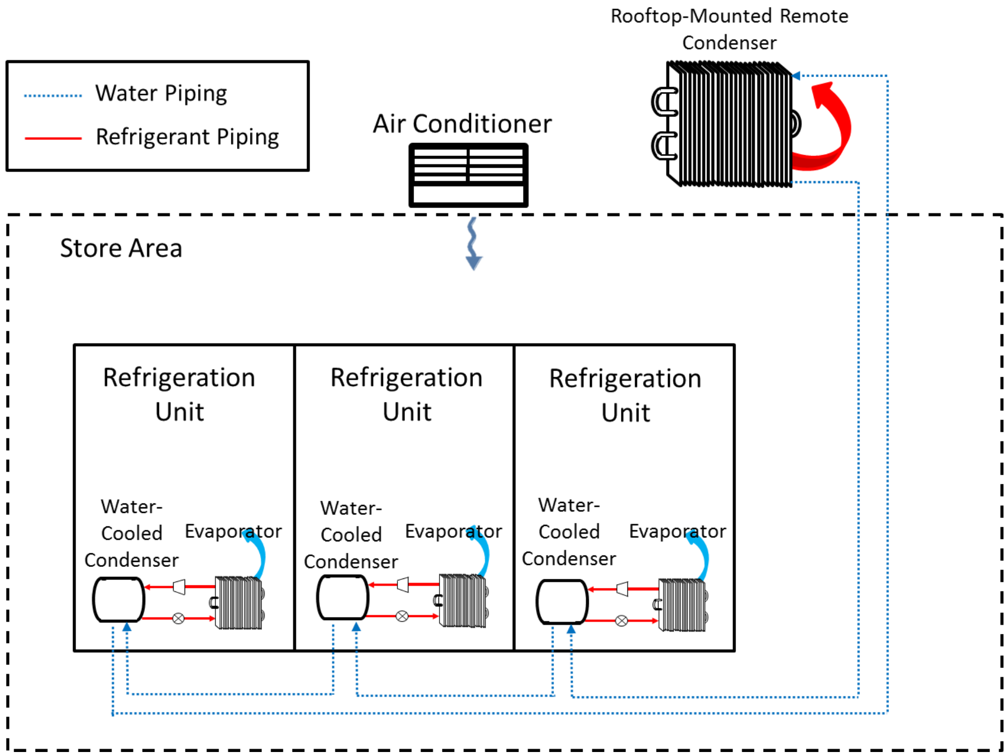


Figure 9: Secondary Loop Refrigeration System, Food Service Buildings

PROJECT DESCRIPTION

On February 6, 2015 Colorado State University entered into a Master Research and Development Agreement with Starbucks Coffee Company. As part of Task Order #3 of this agreement, CSU was tasked with the design of a potential heat recovery solution which would reduce HVAC energy consumption for a typical Starbucks store. The solution selected was a water loop heat recovery system, which would function as a secondary loop refrigeration system. The design would involve retrofitting all of the back-of-house refrigeration units and two front-of-house food coolers with water-cooled condensers and connecting them in series.

Final water loop arrangement would include: a circulation pump; a heat rejection system mounted on the roof consisting of a fan and a large condenser; two heat exchangers, one to provide preheating for the hot water heater and one to provide preheating for the coffee makers; and six water-cooled condensers, one for each refrigeration unit. This system is illustrated in Figure 10:

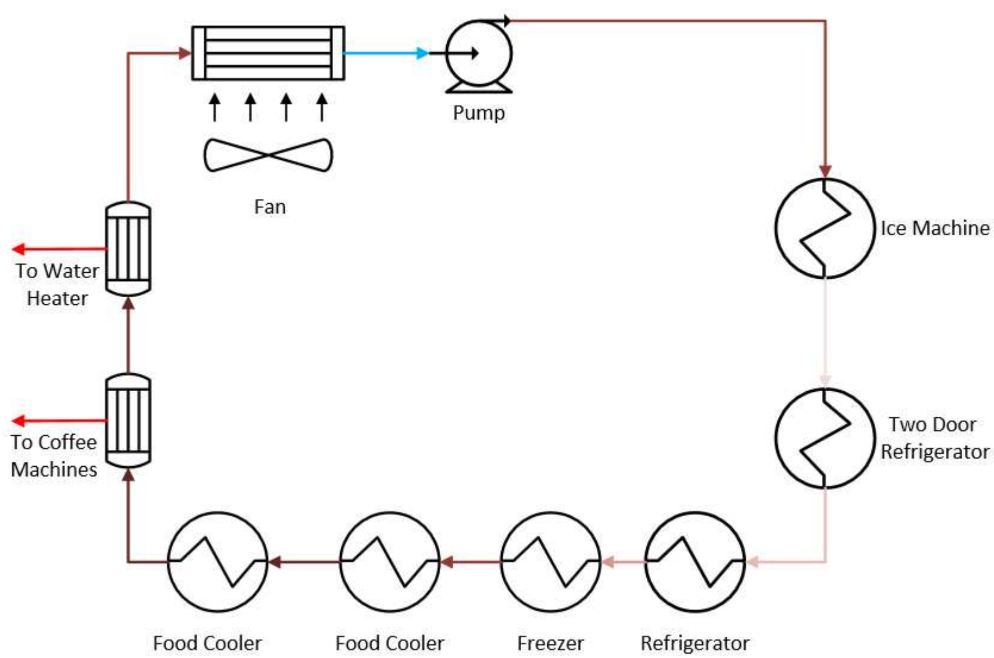


Figure 10: Starbucks Water Loop Heat Recovery System

The objectives of the design are to reduce the air conditioning load by relocating the heat produced by the refrigerators from inside the building to outside the building. In addition, prior to this heat rejection, the heat in the water loop will reduce heating energy consumption for the water heater and coffee makers by providing preheating. Finally, there is expected to be an improvement in terms of refrigeration performance associated with using water as the cooling fluid instead of air.

The installed system will also include several sensors and controls. A temperature probe will be placed at the hottest point in the loop, just before the two heat exchangers. This probe will be connected to the heat rejection system controls, and will be used to cycle the fan on and off. If the system reaches a temperature above a safety point, a solenoid valve will open, allowing cold city water to enter and cool the system. A pressure sensor will be placed after the pump to monitor pumping activity. If the pressure drops below a set point, it will indicate the pump has failed, and the solenoid valve will open.

Finally, the system will include data loggers to monitor the power consumption of each refrigerator, the HVAC system, the pump and the heat rejection fans. Several temperature loggers will be placed in the water loop to monitor heat output of each refrigeration unit.

In their 2014 and 2015 Global Responsibility Reports, Starbucks Coffee Company outlined objectives of achieving a reduction of 25% in energy consumption in each of their stores. (Starbucks Coffee Company, 2014) The Master Research and Development Agreement with CSU is in line with this goal. The ultimate aim of this agreement is the installation of the selected design option in a Starbucks store located in San Diego, CA.

Although the overall project scope is much larger, the scope of this research paper is confined to documenting and discussing the results of the following objectives related to meeting the final project goals:

(1) Create a list of potential heat recovery technologies. Develop a model for each design option to estimate its payback period.

(2) Select one model/design option to pursue for further testing/validation.

(3) Size and purchase selected system components. Install system in laboratory for testing validation.

(4) Data log system performance, including power consumption of pump, fans and all refrigeration units, as well as heat output of all refrigeration units.

In addition, a fifth objective summarizes the final project deliverable and represents the culmination of all previous efforts, including the development of the model, the design and installation of the system, and the experimental validation of the model. This overarching objective, to which all other objectives are directed towards achieving, is summarized below:

(5) Develop a finalized decision tool for estimating the payback period of installing a water loop heat recovery system at any given Starbucks store in the United States.

Objectives (1) and (2) were completed as of December, 2015.

Table 1 summarizes the preliminary modeled outputs of the water loop heat recovery system design. A detailed description of the computation of each source of savings, as well as the additional costs of operation, can be found in APPENDIX I: INITIAL MODELING.

Table 1: Initial Modeling Values for Water Loop Heat Recovery System

Cooling Savings	$2,900	/year
Cooling Load Reduction	1.6	tons
Hot water savings	$1,800	/year
Pump Electrical Use	-$100	/year
Fan Electrical Use	-$360	/year
Improved Refrigeration Efficiency	$540	/year
Net Savings	**$4,780**	**/year**

Objective (3) involved equipment sizing, final purchasing and installation in Colorado State University's Powerhouse laboratory. A detailed description of the engineering analysis used to size each component is included in APPENDIX II: EQUIPMENT SIZING. A list of the equipment purchased and installed for laboratory testing is shown in Table 2, including the manufacturer, the price and the quantity of each item:

Table 2: Equipment List

Item	Manufacturer	Model Number	Price	Quantity
Condenser Coil	Doucette Industries	CX-H-033	$67.50	1
Condenser Coil		CX-H-050	$77.50	2
Condenser Coil		CX-H-150	$137.50	2
Condenser Coil		CX-H-100	$112.50	1
Water-to-Water Heat Exhanger	Bell & Gossett	BP400-20LP	$185.95	2
Rotary Vane Pump	Procon	115B330F31XX	$454.25	1
Pump Motor	Dayton	5K339	$255	1
Air to Water Heat Exchanger	ValuTech Mechanical & Thermal Solutions	HTL 24 x 24	$384	2
Exhaust Fan	Global Industrial	T9FB1960512	$325	2
Overflow Tank	In-House	-	$35	1
Plastic Piping	ADS	NA	$40	1
Piping Insulation	Everbilt	ORS07812	$41.50	1
J-series 2-Way Solonoid Valve	Assured Automation	2036BV06T	$170	1
Temperature On/Off Controller	Omega	DP7000	$99	2
Thermocouple Probe		TC-K-U-NPT-72	$38	2
Pressure Transducer		PX309-015G5V	$225	1
Installation Accessories	Lowes	-	$430	1
		Total	**$4,290**	

The total cost shown in Table 2, $4,290, is the value used for the capital cost in the final model for estimating the payback period of the design. To estimate the cost of labor, two components were considered. First, the cost of installing all of the connections and the piping network within the store was considered. A formal quote was not yet obtained at the time of the writing of this research paper. Instead, the methodology used was to consult ProMatcher Plumbing Service to determine an hourly rate for a plumber in Fort Collins, or about $95/hr. Then, an estimate of 8 hours of labor was made based on the installation process in the laboratory and accounting for increased complexity of an in-store installation. This made the estimated cost for installing the piping and fittings in the store about $760.

The second component of the labor cost is the cost to remove the existing fan-cooled condensers on each refrigeration unit and replace them with water-cooled condensers. For lab testing, six refrigerators identical to the ones in the 14944 Starbucks store were retrofitted with water-cooled condenser. However, this process occurred in stages as equipment became available. The first three refrigerators retrofitted were the smaller, True Refrigeration units. This installation cost totaled $2,161.21. Next, the two food cases and the ice machine were retrofitted. This installation cost totaled $2,652.43. Finally, during early testing, it was discovered that the small, single door freezer was operating poorly. The cause was determined to be an incorrectly sized water-cooled condenser. The cost of re-retrofitting this unit was $521.21.

For the preliminary model, a value of $4,000 was decided on for the cost to retrofit all six units. This resulted in a total installation cost of $9,050. For an annual savings of $4,780, this translated to a payback period of 1.9 years. Objective (4) focuses on the testing and data collection of the installed system in a laboratory environment. Testing was performed from February, 2016 through April, 2016. This objective is outlined in the section entitled LABORATORY TESTING. Objective (5), the final objective, focuses on analyzing the results of laboratory testing and applying them to validate and modify the initial model.

LABORATORY TESTING

The system was installed in the first floor of the Colorado State University Powerhouse for laboratory testing. In addition to the components listed in Table 2, six refrigerators were leased from their respective manufacturers. The final installed system is shown in Figure 11:

Figure 11: Installed System

There were two key goals for the laboratory testing. The first goal was to verify the functionality of the system and each of its components. In order to size and order each piece of equipment, an engineering analysis based on the manufacturer-provided specifications was used. After verification of functionality, the performance characteristics of each component were measured or tested to improve the original model used in sizing. Original sizing and selection of each component, as well as the measured or tested performance characteristics, are outlined in APPENDIX II: EQUIPMENT SIZING.

The second goal was to determine the performance of the water loop heat recovery system compared to the incumbent self-contained, air-cooled refrigeration system, and update the initial model with experimental data. To achieve this goal, each refrigerator was tested in the laboratory using its normal air-cooled condenser to provide a comparative baseline. Then each refrigerator was retrofitted with a water-cooled condenser and tested again. Heat output of the water-cooled units in the laboratory was also data logged during each testing period.

In addition, the power consumption of each refrigerator in the store was data logged. By comparison, the same refrigerator models operating in the laboratory showed significantly higher power consumption when operated in the store. A second comparative baseline was established using the logged data of the air-cooled units in the store.

In total, three sets of power consumption data for each refrigerator model were collected, including the laboratory air-cooled baseline, the water-cooled performance testing, and the in-store air-cooled baseline, each described in the section 'Power Testing'. The testing procedures for determining the heat output of each refrigerator are outlined in the section entitled 'Heat Output Testing'. Because of its unique batch-wise process, testing of the ice machine is described individually in the section entitled 'Ice Machine Testing'. Finally, in order to understand the response of refrigerators to a changing load condition, an experimental setup is described in the section entitled 'Refrigerator Response to Varying Load Condition'. 'Testing Procedures' describes the set up for each testing protocol, and 'Testing Results' demonstrates the results of each testing procedure.

Testing Procedures

Each refrigerator was tested at three water loop temperatures. To set water loop temperature, a thermocouple was inserted into the loop immediately prior to the heat rejection system to measure the highest loop temperature and connected to an Omega DP7000 temperature controller. The temperature controller was also connected to the heat rejection fan, and programmed to switch the fan on when the measured temperature exceeded the set point by 1°F, and power off when the measured temperature was 1°F below the set point. Testing for each water loop temperature set point was conducted over a period of five hours to ensure the refrigerator had time to reach a steady state condition.

During each testing interval, the power consumption of each refrigerator was data logged. For the three smaller refrigerators (1-door freezer, 1-door refrigerator, 2-door refrigerator), power was data logged using Onset's HOBO plug load data loggers. For the three larger units (horizontal food case, vertical food case, ice machine), power was data logged using an ELITEpro XC Dent Instruments Energy Logger. In addition, a HOBO Tidbit v2 water temperature data logger was placed immediately before and after each water-cooled condenser inside the water loop in insertion points like the one in Figure 12 to record the water temperature entering and exiting the condenser. Data loggers recorded data every second.

Figure 12: Temperature Logger Insertion Point

It was initially thought that the discrepancy between similar refrigerators operating in the store versus in the lab was caused by differences in loading conditions (frequency of door openings). Several experiments were conducted to test this theory. The first two are outlined in APPENDIX IV: ADDITIONAL EXPERIMENTATION. A the third experiment was designed to determine a relationship between each of the three smaller refrigerator's power consumption and heat output versus varying load conditions. It was entitled 'Refrigerator Response to Varying Load Condition' and is outlined below.

- Three load conditions tested during three time intervals, each two hours long.
- Low load corresponded to opening each refrigerator once every 15 minutes, or 8 times over two hours, for 10 seconds each time. Medium load corresponded to opening each refrigerator once every 10 minutes, or 12 times over two hours, for 15 seconds each time. High load corresponded to opening each refrigerator once every 6 minutes, or 20 times over two hours, for 20 seconds each time.
- Water loop temperature was set to 95°F to approximate in-store conditions.
- The water loop flow rate was fixed at 5.5 gallons per minute.
- Ambient air temperature was 70°F through the duration of the experiment.
- Power consumption of each refrigerator was data logged. Water loop temperature entering and exiting each refrigerator's condenser coil was data logged.

For all testing procedures, laboratory room temperature during testing was maintained at 70°F. Preliminary testing and analysis of refrigeration duty cycle suggested 95°F as the upper limit of testing. For temperatures significantly above 100°F, most refrigerators tended to operate with a 100% duty cycle.

Testing Results

Power Testing

An analysis of the logged power consumption data was done to determine four key variables, each described below:

Duty Cycle—Percentage of time the refrigerator spends in the loaded state (compressor on).

Loaded Power—Power draw of the refrigerator when the compressor is on.

Unloaded Power—Power draw of the refrigerator when the compressor is off.

Average Power—Average power consumption taken over an entire refrigeration cycle.

Laboratory Air-Cooled Baseline

Data logging of the five refrigeration units (the ice machine is not included) in the laboratory operating with air-cooled condensers and subsequent data analysis resulted in the following table:

Table 3: Laboratory Air-Cooled Baseline, Data Logged Values

	1-Door Freezer	1-Door Refrigerator	2-Door Refrigerator	Vertical Food Case	Horizontal Food Case
Duty Cycle	39%	28%	24%	68%	35%
Loaded Power, W	456	297	596	1,056	1,496
Unloaded Power, W	27	31	56	110	215
Average Power, W	195	105	185	761	641

In-Store Air-Cooled Baseline

Data logging of the three smaller refrigerators operating with air-cooled condensers in the 14944 Starbucks store and subsequent data analysis resulted in the following table:

Table 4: In-Store Baseline, Data Logged Values

	1-Door Freezer	1-Door Refrigerator	2-Door Refrigerator
Duty Cycle	50%	34%	54%
Loaded Power, W	507	306	768
Unloaded Power, W	29	0.1	103
Average Power, W	268	126	460

Water-Cooled Testing

The power consumption of each refrigerator was determined for three temperature increments. The ice machine was excluded from this testing due to its batch-wise process. Table 5 and Figure 13 display the results of this testing, and Table 6 contains the predictive equations derived from the line of best fit for each data set. For each of the smaller three refrigerators, each point on the graph represents an entire refrigeration cycle, with both the heat output and the entering water temperature into the condenser coil averaged over the cycle. For the two food cases, each point represents an average taken over a testing period of four hours. A more detailed look at the results of each individual unit's testing is shown in APPENDIX III: DETAILED TESTING RESULTS.

Table 5: Results of Power Consumption Data Logging for Three Water Loop Temperatures

Refrigerator	Water Loop Temperature, °F	Loaded Power, W	Duty Cycle	Average Power, W
1-Door Freezer	79	386	46%	193
	87	403	49%	210
	99	425	54%	240
1-Door Refrigerator	79	278	19%	78
	88	290	20%	82
	99	305	23%	93
2-Door Refrigerator	78	561	23%	171
	87	578	27%	198
	97	599	32%	230
Horizontal Food Case	76	1,445	35%	616
	85	1,496	35%	661
	91	1,500	35%	665
Vertical Food Case	79	1,116	100%	1,116
	83	1,144	100%	1,144
	89	1,263	100%	1,263
Ice Machine	84	1,480	NA	1,480
	97	1,330	NA	1,330

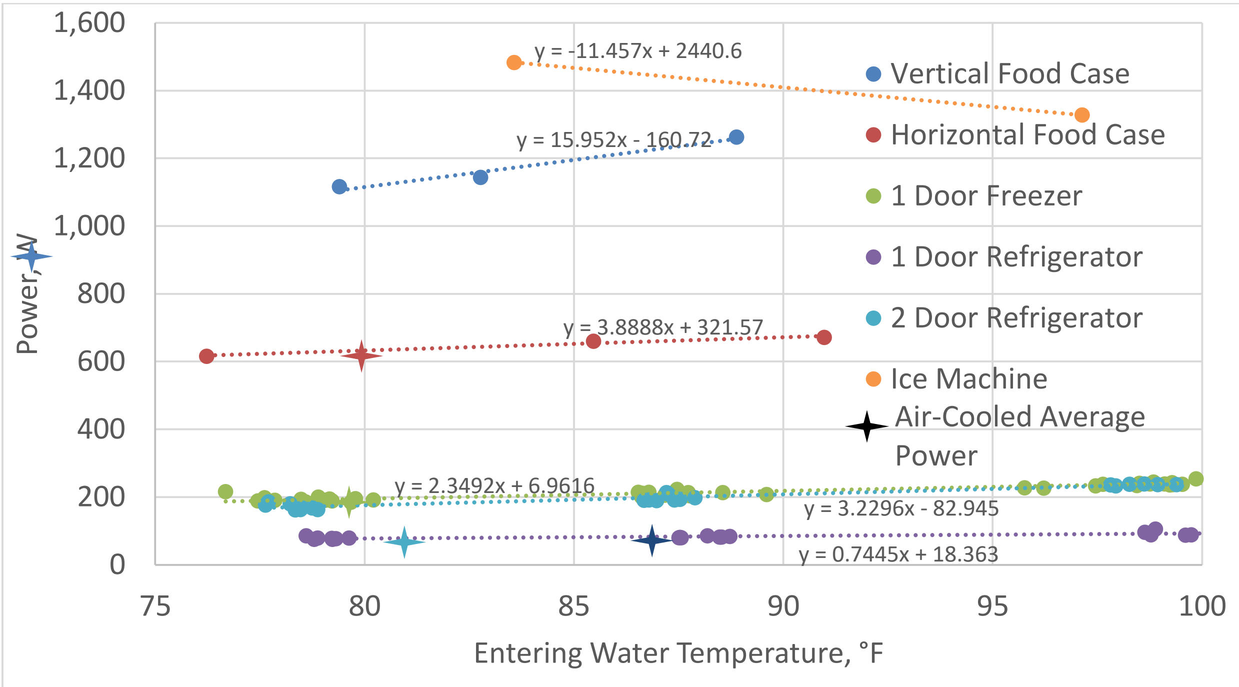


Figure 13: Refrigerator Average Power Consumption Response to Varying Water Loop Temperatures

Table 6: Refrigerator Power Consumption Predictive Equations

Refrigeration Unit	Predictive Equation
1-Door Freezer	y = 2.3492+6.96.6
1-Door Refrigerator	y = 0.7445x+18.363
2-Door Refrigerator	y = 3.2296x-82.945
Horizontal Food Case	y = 3.8888x+321.57
Vertical Food Case	y = 15.952x-160.72
Ice Machine	y = 11.457x+2440.6

Heat Output Testing

The heat output of each refrigerator was determined for three temperature increments. Table 7 and Figure 14 display the results of this testing, and Table 8 contains the predictive equations derived from the line of best fit for each data set. The key variables for heat output testing are described below:

Heat Output— Heat output of each refrigerator was calculated as follows:

$$\dot{q} = \dot{m} C_p \Delta T$$

Where,

$\dot{q}$ = heat transfer rate, Btu/hr

$\dot{m}$ = mass flow rate of water, 5.5 ga/min or 2,752 lbm/h

C_p = heat capacity of water, 1 Btu/lbm-°F

ΔT = exiting water temperature minus entering water temperature

This computation was performed for each second of data logging. Heat output per refrigeration cycle was obtained by averaging all of the calculated values of heat output for the given cycle.

The data loggers used to record the water temperature entering and exiting condenser coils have an accuracy of ±0.38°F. Table 7 shows the error bounds for the measured temperature values and the computed heat output values.

Table 7: Results of Heat Output Data Logging for Three Water Loop Temperatures

Refrigerator	Temperature Entering Condenser, °F	Temperature Exiting Condenser, °F	Heat Output, Btu/hr
1-Door Freezer	78.69 ± 0.38	79.05 ± 0.38	1,008 ± 2,090
	87.29 ± 0.38	87.64 ± 0.38	956 ± 2,090
	98.6 ± 0.38	98.91 ± 0.38	847 ± 2,090
1-Door Refrigerator	79.1 ± 0.38	79.27 ± 0.38	486 ± 2,090
	88.07 ± 0.38	88.19 ± 0.38	329 ± 2,090
	99.28 ± 0.38	99.36 ± 0.38	227 ± 2,090
2-Door Refrigerator	78.31 ± 0.38	78.7 ± 0.38	847 ± 2,090
	87.19 ± 0.38	87.59 ± 0.38	1103 ± 2,090
	96.91 ± 0.38	97.32 ± 0.38	1,116 ± 2,090
Horizontal Food Case	76.23 ± 0.38	77.96 ± 0.38	4,768 ± 2,090
	85.47 ± 0.38	86.78 ± 0.38	3,623 ± 2,090
	90.98 ± 0.38	92.13 ± 0.38	3,178 ± 2,090
Vertical Food Case	79.4 ± 0.38	82.22 ± 0.38	7,772 ± 2,090
	82.77 ± 0.38	85.8 ± 0.38	8,336 ± 2,090
	88.88 ± 0.38	91.96 ± 0.38	8,465 ± 2,090
Ice Machine	83.57 ± 0.38	87.83 ± 0.38	11,713 ± 2,090
	97.14 ± 0.38	100.91 ± 0.38	10,384 ± 2,090

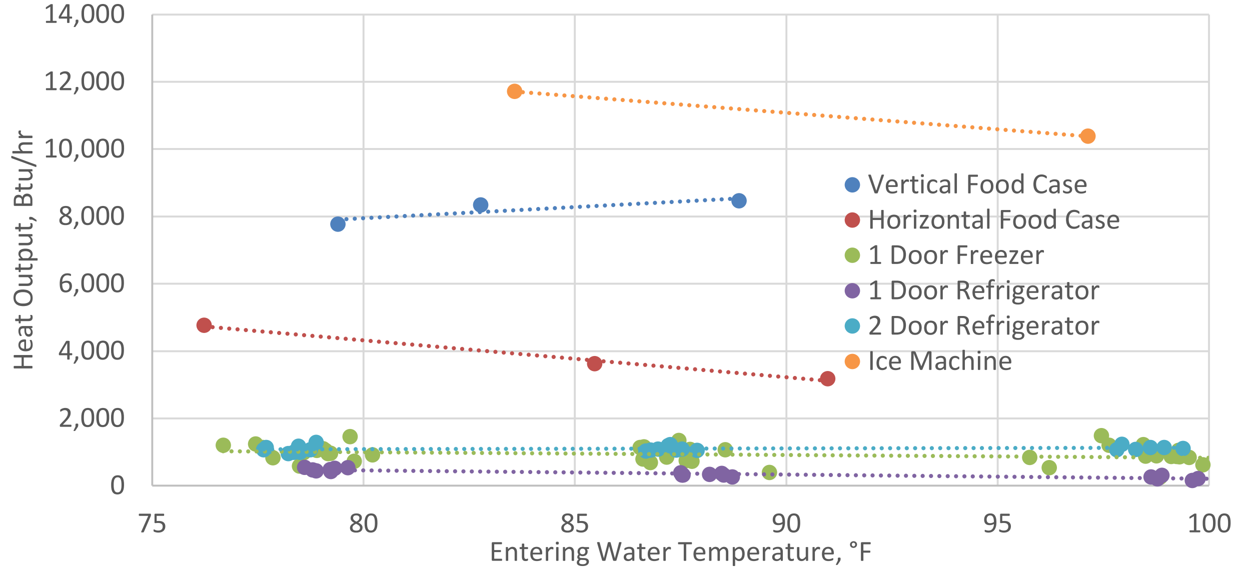


Figure 14: Refrigerator Heat Output vs. Water Loop Temperature

Table 8: Refrigerator Heat Output Predictive Equations

Refrigeration Unit	Predictive Equation
1-Door Freezer	y = -8.4422x+1,670.7
1-Door Refrigerator	y = -12.815x+1,484.6
2-Door Refrigerator	y = 2.2589x+902.53
Horizontal Food Case	y = -109.42x+13,072
Vertical Food Case	y = 66.84x+2,597.5
Ice Machine	y = -97.985x+19,902

Ice Machine Testing

Operation of the ice machine is different than operation of the other refrigerators in that it is a batch-wise process. The ice machine only consumes power when it is producing ice. Therefore, the average power consumption of the unit is directly related to demand for ice.

To estimate the daily demand for ice of a Starbucks store, six 1-hour visits randomly selected over the course of a week were made to the 14944 store. During these visits, the number of five gallon buckets of ice taken from the ice machine was counted. The average over the six visits was taken, then projected over an entire day. This number was used in the final modeling as the daily ice demand.

Testing of the ice machine in the laboratory was limited by the heat output of the unit. A water loop temperature of under 80°F was unable to be attained while operating the ice machine. Two water loop temperature set points were used, including an average entering water temperature of 84°F and 97°F. For each set point, the following values were logged or measured:

- Time per batch
- Weight per batch
- Power consumption
- Heat output

According to the manufacturer, the energy consumption and heat output of the unit depends only on the number of batches. The number of batches, in turn, is affected by environmental conditions, but the energy requirements per batch are constant. This was verified in the lab. Table 9 demonstrates the results of the ice machine testing:

Table 9: Ice Machine Testing

	Test #1	Test #2	Manufacturer Data
Water Loop Temperature, °F	97.1	83.6	70
Average Power, kW	1.48	1.33	1.53
Average Heat Output, Btu/hr	11,713	10,384	15,355
Number of Batches per Hour	3.9	4.4	4.6
Weight of Batch, lbs	7.8	7.8	7.8
Hourly Ice Production, lbs	30.7	34.7	35.7
Daily Ice Production, lbs	737	832	856
Daily Energy Usage, kWh	31.9	35.6	36.8
kWh/lb	**0.043**	**0.043**	**0.043**

Six visits to the 14944 Starbucks store were conducted at random times in the week. From these visits, the average hourly ice usage was determined to be about 9 gallons. This would require an average power consumption of 1.82 kW for an air-cooled unit, or 43.8 daily kWh. By comparison, a water-cooled unit would require an average power consumption of 1.5 kW, and 36.2 kWh of daily energy usage. This equates to an energy reduction of over 18%.

Manufacturer data was used to develop the baseline for the ice machine. The table below shows a comparison of the manufacturer-supplied data for an air-cooled ice machine and a water-cooled ice machine:

Table 10: Ice Machine Performance Data

	kWh per 100 lbs of ice	Heat Output, Btu/hr
Air-Cooled	5.2	16,024
Water-Cooled	4.3	15,355

Refrigerator Response to Varying Load Conditions

The response of each of the smaller refrigerators to low (4 openings per hour, 10 seconds per opening), medium (6 openings per hour, 15 seconds per opening), and high (10 openings per hour, 20 seconds per opening) load conditions are shown below in Table 11:

Table 11: Response of Refrigerators to Varying Load Conditions, Water Loop Temperature 95°F

	1-Door Freezer			1-Door Refrigerator			2 Door Refrigerator		
	Low	Med.	High	Low	Med.	High	Low	Med.	High
Duty Cycle	75%	84%	98%	35%	35%	63%	44%	57%	91%
Loaded Power, W	426	404	393	299	299	302	583	579	591
Unloaded Power, W	26	28	29	32	32	34	57	56	57
Average Power, W	324	344	384	124	127	202	288	354	543
Average Heat Output, Btu/hr	1,324	1,677	1,786	767	810	1,575	1,598	2,076	3,560

Discussion

Referring to Table 11, it is clear that each refrigerator's duty cycle is significantly impacted by changes in loading conditions. By contrast, the loaded power of each refrigerator seems largely unaffected by changes in load condition. This would seem to indicate that refrigerator loading primarily affects the refrigeration cycles, or more specifically the compressor's duration of operation.

Table 5 demonstrates the effects of changing water loop temperature on each refrigerator's performance. A hotter water loop temperature reduces the effectiveness of the heat transfer from the condenser. As expected, Table 5 shows that as water loop temperature increases, the refrigerator's duty cycle also increases. However, the increase is not as substantial as in the case of the varying load conditions, indicating that refrigerator load condition is more affective of duty cycle than the heat transfer conditions of the condenser. But in the case of varying water loop temperatures, each refrigerator's loaded power draw exhibited a strong relationship to the temperature of the water, indicating that the power draw of the compressor is primarily affected by changes to conditions around the condenser.

Both the duty cycle and the loaded power draw of the compressor differ substantially between laboratory units and in-store units. Most likely the difference arises from a combination of different load conditions and differences in heat transfer conditions for the condenser. In addition, the units operating in the store have potentially aged significantly. This likely increases the risk of refrigerant leakage, compressor wear, and dust on the condenser, all of which can affect the power draw of the units.

FINAL MODEL/DECISION TOOL

Laboratory testing provided reliable data in terms of comparing the performance of each refrigerator operating with an air-cooled condenser versus with a water-cooled condenser. However, the source of discrepancy between similar refrigerators operating in the lab and in the store, as highlighted in Table 3 and Table 4, was not immediately known. In-store refrigerators were observed to operate with significantly higher power consumption than laboratory refrigerators. Furthermore, as seen in Table 7 and Table 11, the values for heat output of each refrigerator were seen to vary substantially for different operating conditions. Since the heat output of in-store, air-cooled refrigerators was not directly measurable, understanding the difference in operating conditions between in-store and in-lab units was important in terms of modeling heat output of the in-store units.

Three distinct models are considered to help understand the differences between the in-store and in-lab refrigerators, and are each described in 'Refrigerator Heat Output Modeling Approaches'. The final modeling approach that is used, and the subsequent savings it predicts, are described in 'Selected Approach and Resulting Savings'. Finally, the decision tool outcomes are described in 'Decision Tool Outcomes'.

Refrigerator Heat Output Modeling Approaches

According to the laboratory testing results, both power consumption and heat output of a refrigerator vary significantly with changes in both refrigerator loading and environmental conditions. The major source of savings for the water loop heat recovery system, the reduced air conditioning load associated with the heat removal, is entirely a function of the heat output of the air-cooled refrigerators in the store. However, direct data for the heat output of the air-cooled units was not available. Instead, three modeling approaches were considered to determine air-cooled heat output.

Approach 1--Refrigeration Loading Accounts for 100% of Discrepancy between In-store and In-lab Units

It was initially hypothesized that the difference in power consumption between in-lab refrigerators and in-store refrigerators is primarily due to differences in loading conditions. For example, the in-store units have their doors regularly opened, and have items like milk and fruit regularly added to them. The first modeling approach assumes that 100% of the discrepancy between in-store and in-lab units is due to load conditions.

The results for testing each refrigerator's response to varying load conditions, shown in Table 11, provide each refrigerator's average power consumption as a function of load condition. For modeling approach #1, a refrigerator loading condition is assigned to each laboratory refrigerator such that its average power draw will equal the average power draw of the similar in-store refrigerator. Then, for this load condition, Table 11 also provides each refrigerator's duty cycle, loaded power, and heat output. Finally, in order to provide a basis for comparison, both the loaded power and the duty cycle of the laboratory refrigerator for the given load condition are compared to the actual loaded power and duty cycle of the in-store refrigerator. This process is outlined in Table 12.

Table 12: Required Laboratory Load Condition to Match In-Store Average Power, Water Loop Temperature 95°F

Unit	Laboratory Loading Condition to Match In-Store Average Power	Predicted Duty Cycle	Actual In-Store Duty Cycle	Predicted Loaded Power, W	Actual In-Store Loaded Power, W	Predicted Heat Output, Btu/hr
1-Door Freezer	< Low	63%	50%	407	507	1,167
1-Door Refrigerator	Low	35%	34%	300	306	767
2-Door Refrigerator	Med-High	76%	54%	584	768	2,927

The loaded power draw of the in-lab units was significantly lower than the loaded power draw of the in-store units. This observation challenges the first modeling approach, since it was also observed that loaded power consumption is not significantly impacted by changes in load condition (see APPENDIX IV: ADDITIONAL EXPERIMENTATION). Therefore, this first modeling approach tend to over predict the duty cycle to compensate for its non-inclusion of loaded power changes, as seen in Table 12.

Approach 2--Condenser Heat Transfer Differences Account for 100% of Discrepancy between In-store and In-lab Units

One of the key discrepancies between the laboratory refrigerators and the in-store refrigerators is their respective set ups. The laboratory refrigerators sit in the middle of a large, well ventilated room, far from the wall, with the air kept at a consistent 70°F. By contrast, the in-store units are located in a tight room, such that the condenser fans blow directly against a wall, and the air temperature is around 80°F. As an additional observation, since the power consumption of the in-store food cases was unable to be logged, the duty cycle was instead monitored. During multiple site visits, the vertical food case was observed to operate with a duty cycle of 100%, compared to a duty cycle of 68% for the same unit operating in the lab.

The second modeling approach follows the observation that although a refrigerator's loaded power does not change with load condition, it does change with differences in water loop temperature. For every refrigerator, as water loop temperature was increased, the loaded power of the refrigerator increased as well. Changing water loop temperature is representative of changing environmental conditions. Therefore, modeling approach #2 assumes the difference in heat transfer of the condenser associated with differences in environmental conditions accounts for 100% of the discrepancy between in-store and in-lab units.

APPENDIX III: DETAILED TESTING RESULTS demonstrates duty cycle, average power, loaded power and heat output of each refrigerator as a function of water loop temperature. Each figure contains an empirical equation for predicting the variable of interest. For modeling approach #2, a comparison was made between the loaded power consumption of the in-store units versus the loaded power consumption of the in-lab units. As previously mentioned, the in-store units operate with a significantly higher loaded power draw than the in-store units. Following this observation, for each refrigerator, the graph showing loaded power versus water loop temperature was consulted to determine a water loop temperature such that the water-cooled unit in the lab would have an equal loaded power draw to a similar unit operating in the store. Then, with this water loop temperature, an average power was predicted using the graph of each refrigerator's average power versus water loop temperature. Finally, this process was repeated for heat output of each refrigerator. The results of this modeling approach are outlined in Table 13.

Table 13: Projected Values Resulting from Required Laboratory Water Loop Temperature to Match In-Store Loaded Power

Unit	Required Water Loop Temperature to Achieve In-Store Loaded Power, °F	Predicted Duty Cycle	Actual In-Store Duty Cycle	Predicted Average Power, W	Actual In-Store Average Power, W	Predicted Heat Output, Btu/hr
1-Door Freezer	142	69%	50%	341	268	472
1-Door Refrigerator	100	22%	34%	93	126	203
2-Door Refrigerator	180	73%	54%	498	460	1,309

In this case the average powers predicted by the model are fairly close to the in-store values, but there still exists enough of a difference to suggest additional sources beyond environmental conditions. An important observation is that the heat output predicted by the first approach versus the heat output predicted by the second method differ substantially.

Approach 3--Manufacturer-Supplied Data

A third modeling approach involved calling the manufacturer of each refrigerator and inquiring about its expected heat load. For the three smaller refrigerators (1-door freezer, 1-door refrigerator, 2-door refrigerator) and the ice machine, this information was readily available. However, for the two food cases, the manufacturers were unwilling to divulge any operating information on the specific models for proprietary reasons.

Selected Approach and Resulting Savings

For the three smaller refrigeration units, the uncertainties surrounding the discrepancy between laboratory values and in-store values, combined with the large error bounds found in the computation of heat output, were determined to be too great to allow for sufficient confidence in any calculated value for heat output. Instead, the manufacturer-supplied data was used. For the horizontal food case, the average power of the air-cooled unit in the lab was found to be 663 W. During testing of the same unit with a water-cooled condenser, the water loop temperature that corresponded to the same average power consumption was 84°F. At this temperature, the heat output of the unit was determined to be 3,623 Btu/hr. Although differences exist between the in-store operation of the food case and the laboratory operation of the food case, these discrepancies are not expected to be as significant since the unit is located in the cooler and relatively well ventilated front-of-house.

The heat output of the vertical food case involved the greatest degree of uncertainty since the water-cooled coil in the lab was undersized. The unit was tested at three temperature increments. A conservative selection of the lowest heat output value from these tests was chosen to mitigate the risks of over prediction associated with the uncertainties. This instantaneous heat output value was then projected onto a 100% duty cycle, similar to the in-store observations. The heat output of the ice machine was determined as a function of ice production. The heat output per batch was experimentally determined to be equal to manufacturer-supplied specifications, so the final model used for the ice machine is based on manufacturer specifications. The final heat output values used in the predictive model are shown below:

Table 14: Final Heat Output Values Used to Model AC Savings

Refrigerator	Heat Output, Btu/hr
1-Door Freezer	1,194
1-Door Refrigerator	682
2-Door Refrigerator	1,706
Horizontal Food Case	3,623
Vertical Food Case	7,772
Ice Machine	6,950

The values in Table 14 are used in the model as constant year round. Although heat output of the refrigerators is a function of water loop temperature, the water loop system design includes a control set up to maintain a water loop temperature within a prescribed range. During laboratory testing of all units, the purchased heat rejection system can maintain a water loop temperature as low as 95°F. This temperature was therefore selected as the operating point for the system.

With the heat output values of each refrigerator known, the final model was able to predict savings for installing the water loop heat recovery system. Just as with the initial model, the final model includes three sources of savings. These include air conditioning savings associated with removing the refrigerator heat from the building, water preheat savings for the coffee brewers and hot water heater, and refrigeration efficiency savings associated with changing from air-cooled units to water-cooled units. In addition, the model also accounts for the additional costs of operating the circulation pump and the heat rejection fan.

Final Decision Tool—Air Conditioning Savings

Air conditioning savings are estimated using the total heat output of all air-cooled refrigerators (Table 14), the annual operating hours of the air conditioning units and the efficiency of the air conditioning units, as described by the EER value. The EER (energy efficiency ratio) of an AC unit is a ratio of the cooling capacity per power input, with higher EER values equating to more efficient units. An EER value of 12 means that an AC unit can provide 1 ton of cooling capacity for 1 kilowatt of input power. Therefore, dividing 12 by a unit's EER value provides the kW usage per ton of cooling. Equation 1 was used to estimate the savings associated with the reduced AC load:

Equation 1

$$(\boldsymbol{REF_1} + \boldsymbol{REF_2} + \boldsymbol{REF_3} + \boldsymbol{REF_4} + \boldsymbol{REF_5} + \boldsymbol{REF_6}) \times \boldsymbol{C_1} \times \frac{\boldsymbol{12}}{\boldsymbol{EER}} \times \boldsymbol{OH}$$

Where,

REF = Refrigerator heat output (1-6 used to represent six refrigerators from Table 24)

C_1 = Conversion factor, 1 Ton/12,000 Btu/hr

EER = Energy efficiency ratio, 12.7

OH = Annual operating hours, 8,760 (assumed)

Therefore, as a sample calculation,

$$(1{,}194 + 682 + 1{,}706 + 3{,}623 + 7{,}772 + 6{,}950) \times \frac{1}{12{,}000} \times \frac{12}{12.7} \times 8{,}760 = 15{,}124\ kWh$$

For a given utility rate of \$0.22/kWh, the resulting annual savings are \$3,330.

Final Decision Tool-Water Preheat Savings

For a 1-month period from October to November, Starbucks logged several water flow rates in store # 14944, including the water delivered to the hot water heater and the water filter. These figures are shown below:

Figure 15: Starbucks Store #14944 Filtered Water Use

Figure 16: Starbucks Store #14944 Domestic Hot Water Use

The final averaged flow rates taken from the graphs were 6 gallons per hour for the domestic hot water heater and 4.8 gallons per hour for the water filter. It was assumed that 50% of filtered water use was directed towards coffee brewing. In estimating the savings associated with using the hot water in the water loop design to provide preheating for each of these applications, 24/7 operation of both the filtered water and the domestic hot water was assumed, since the average flow rate for each was taken over a month of operation and included 24 hours of each day.

The water loop heat recovery system design includes two water-water heat exchangers, with the hot water loop being the hot water side of each heat exchanger, and the city water flow to the hot water heater and the coffee brewers being the heated water side.

During initial modeling, the heat exchanger effectiveness was assumed to be 1. This assumption was modified by consulting manufacturer data for the two Bell & Gosset brazed plate heat exchangers, which provided the following operating characteristics: hot water side supply temperature of 180°F, hot water side return temperature of 132°F, hot water side flow rate of 5.2 gallons per minute, cold water side supply temperature of 50°F, cold water side return temperature of 140°F, cold water side flow rate of 2.8 gallons per minute, and heat exchange of 125,000 Btu/hr. This data was applied to the log mean temperature difference equation for counterflow heat exchangers to determine the overall heat transfer coefficient, UA, as follows:

Equation 2

$$LMTD = \frac{\Delta T_o - \Delta T_i}{\ln \frac{\Delta T_o}{\Delta T_i}}$$

Equation 3

$$UA = \frac{\dot{q}}{LMTD}$$

Where,

LMTD = log mean temperature difference

ΔT_o = outlet primary fluid temperature minus inlet secondary fluid temperature, 132°F – 50°F

ΔT_i = inlet primary fluid temperature minus outlet secondary fluid temperature, 180°F – 140°F

$\dot{q}$ = heat transfer rate, 125,000 Btu/hr

Therefore,

$$LMTD = \frac{(132°F - 50°F) - (180°F - 140°F)}{\ln \frac{(132°F - 50°F)}{(180°F - 140°F)}} = 58.5°F$$

$$UA = \frac{125{,}000\ Btu/hr}{58.5°F} = 2{,}137 \frac{Btu}{hr - °F}$$

Next, flow rates for filtered water (cold water flow) and the domestic hot water heater are shown over the course of a single day. For the filtered water, the flow rate never exceeds 1 gpm, but for the domestic hot water, the flow regularly exceeds 1 gpm, reaching as high as 5 gpm.

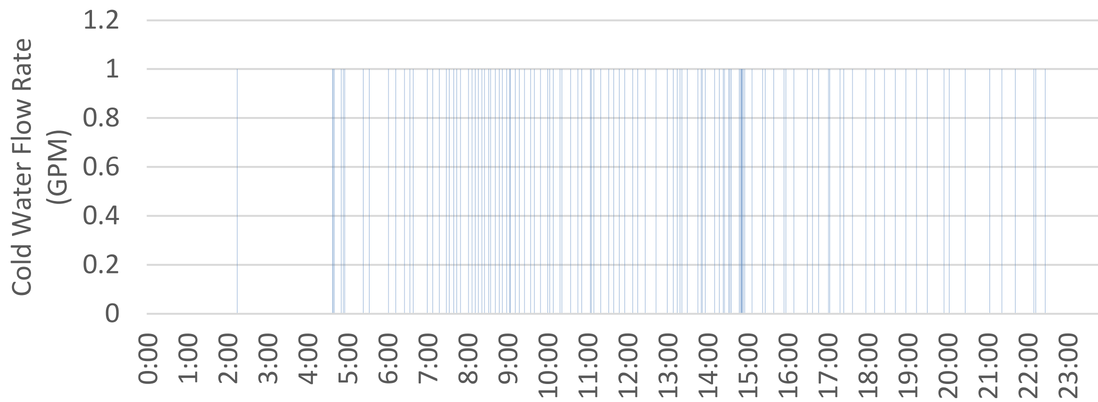


Figure 17: Daily Filtered Water Usage

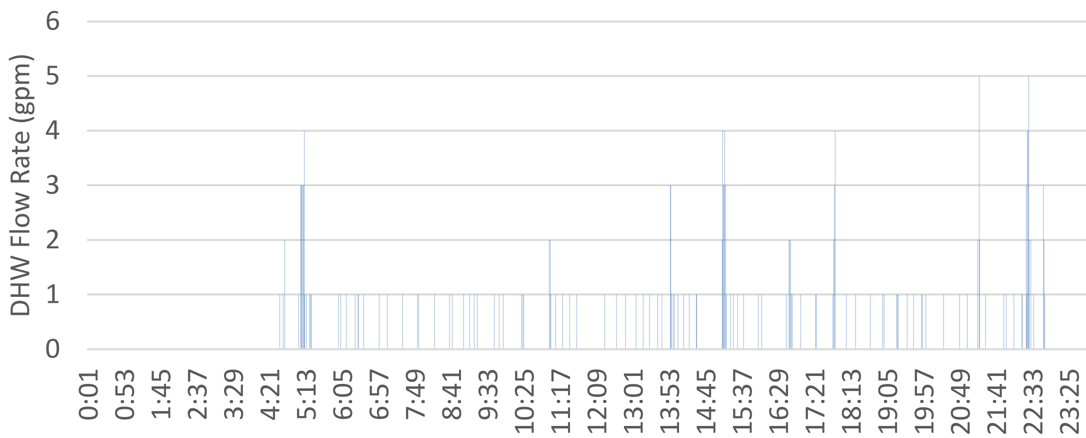


Figure 18: Daily Domestic Hot Water Use

With the overall heat transfer coefficient, a model for the water-water heat exchangers was developed using the following equations:

Equation 4

$$\dot{q} = \dot{m}_c C_{p_c}(Tc_o - Tc_i) = UA \times \frac{(Th_o - Tc_i) - (Th_i - Tc_o)}{\ln \frac{Th_o - Tc_i}{Th_i - Tc_o}} = \dot{m}_h C_{p_h}(Th_o - Th_i)$$

Subscripts *c* and *h* denote cold water and hot water, respectively. Subscripts *i* and *o* denote inlet and outlet, respectively. The mass flow rate of the hot water is 5.5 gallons per minute, determined as the flow rate delivered by the pump (refer to APPENDIX II: EQUIPMENT SIZING). The specific heat capacity of both the hot and cold water is 1.0 Btu/lbm-°F. The entering temperature of the hot water and the cold water are 100°F (approximate water loop temperature) and 60°F (city water temperature).

The unknowns in the equation are the heat transfer rate, the flow rate of the cold side, and the exiting water temperatures of both the hot side and cold side.

For a given cold side water flow rate, a cold side water temperature value such that all sides of Equation 4 are balanced can be determined by trial and error. Referring to the daily flow rates of both the hot water heater and the filtered water, 5 gallons per minutes is highest value the cold side water of the heat exchanger will see. Using increments of cold water side flow rates from 0 to 5 gallons, an exiting cold side water temperature was computed for each increment to balance the above equation.

Equation 4 represents the actual heat transfer through the heat exchanger. The theoretical maximum heat transfer that could occur through the heat exchanger for each flow rate increment is given as:

Equation 5

$$\boldsymbol{\dot{q} = \varepsilon \dot{m} C_p ETD}$$

Where,

$\dot{q}$	=	heat transfer rate, Btu/hr
$\dot{m}$	=	mass flow rate of cold side water, between 0 – 5 gallons per minute
C_p	=	heat capacity of water, 1 Btu/lbm-°F
ETD	=	entering temperature difference of hot water (100°F) and heated water (60°F)
$\boldsymbol{\varepsilon}$	**=**	heat exchanger effectiveness (1.0 for theoretical maximum)

The heat exchanger effectiveness is then given as:

Equation 6

$$\boldsymbol{\varepsilon} = \frac{\dot{q}_{actual}}{\dot{q}_{theoretical}}$$

Applying each of the above equations to each temperature increment of cold water flow, the following performance table for the water-water heat exchangers, as installed in the water loop heat recovery system, was developed:

Table 15: Outlet Conditions of Heated Water for Brazetek Heat Exchanger, Water Loop at 100°F, City Water at 60°F

Cold Water Flow Rate, gpm	Cold Water Exiting Temperature, °F	Actual Heat Transfer, Btu/hr	Theoretical Maximum Heat Transfer, Btu/hr	Heat Exchanger Effectiveness
1	99	19,518	20,016	0.975
2	92.8	32,780	40,032	0.82
3	86.7	39.998	60,048	0.6675
4	82.1	44,381	80,064	0.5525
5	78.8	47,107	100,080	0.47

With Table 15, for a known entering water flow rate into the heat exchanger, the heat exchanger effectiveness is known. This means the preheating provided by the heat exchanger could be estimated as a function of the entering water flow rate of the *heated* water. Logged flow rate data from Figure 17 and Figure 18 (filtered water use and domestic hot water use) provides the flow rate of water for each end use for every minute during a one-month period. For each minute of flow data, Table 14 was consulted to determine a heat exchanger effectiveness value and the rate of heat transfer.

Since the flow rate of the filtered water never exceeds 1 gallon per minute, the heat exchanger performs at very near its theoretical maximum value. For the entire data logged duration for the domestic hot water heater, the summed total of each minute's theoretical maximum heat transfer rate divided by the total data logging hours yielded a value of 2,312 Btu/hr. The summed total of each minute's actual heat transfer rate incorporating the heat exchanger effectiveness for each flow rate value, and again divided by the total data logging duration, yielded a value of 2,053 Btu/hr. Applying Equation 6 results in a representative average heat exchanger effectiveness value of 0.87.

For the final model of both water-water heat exchangers, the heat exchanger effectiveness equation (Equation 5) is used to estimate the heat transfer rate. The values for heat exchanger effectiveness, as previously outlined, are 1.0 for the filtered water and 0.87 for the domestic hot water heater. The flow rates are 6 gallons per hour for the domestic hot water heater (50 pounds per hour) and 2.4 gallons per hour for the filtered water (20 pounds per hour). The heat transfer rates for each are calculated as follows, and Equation 7 is then used to calculate the annual energy savings:

$$\dot{q}_{coffee\ brewers} = 1.0 \times 20 \times 1 \times (100 - 60) = 800\ Btu/hr$$

$$\dot{q}_{water\ heater} = 0.87 \times 50 \times 1 \times (100 - 60) = 1{,}740\ Btu/hr$$

Equation 7

$$\frac{\dot{\mathbf{q}} \times \mathbf{OH} \times \mathbf{C_2}}{\boldsymbol{\eta}}$$

Where,

OH = Annual operating hours, 8,760

C2 = Conversion factor, 0.000293 Btu/hr per kW

η = End use efficiency, 0.92 for hot water heater, 0.80 for coffee brewers

Therefore,

$$\frac{800 \times 8{,}760 \times 0.000293}{0.80} = 2{,}567\ kWh$$

$$\frac{1{,}740 \times 8{,}760 \times 0.000293}{0.92} = 4{,}854\ kWh$$

The combined annual savings for water preheating using the equations and assumptions above were estimated to be 7,421 kWh, or $1,633 for an electricity rate of $0.22/kWh.

Final Decision Tool-Refrigeration Efficiency Savings

Water provides a substantially higher heat transfer coefficient than air. This means that for similar temperature differentials, and similar energy inputs into the fluid movement devices, greater heat transfer can be expected from water-cooled units versus air-cooled units. However, applying this principle to the water loop heat recovery system is complicated by the fact that the water loop temperature is held at 95°F, versus the air-cooled units operating somewhere around 85°F (according to conversations with Stephen Gibson, back-of-house temperatures can be expected to be above 80°F, and even hotter in the tight space between the wall and the refrigerator's condenser coils).

For each of the three smaller refrigerators, as well as the horizontal food case, a water loop temperature was determined which provided an average power consumption equal to the air-cooled baseline. Then, this temperature was compared to the air-cooled, ambient temperature to get a temperature differential between air and water-cooled units for similar performance characteristics. These results are shown in Table 16:

Table 16: Determination of Fluid Temperature Differential

Unit	Air-Cooled (70°F) Baseline Power Consumption, W	Corresponding Water Loop Temperature to Achieve Similar Water-Cooled Power, °F	Water-Air Temperature Differential to Achieve Similar Operating Parameters, °F
1-Door Freezer	195	80	10
1-Door Refrigerator	105	116	46
2-Door Refrigerator	185	83	13
Horizontal Food Case	641	82	12

Next, using the predictive equations for refrigerator's power consumption, power consumption was determined for a set of water loop conditions, beginning with a water-air temperature differential of 0°F and up to 15°F. The power consumption at each temperature increment was then compared to the air-cooled baseline to acquire a percent change. These results are shown in Table 17. Note that negative values reflect an increase in power consumption.

Table 17: Water Loop Performance Improvement vs. Air-Cooled Units

Water Loop Temperature Above Air-Cooled Baseline, °F	Percent Reduction in Refrigerator Power Consumption			
	1-Door Freezer	1-Door Refrigerator	2-Door Refrigerator	Horizontal Food Case
0	12.10%	32.88%	22.63%	7.24%
1	10.89%	32.17%	20.89%	6.63%
2	9.69%	31.46%	19.14%	6.02%
3	8.49%	30.75%	17.40%	5.41%
4	7.28%	30.04%	15.65%	4.80%
5	6.08%	29.33%	13.91%	4.20%
6	4.87%	28.62%	12.16%	3.59%
7	3.67%	27.91%	10.41%	2.98%
8	2.46%	27.21%	8.67%	2.37%
9	1.26%	26.50%	6.92%	1.76%
10	0.05%	25.79%	5.18%	1.15%
11	-1.15%	25.08%	3.43%	0.55%
12	-2.36%	24.37%	1.69%	-0.06%
13	-3.56%	23.66%	-0.06%	-0.67%
14	-4.77%	22.95%	-1.81%	-1.28%
15	-5.97%	22.24%	-3.55%	-1.89%

Table 17 provides means for predicting the power consumption of the refrigerators after installing the water loop heat recovery system. Water loop temperature is set to 95°F. For each of the smaller refrigerators, the in-store air-cooled temperature is around 85°F, creating a fluid temperature differential of 10°F. For the horizontal food case operated in the front-of-house, the in-store air-cooled temperature is more likely to be around 75°F, creating a fluid temperature differential of 20°F (a value not included in Table 17, but corresponding to an *increase* in power consumption of 4.93%).

Although given the vertical food case's operating conditions (located in the hotter back-of-house) it is expected that the unit will operate more efficiently with the water-cooled system, due to uncertainties surrounding its laboratory testing results (due to the improperly sized condenser coil), for the purpose of developing a final model, the unit is assumed to experience no change in power draw.

For the ice machine, the energy savings associated with using the water-cooled condensers were modeled using manufacturer data (which was validated via laboratory testing). The daily energy consumption of the air-cooled unit is 29.2 kWh, compared to the daily energy consumption of the water-cooled unit, which is 24.1 kWh. Duty cycle calculations are considered in determining the savings.

For the three smaller refrigerators and the horizontal food case, the respective percent-change from Table 17 was multiplied by the in-store air-cooled baseline value (for the horizontal food case the laboratory air-cooled baseline was used), then multiplied by the annual operating hours to obtain the energy savings/costs. The final values for changes in refrigerator energy usage are shown in Table 18:

Table 18: Sample Refrigerator Efficiency Improvement Savings for Installing Water Loop Heat Recovery System (Water Loop Temperature 95°F, In-store Air Temperature 85°F)

Unit	Annual Energy Savings, kWh
1-Door Freezer	0
1-Door Refrigerator	285
2-Door Refrigerator	209
Horizontal Food Case	-277
Vertical Food Case	0
Ice Machine	1,845

Final Decision Tool-Secondary Loop Additional Operating Costs

A secondary loop requires the addition of two pieces of energy consuming equipment, including the circulation pump and the heat rejection fans. For a detailed description of the sizing process and modeling considerations for the circulation pump, refer to APPENDIX II: INITIAL MODELING

Circulation Pump Energy Usage

The addition of the secondary loop requires 24/7 operation of a circulation pump. For a power consumption of 400 Watts, and annual operating hours of 8,760, this equates to annual energy usage of 3,504 kWh.

Fan Energy Usage

The energy usage of fan is determined as the product of the power consumption and the operating hours. Power consumption was measured to be 160 Watts. Operating hours required a more detailed analysis.

The National Renewable Energy Laboratory created the National Solar Radiation Data Base, which contains hourly temperature data for 1,019 locations across the United States. Hourly data for all 1,019 locations was taken for the year 2010 and aggregated into a workable spreadsheet to be used in the final decision tool. The first step in determining the fan runtime was to develop an aggregated count of the number of annual hours at given outdoor temperature increments for a selected region. For example, in Limon, CO, there were 354 hours in which the average temperature was between 57.5°F and 60°F, 501 hours in which the temperature was between 60°F and 62.5°F, and so on, until all 8,760 hours of the year are accounted for.

In order to estimate the heat transfer for each temperature increment, an iterative process was used. First, the general convective heat transfer equation is used to determine a first iterative value for heat transfer, as outlined below:

Equation 8

$$\dot{q} = hA\Delta T$$

Where,

$\dot{q}$ = heat transfer rate, Btu/hr

hA = heat transfer coefficient of outside condenser, (56.75 Btu/hr-ft^2°F, see APPENDIX II)

ΔT = difference between ambient air temperature and water loop temperature

After the first iterative value for the heat transfer rate was determined, it was then applied to the general heat transfer equation for each individual fluid (air and water) to determine the exiting temperature of the fluid. The flow rate of air used in the model is 3,000 cfm, as determined by the manufacturer. Then, the inlet and outlet temperatures of each fluid were applied to the log mean temperature difference equation to determine the final value for heat output. A sample of this process is outlined in Table 19. Note that the heat transfer coefficient used in the calculations is 862 Btu/hr-°F, the entering water temperature is 95°F, and the ambient air temperature is 70°F, and the water loop flow rate is 5.5 gallons per minute.

Table 19: Sample of Decision Tool Heat Rejection Calculations

Temperature Range, °F		Number of Annual Hours within Range	Difference Between Water Loop Temperature and Ambient Temperature	Fan Off Heat Rejection (1st Iteration)	Fan On Heat Rejection (1st Iteration)	Fan Off Heat Rejection (2nd Iteration)	Fan On Heat Rejection (2nd Iteration)
From	To		°F	Btu/hr	Btu/hr	Btu/hr	Btu/hr
30	32.5	145	63.75	1,493	54,938	1,488	48,213
32.5	35	88	61.25	1,434	52,784	1,430	46,302
35	37.5	194	58.75	1,376	50,630	1,371	44,393
37.5	40	274	56.25	1,317	48,475	1,313	42,486
40	42.5	208	53.75	1,259	46,321	1,254	40,580
42.5	45	300	51.25	1,200	44,166	1,196	38,675
45	47.5	239	48.75	1,142	42,012	1,138	36,772
47.5	50	253	46.25	1,083	39,857	1,079	34,871
50	52.5	349	43.75	1,025	37,703	1,021	32,971
52.5	55	254	41.25	966	35,548	963	31,073

This computed value for the system heat rejection potential is then compared to the actual heat rejection of the refrigeration units. There are two possible outcomes. If the heat rejection potential provided by the outside condenser is greater than the heat rejected by the refrigerators, the fan will cycle with a duty cycle approximately equal to the refrigerator heat output divided by the outside condenser heat rejection potential. If the heat output of the refrigerators is greater than the heat rejection potential of the system, the fan will operate continuously, and a second fan/condenser unit will activate to reject the remaining heat. If the second fan is unable to reject all of the remaining heat, the system is in danger of overheating and cold city water is required to cool the system.

Decision Tool Outcomes

The primary decision criteria used in determining design implementation, as stated by Starbucks Coffee Company, is a payback period of less than two years. However, according to model/decision tool, the circumstances requiring a payback period of less than two years are unlikely. For example, a store operating its air conditioning 20 hours each day annually, with a utility rate of $0.22/kWh, and operating all six refrigeration units (making it a larger store), the payback period for installing the water loop heat recovery system is 2.3 years.

A diagram of the inputs and output for the decision tool is shown in Figure 19 below, and the input/output page from the decision tool is shown in Figure 20:

Inputs		Outputs
Location		Air Conditioning Savings
Number and Type of Refrigerators		Additional Heating Requirements
AC Runtime, Efficiency		Coffee Water Preheat Savings
Water Loop Temperature	→	Domestic Hot Water Heater Preheat Savings
Average Daily Hours of Operation		Refrigeration Efficiency Savings
Utility Rates		Circulation Pump Energy Usage
		Cooling Fans Energy Usage
		Cold Water Required
		Payback Period

Figure 19: Starbucks Decision Tool Inputs/Outputs

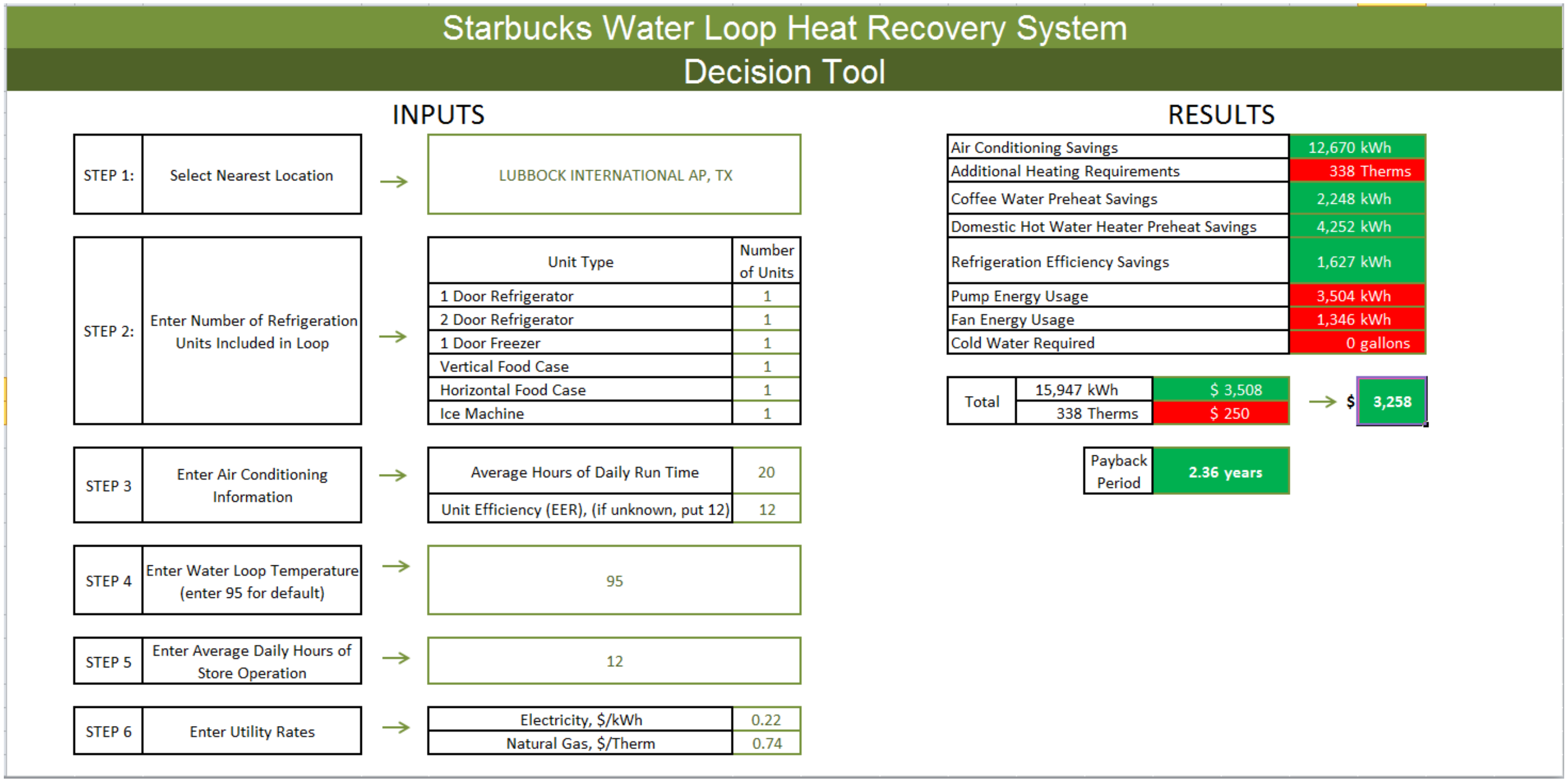


Figure 20: Starbucks Water Loop Heat Recovery System Decision Tool

The decision tool was used to predict the payback period of installing the water loop heat recovery system in 18 cities across the United States. For each location the number of refrigerators was fixed at six, including one ice machine and two food cases. The utility rate was determined by consulting both the Bureau of Labor Statistics and the Energy Information Administration. In order to determine the run time of the air conditioning units at each location, several data sources were consulted. The first data source consulted was logged data obtained by Starbucks for the run time of AC units in 9,311 stores across the United States over a one-year time period ending on 4/28/2016. The second data source consulted was the EIA's CBECS data, which contains commercial building electricity usage data by region. Both data sources were analyzed and integrated to develop an algorithm for determining the run time of the AC units in each of the 19 cities considered. A detailed description of this algorithm is included in APPENDIX V: AC RUNTIME ALGORITHM.

The run time of AC units in Starbucks stores varies substantially even within a given city. For each of the 18 locations considered, four scenarios were simulated, including a high AC run time case to represent stores at the location operating with exceptionally high AC usage, and an average AC run time case to represent a more average figure for stores at the given location. In addition, for both the high and average AC run time cases, a fixed EER scenario and a seasonally variable EER scenario were simulated. Therefore, the four test scenarios for each location are outlined below:

Scenario 1: High AC run time, fixed EER

Scenario 2: High AC run time, seasonally variable EER

Scenario 3: Average AC run time, fixed EER

Scenario 4: Average AC run time, seasonally variable EER

The results of each simulation are shown below:

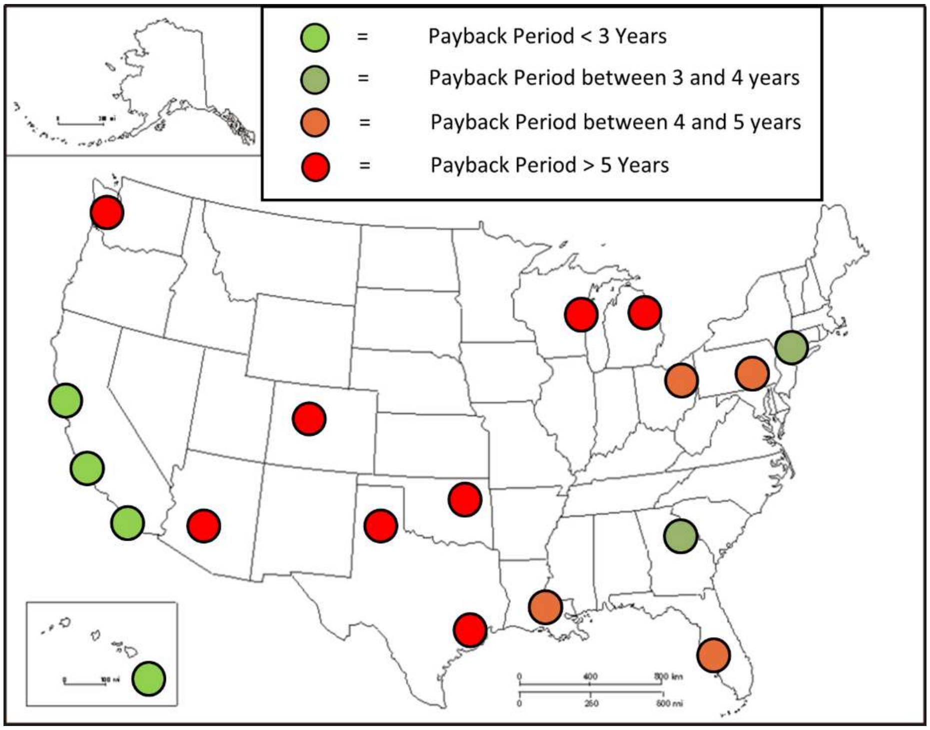


Figure 21: Payback Periods for Scenario 1 (High AC Runtime, Fixed EER)

Table 20: Payback Periods for Scenario 1 (High AC Runtime, Fixed EER)

City	State	AC Runtime	Electricity Rate, $/kWh	Annual Energy Savings, kWh	Payback Period
Seattle	WA	18.8	$0.102	17,747	**7.55**
San Francisco	CA	21.0	$0.233	19,187	**2.53**
Los Angeles	CA	21.0	$0.206	19,155	**2.88**
San Diego	CA	21.0	$0.220	19,203	**2.68**
Phoenix	AZ	21.8	$0.100	19,320	**6.07**
Amarillo	TX	22.5	$0.070	19,805	**8.43**
Houston	TX	22.5	$0.107	19,911	**5.32**
Tulsa	OK	22.3	$0.070	19,934	**8.48**
Denver	CO	18.8	$0.110	17,704	**6.89**
Detroit	MI	17.4	$0.147	16,659	**5.48**
Cleveland	OH	19.3	$0.151	18,084	**4.51**
Philadelphia	PA	19.0	$0.160	17,930	**4.30**
New York City	NY	16.8	$0.207	16,276	**3.79**
Atlanta	GA	21.8	$0.150	19,578	**3.88**
Orlando	FL	23.7	$0.110	20,875	**4.72**
New Orleans	LA	19.4	$0.086	17,684	**9.12**
Green Bay	WI	18.8	$0.109	17,738	**6.95**
Honolulu	HI	23.7	$0.270	20,639	**1.93**

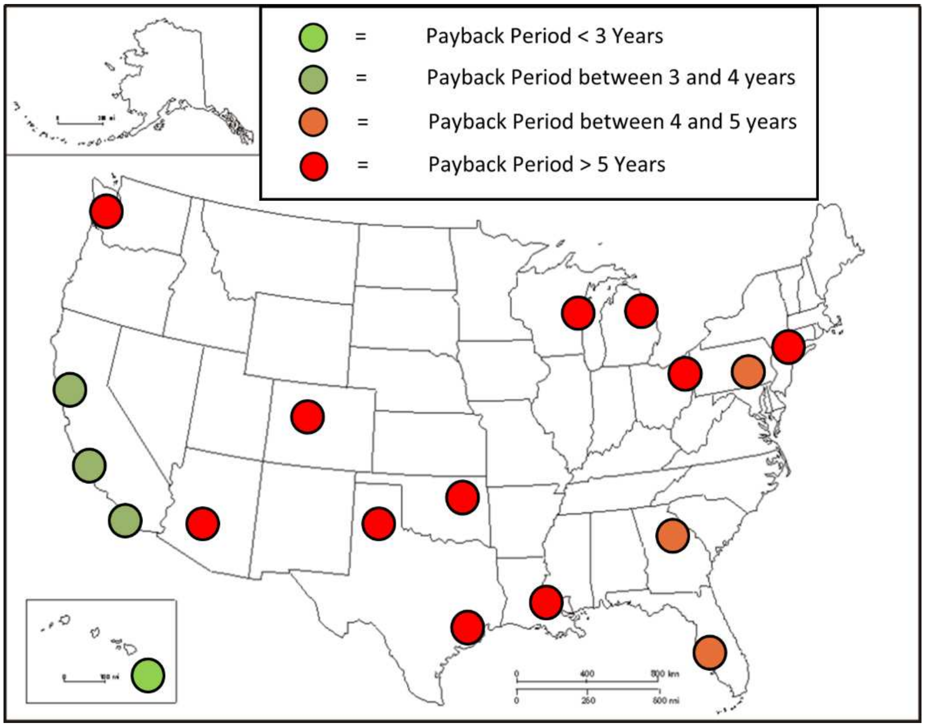


Figure 22: Payback Periods for Scenario 2 (High AC Runtime, Seasonally Variable EER)

Table 21: Payback Periods for Scenario 2 (High AC Runtime, Seasonally Variable EER)

City	State	AC Runtime	Electricity Rate, $/kWh	Annual Energy Savings, kWh	Payback Period
Seattle	WA	18.8	$0.102	13,099	**11.32**
San Francisco	CA	21.0	$0.233	15,836	**3.10**
Los Angeles	CA	21.0	$0.206	16,091	**3.47**
San Diego	CA	21.0	$0.220	15,799	**3.30**
Phoenix	AZ	21.8	$0.100	20,335	**5.74**
Amarillo	TX	22.5	$0.070	20,434	**8.15**
Houston	TX	22.5	$0.107	19,445	**5.46**
Tulsa	OK	22.3	$0.070	18,045	**9.47**
Denver	CO	18.8	$0.110	13,148	**10.15**
Detroit	MI	17.4	$0.147	12,590	**7.89**
Cleveland	OH	19.3	$0.151	13,675	**6.25**
Philadelphia	PA	19.0	$0.160	13,325	**6.09**
New York City	NY	16.8	$0.207	11,901	**5.57**
Atlanta	GA	21.8	$0.150	17,519	**4.36**
Orlando	FL	23.7	$0.110	20,287	**4.85**
New Orleans	LA	19.4	$0.086	16,550	**9.94**
Green Bay	WI	18.8	$0.109	12,876	**10.58**
Honolulu	HI	23.7	$0.270	21,998	**1.81**

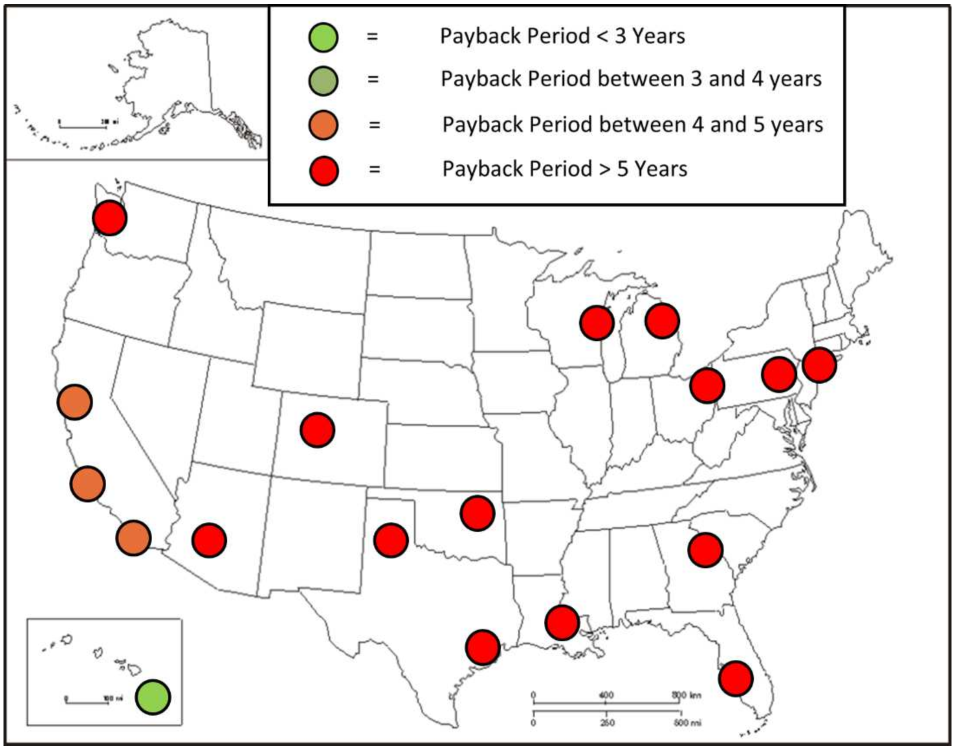


Figure 23: Payback Periods for Scenario 3 (Average AC Runtime, Fixed EER)

Table 22: Payback Periods for Scenario 3 (Average AC Runtime, Fixed EER)

City	State	AC Runtime	Electricity Rate, $/kWh	Annual Energy Savings, kWh	Payback Period
Seattle	WA	18.8	$0.102	12,492	**> 20 years**
San Francisco	CA	21.0	$0.233	13,932	**4.29**
Los Angeles	CA	21.0	$0.206	13,899	**5.06**
San Diego	CA	21.0	$0.220	13,948	**4.61**
Phoenix	AZ	21.8	$0.100	14,065	**14.78**
Amarillo	TX	22.5	$0.070	14,550	**> 20 years**
Houston	TX	22.5	$0.107	14,656	**11.45**
Tulsa	OK	22.3	$0.070	14,679	**> 20 years**
Denver	CO	18.8	$0.110	12,449	**> 20 years**
Detroit	MI	17.4	$0.147	11,404	**16.08**
Cleveland	OH	19.3	$0.151	12,829	**10.03**
Philadelphia	PA	19.0	$0.160	12,674	**9.41**
New York City	NY	16.8	$0.207	11,020	**8.74**
Atlanta	GA	21.8	$0.150	14,323	**7.34**
Orlando	FL	23.7	$0.110	15,620	**9.10**
New Orleans	LA	19.4	$0.086	12,429	**> 20 years**
Green Bay	WI	18.8	$0.109	12,483	**23.66**
Honolulu	HI	23.7	$0.270	15,383	**2.97**

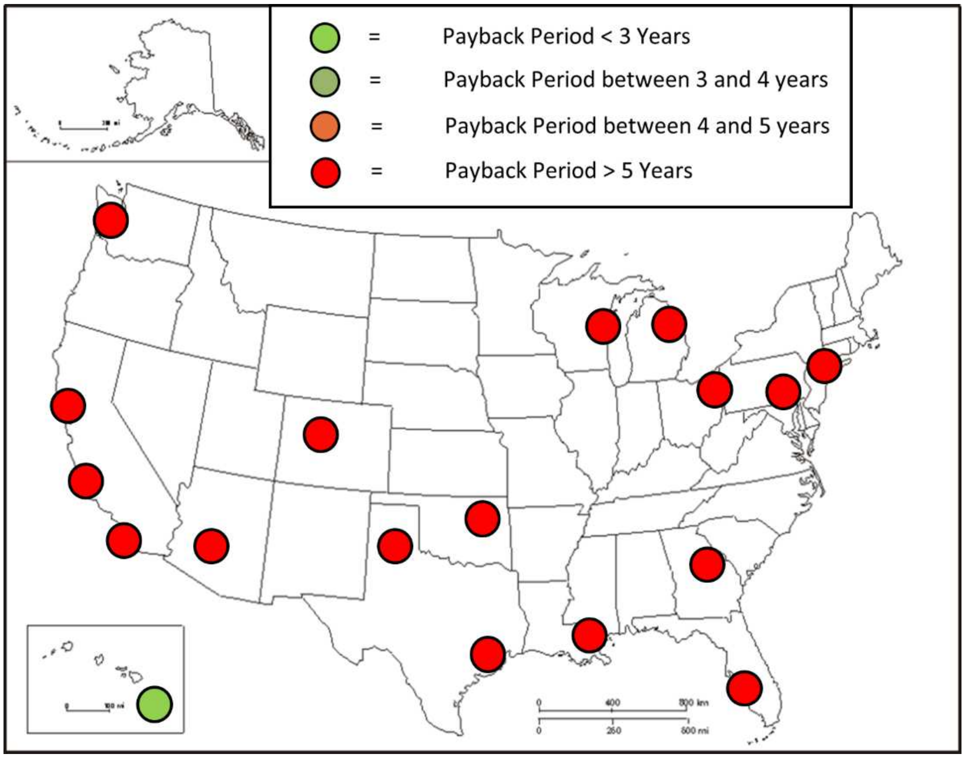


Figure 24: Payback Periods for Scenario 3 (Average AC Runtime Seasonally Variable EER)

Table 23: Payback Periods for Scenario 3 (Average AC Runtime, Seasonally Variable EER)

City	State	AC Runtime	Electricity Rate, $/kWh	Annual Energy Savings, kWh	Payback Period
Seattle	WA	18.8	$0.102	9,577	**> 20 years**
San Francisco	CA	21.0	$0.233	11,698	**5.41**
Los Angeles	CA	21.0	$0.206	11,857	**6.31**
San Diego	CA	21.0	$0.220	11,679	**5.88**
Phoenix	AZ	21.8	$0.100	14,754	**13.50**
Amarillo	TX	22.5	$0.070	14,983	**> 20 years**
Houston	TX	22.5	$0.107	14,335	**11.89**
Tulsa	OK	22.3	$0.070	13,384	**> 20 years**
Denver	CO	18.8	$0.110	9,593	**> 20 years**
Detroit	MI	17.4	$0.147	8,973	**> 20 years**
Cleveland	OH	19.3	$0.151	10,019	**16.61**
Philadelphia	PA	19.0	$0.160	9,764	**15.91**
New York City	NY	16.8	$0.207	8,469	**15.32**
Atlanta	GA	21.8	$0.150	12,926	**8.56**
Orlando	FL	23.7	$0.110	15,205	**9.46**
New Orleans	LA	19.4	$0.086	11,705	**> 20 years**
Green Bay	WI	18.8	$0.109	9,435	**> 20 years**
Honolulu	HI	23.7	$0.270	16,342	**2.77**

Of the 9,311 Starbucks stores with one year of AC run time data, only 146 (1.6%) have greater than 20 hours of average daily run time. The high AC run time scenario is therefore very uncommon. However, for both the fixed EER and the seasonally variable EER, the high AC run time scenario does show several locations with a payback period that is around 3 years.

The purpose of including the seasonally variable EER test scenario was to demonstrate the potential impacts of climate. Comparing test scenario 1 and 2 (both high AC run time, scenario 1 with fixed EER, scenario 2 with seasonable variable EER) isolates climate as the variable of interest. The range of energy savings when considering the fixed EER scenario is from 16,000 kWh to 20,000 kWh, whereas for the seasonably variable EER scenario the range is from 11,000 kWh to 22,000 kWh. In addition, the inclusion of EER as a function of location had the effect of generally increasing the payback period. This is because the rated EER used in the fixed EER scenario, which is provided by manufacturers, is often a very conservative figure for most climate scenarios.

In summary, the decision tool predicts a payback period close to the desired goal of 2 years when the following criteria are met:

- Average daily AC run time of approximately 20 hours or greater
- Utility rates at or above $0.20/kWh
- Water loop contains at least one vertical food case (or equivalent unit) and one ice machine (or equivalent unit)

FUTURE WORK

As previously mentioned, Starbucks Coffee Company has an objective of achieving a reduction of 25% of the energy consumption in each of their stores. (Starbucks Coffee Company, 2014) The purpose of testing the water loop heat recovery system is to determine whether it can be a viable technology which can contribute towards these goals. In this context, the ultimate goal of the water loop heat recovery system is a full scale roll out into every store in which the design can be expected to demonstrate an economically appealing payback period. The project's overall progression towards this end goal of full-scale roll out can be divided into several phases, as outlined in Figure 25:

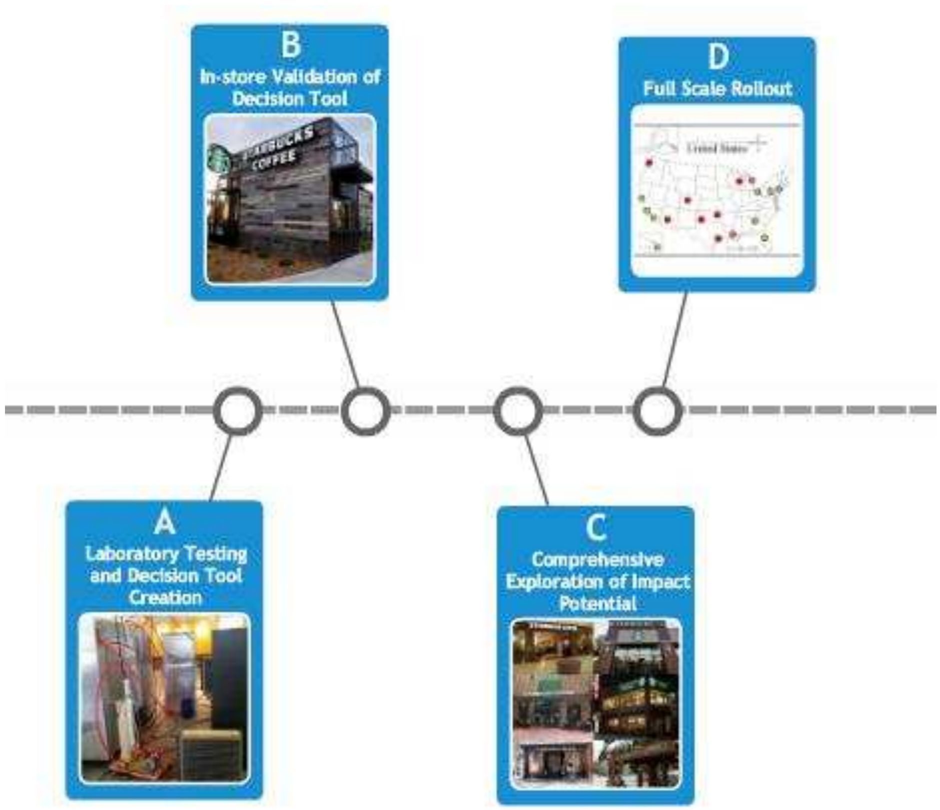


Figure 25: Overall Project Stages

Phase A represents the work detailed in this paper. Phase B represents the pilot testing of the system within a Starbucks store. Phase C represents a more detailed exploration of the design, including multiple in-store testing to verify the system's performance for different locations. Phase D represents the adoption of the technology as a proven energy saver and the subsequent large-scale implementation. A detailed discussion of the future work requirements of each phase is provided below.

Phase A

Phase A represents the work outlined in this paper. However, in order to ensure the system is ready for pilot testing in a Starbucks store, several items require consideration. For example, the current decision tool has several shortcomings, outlined in the section entitled 'Model Deficiencies.' In addition, several logistical items should be considered prior to installing the system, as outlined in the section entitled 'Additional Logistical Requirements.' Finally, several improvements to the design are considered in 'Potential Design Improvements.

Model Deficiencies

The most substantial deficiency to the existing model is the lack of data for the performance of a properly sized water-cooled condenser coil for the vertical food case. In addition, testing of the system in the store will provide conclusive numbers for the heat output each unit during store operation as well as power consumption data. Finally, several values used in the model were specific to the Fort Collins store, including:

- Refrigerator models
- Flow rate values for the hot water heater and coffee brewing.
- Rate of ice production.
- City water temperature.
- Hot water heater efficiency.

Additional Logistical Requirements

The list of items required immediate action or consideration before in-store installation occurs is given below:

- Careful care given to the proper sizing of the water-cooled condenser coils. Consideration should be given to the water loop temperature in sizing the coils.
- Ensure the heat rejection fans are rated for outdoor use.
- Design work for the control system in case of pump failure or water overheating is still required.
- Careful consideration given to the selection of the circulation pump, with an emphasis on minimizing energy usage.

Potential Design Improvements

Most individual components performed according to their sizing expectations. The heat rejection system, however, struggled to maintain water loop temperatures below 100°F when all refrigerators were operating. Using an oversized heat rejection system would allow for more flexibility in water loop temperature control. It would also allow for the possibility of using a variable frequency drive to regulate pump speed, and reduce the speed/power consumption during evenings and slow periods.

Another improvement to the design would be to allow for the option within the system to reject the heat inside the building. If the building is in heating mode, rejecting the heat outside is counterproductive. This would include a separate control system tied to the building's heating or air conditioning system, and an additional, optional, heat rejection system inside the building connected in parallel with the rest of the loop.

Phase B

Phase B involves the pilot testing of the water loop heat recovery system within a Starbucks store. Ideally, the system could be installed in two stores within the San Diego region, one to represent a high AC runtime store, and one to represent an average AC runtime store. In both cases, the existing decision tool provides an estimate for the system's performance. The purpose of the in-store testing will be primarily model validation, and recalibration of the model according to the results.

According to the existing model, the requirements for a desirable payback period for the water loop heat recovery system are outlined below:

- Average daily AC run time of approximately 20 hours or greater
- Utility rates at or above $0.20/kWh
- Water loop contains at least one vertical food case (or equivalent unit) and one ice machine (or equivalent unit)

In reality, either the existing model is too conservative or too generous. If it is too generous, these requirements will become even more stringent, and the maps in Figure 21, Figure 22, Figure 23, and Figure 24 will shift to include more red dots and less green dots. In this case, Starbucks will likely have to reevaluate the project scope.

If the pilot testing demonstrates the model to be overly conservative, the requirements for a desirable payback period will loosen, allowing the maps in Figure 21, Figure 22, Figure 23, and Figure 24 to contain more green dots. In this scenario, it is likely that Starbucks will want to further validate the decision tool's performance, particularly in stores with a border line payback period.

Phase C

The refined decision tool will be used to generate a new map detailing the results of installing the system in different geographic locations. However, it is unlikely that full-scale rollout of the water loop heat recovery system will begin after pilot testing. Most likely, the next step will involve additional in-store testing in multiple locations to verify the system's performance in different environments. The refined decision tool will likely be used to suggest stores which are border line between an ideal candidate and a non-candidate. Then, several of these locations will be selected for in-store testing.

The aim of Phase C is to verify the economic viability of the system for different factors, such as climate, store size and utility rates. For Phase C, it will be important for Starbucks personnel to consider the dynamic landscape, in terms of each of these factors. For example, although a given store might not make an ideal candidate for the water loop heat recovery system in the present, if the utility rates increase by a given amount, the design may become a viable option for the store. Similarly, climate can fluctuate significantly from year to year, and a given store can grow. These items should be regularly monitored and considered for future development.

Phase D

Phase D represents the final full-scale roll out of the design. The final decision tool, recalibrated in both Phase B and Phase C, will be used to predict which stores would make economic sense to install the water loop heat recovery system. If 10% of all Starbucks stores are good candidates, and the system delivers energy savings of 20% of a store's annual energy usage, this design would represent around a 2% reduction in overall, national energy usage.

SUMMARY AND CONCLUSION

The objective of this study was to investigate the feasibility of an indirect (secondary loop) refrigeration system within food service commercial buildings. Currently indirect systems are used in facilities such as large dairies and butcheries, where large piping systems make them a simple alternative to a conventional direct expansion system. In addition, growing concerns of increased greenhouse gas effects associated with refrigerant leakage have prompted numerous studies of indirect systems, with a majority of those studies focused on supermarkets.

Within food sales commercial buildings, there is a unique incentive for implementing an indirect refrigeration system, which is the reduced air conditioning load of the building. As part of an agreement with Starbucks Coffee Company and Colorado State University, an experimental investigation of a water loop heat recovery system was performed. This system included six Starbucks refrigeration units, selected as the most likely candidates to be part of the loop, cooled using water-cooled condensers connected in series. The loop will provide preheating for the coffee brewers and the domestic hot water heaters, and excess heat is dumped outside the building.

A comprehensive data analysis was performed for all data collected during laboratory testing, with the objective creating final decision tool for Starbucks to determine candidate stores for the installation of the system. The criterion imposed by Starbucks was a two-year payback period. The model predicted the major requirements to meet this criterion to be high energy costs (>$0.22/kWh), a warm to hot climate (AC runtime > 20 hours per day), and a large store (containing multiple large food cases or ice machines).

WORKS CITED

Baxter, V. D. (2003). *Advances in Supermarket Refrigeration Systems.* Tennessee: Oak Ridge National Laboratory.

Better Buildings: U.S. Department of Energy. (n.d.).

Better Buildings: U.S. Department of Energy. (2015, April). Showcase Project: 1751 Howell Mill Road. Atlanta, Georgia, United States.

Better Buildings: U.S. Department of Energy. (2016). Showcase Project: Shari's of Roseburg--A Smaller Slice of the Energy Pie. Roseburg, Oregon, United States.

Better Buildings: U.S. Department of Energy. (2016). Showcase Project: Wendy's #2142 - Maxtown Road. Westerville, Ohio, United States.

California Energy Commission. (2004, March). Investigation of Secondary Loop Supermarket Refrigeration Systems. CA, United States.

City of Vancouver. (2012, September). Business Energy Efficiency: Coffee Shops. Vancouver, British Columbia, Canada.

CRC Press. (1997). Electrical Energy Management in Buildings. In F. Kreith, & R. West, *CRC Handbook of Energy Efficiency.*

D. Clodic, C. L. (1998). *Comparison of Energy Efficiencies of Commercial Refrigeration Direct and Indirect Systems.* Purdue University.

Effsys2 P2. (2010). *Handbook on indirect refrigeration and heat pump systems.* Kullavik, Sweden: Svenska Kyltekniska Foreningen.

Environmental Protection Agency. (1995, October). Compliance Guidance for Industrial Process Refrigeration Leak Repair Under Section 608 of the Clean Air Act.

Finn, S. S. (2012). *Experimental Feasibility Investigation of the Use of Indirect Refrgeration Systems (Secondary Loops) in Climate Controlled Transport Systems.* The Institute of Refrigeration.

Fisher, D., & Karas, C. a. (2012). Commercial Ice Machines: The Potential for Energy Efficiency and Demand Response. *ACEEE*, (9-103)-(9-114).

Fricke, B. (2016). High Efficiency Low Emission Refrigeration System. *2016 Building Technologies Office Peer Review* (p. 19). U.S. Department of Energy.

Gibson, S. (2016, April 20). Corporate Architect. (C. Anderson, Interviewer)

Hill Phoenix Refrigeration Systems. (2011). Second Nature: A Comparative Study of Secondary Glycol and Direct Expansion Refrigeration Systems. Conyers, GA, United States.

Minea, V. (2007). Supermarket Refrigeration with Completely Secondary Loops. *ASHRAE*, 40-56.

National Renewable Energy Laboratory. (2011, June). Saving Energy in Commercial Buildings. Golden, CO, United States.

Pacific Northwest Laboratories. (2009, April). Energy Efficiency Potential in Existing Commercial Buildings: Review of Selected Recent Studies. Richland, WA, United States.

Starbucks Coffee Company. (2014). Global Responsibility Report 2014. United States.

Stevens Engineers. (2009). Application of Climate-Friendly Ice Rink Technologies: Brooklyn Park Community Activity Center. Brooklyn, New York, United States.

U.S. Department of Energy: Energy Efficiency & Renewable Energy. (2016, April). Energy Efficiency in Separate Tenant Spaces - A Feasibility Study.

U.S. Department of Energy: Energy Efficiency and Renewable Building Technologies Program. (2009). *Energy Savings Potential and R&D Opportunities for Commercial Refrigeration.* U.S. Department of Energy.

U.S. Energy Information Administration. (2016, May 17). Commercial Buildings Energy Consumption Survey (CBECS).

U.S. Environmental Protection Agency. (2012). Profile of an Average U.S. Supermarket's Greenhouse Gas Impacts from Refrigeration Leaks Compared to Electriciy Consumption. United States.

Zhang, M. (2006). *Energy Analysis of Various Supermarket Refrigeration Systems.* Purdue University.

APPENDIX I: INITIAL MODELING

The design selected for further investigation and in-store installation was the water loop heat recovery system. In order to construct the preliminary model for this design option, several assumptions were made. These assumptions are listed below:

- Utility rate of $0.22/kWh. Starbucks Coffee Company is primarily interested in the system's performance in the San Diego region, where $0.22/kWh is the typical rate.
- High outdoor temperature of 87°F, corresponding to the San Diego climate.
- 24/7 air conditioning run time. According to early conversations with Starbucks corporate personnel, the rooftop AC units operate continually year round within the regions of interest.
- RTU EER value of 12.7, taken directly from the roof top units on the Fort Collins store.
- A circulating pump size of 1/12 hp. Sizing was determined by preliminary modeling of the required flow rate for the water-cooled system.
- A maximum water loop temperature of 100°F, as per conversations with refrigerator manufacturers.
- City water temperature of 60°F, average city water temperature in Fort Collins.
- Domestic hot water efficiency of 0.92. This value is a standard value, and also the value of the water heater in the Fort Collins Store.
- Coffee warmer efficiency of 0.80, determined as a representative general value for coffee heaters.
- 50% of filtered water use directed to coffee. The exact quantity is unknown. Filtered water is also used for ice production. A value of 50% was selected to minimize uncertainty.

- 12 hours of daily operation. Each Starbucks store is different, and hours can vary by season within a store.

The installation of a water loop heat recovery system is expected to have three sources of savings. The first and primary source of savings is the savings associated with the reduction in the HVAC load that will result from removing the refrigeration heat from the buildings. The second source of savings are found in the reduced heating requirements of the hot water heater and the coffee machines, given the preheating provided by the water loop. The third and final source of savings is derived from the expected improvement in efficiency of the refrigeration units associated with using water-cooled condensers instead of the incumbent air-cooled condensers.

Savings Source 1: Reduced Air Conditioning Load

For the purposes of estimating the first source of savings, the reduction in AC load, the heat output of each refrigerator had to be estimated. Starbucks maintains an equipment load worksheet for store # 14944. However, this data was subject to variability due to equipment changes. The six refrigeration units selected for the water loop heat recovery system are shown below. In addition, the values for each unit's power consumption and heat output were determined from nameplate data and the Starbucks equipment load worksheet:

Table 24: Power Draw and Waste Heat per Refrigeration Unit, Preliminary Model Values

Refrigeration Unit	Manufacturer	Model #	Average Power (W)	Waste Heat (Btu/hr)
1-Door Refrigerator	True Manufacturing	TG1R-1S	176	600
2-Door Refrigerator	True Manufacturing	TG2R-2S	586	2,000
1-Door Freezer	True Manufacturing	T-23F	703	2,400
Vertical Food Case	Structural Concepts	SBB45	1,465	5,000
Horizontal Food Case	Structural Concepts	SB5766.3923A	938	3,200
Ice Machine	Ice-O-Matic	ICE1006HA	1,758	6,000
		Total	**5,626**	**19,200**

The EER (energy efficiency ratio) of an AC unit is a ratio of the cooling capacity per power input, with higher EER values equating to more efficient units. An EER value of 12 means that an AC unit can provide 1 ton of cooling capacity for 1 kilowatt of input power. Therefore, dividing 12 by a unit's EER value provides the kW usage per ton of cooling. Equation 1 below was used to estimate the savings associated with the reduced AC load:

Equation 9

$$(\boldsymbol{REF_1} + \boldsymbol{REF_2} + \boldsymbol{REF_3} + \boldsymbol{REF_4} + \boldsymbol{REF_5} + \boldsymbol{REF_6}) \times \boldsymbol{C_1} \times \frac{\mathbf{12}}{\boldsymbol{EER}} \times \boldsymbol{OH}$$

Where,

REF	=	Refrigerator heat output (1-6 used to represent six refrigerators from Table 24)
C_1	=	Conversion factor, 1 Ton/12,000 Btu/hr
EER	=	Energy efficiency ratio, 12.7
OH	=	Annual operating hours, 8,760

Therefore,

$$(600 + 2{,}000 + 2{,}400 + 5{,}000 + 3{,}200 + 6{,}000) \times \frac{1}{12{,}000} \times \frac{12}{12.7} \times 8{,}760 = 13{,}240\ kWh$$

For the given utility rate of $0.22/kWh, the resulting annual savings are $2,900.

<u>Savings Source 2: Water Preheating</u>

The only difference between the initial modeling of the water-water heat exchangers and the final modeling was the inclusion of the heat exchanger effectiveness term. For preliminary estimates, an assumed value for the water-water heat exchanger effectiveness of 1 was used. Equation 10 was used to estimate the heat transfer in each heat exchanger, and Equation 11 was used to estimate the annual energy savings associated with preheating provided by the hot water loop for both the coffee brewers and the hot water heater. Note that Equation 10, which solves for the heat transfer rate, $\dot{q}$, is a simplified version of the heat exchanger effectiveness equation since it assumes a heat exchanger effectiveness of 1 and uses the significantly lower mass flow rate of the cold water side.

Equation 10

$$\dot{q} = \dot{m} C_p \Delta T$$

Where,

$\dot{m}$	=	mass flow rate of water; for coffee, 2.4 gph or 20 lbm/hr, for hot water heater, 6 gph or 50 lbm/hr
C_p	=	heat capacity of water, 1 Btu/lbm-°F
ΔT	=	entering temperature difference of hot water (100°F) and heated water (60°F)

Equation 11

$$\frac{\dot{q} \times OH \times C_2}{\eta}$$

Where,

OH	=	Annual operating hours, 8,760
C2	=	Conversion factor, 0.000293 Btu/hr per kW
η	=	End use efficiency, 0.92 for hot water heater, 0.80 for coffee brewers

Therefore,

$$\dot{q}_{coffee\ brewers} = 20 \times 1 \times (100 - 60) = 800\ Btu/hr$$

$$\dot{q}_{water\ heater} = 50 \times 1 \times (100 - 60) = 2{,}000\ Btu/hr$$

$$\frac{800 \times 8{,}760 \times 0.000293}{0.80} = 2{,}567\ kWh$$

$$\frac{2{,}000 \times 8{,}760 \times 0.000293}{0.92} = 5{,}580\ kWh$$

The combined annual savings for water preheating using the equations and assumptions above were estimated to be 8,150 kWh, or $1,800.

Savings Source 3: Refrigeration Efficiency

The third source of savings, associated with the expected improved refrigeration efficiency of the water-cooled condensers, involved the least certainty of all sources of savings. According to the CRC handbook of energy efficiency, refrigerator efficiency can increase by up to 2% per °F reduction in heat sink temperature of the condenser. (CRC Press, 1997) However, no measurements were available for condenser temperatures on either air-cooled units or water-cooled units. After some deliberation comparing the functionality of air-cooled condensers operating in a store and water-cooled condensers relying on 100°F water, a value of 5% improvement in efficiency was selected. This would equal energy savings of 2,460 kWh annually, or about $540.

Additional Costs and Final Values

The use of a secondary water loop to provide cooling for the system requires additional sources of energy consumption, including a circulation pump and a heat rejection fan. The final design is intended to be installed in a San Diego store, where the annual high temperature is around 87°F. Assuming the heat rejection system can cool the water to within 5°F of the outside temperature, this would mean that for the hottest day of the year in San Diego, the water loop would fluctuate between 92°F and 100°F. Equation 10 was used to solve for the required mass flow rate of water given this temperature difference and the heat output of the refrigerators.

Inputting the known values into the equation results in a preliminary flow rate requirement estimate of 288 gallons per hour, or about 5 gallons per minute. A 1/12 hp centrifugal circulation pump could deliver this flow. Operating 24/7, the pump would consume electricity equal to 540 kWh annually, costing about $120. The manufacturer Valutech was consulted about the heat rejection system. They suggested that a ¼ hp fan could provide sufficient heat rejection for the system. Operating 24/7, the fan would consume electricity equal to 1,630 kWh annually, costing about $360.

APPENDIX II: EQUIPMENT SIZING

The design of the water loop heat recovery system required sizing multiple components. These components included six water-cooled condenser coils, two water-water heat exchangers, a heat rejection system, and a circulation pump/motor.

Condenser Coils

Doucette Industries Inc. manufactures coaxial condenser coils. These condenser coils are built in incremental sizes, with each increment corresponding to a particular refrigeration compressor size. Table 25 summarizes the sizing of each condenser, and Figure 26 illustrates the fan-cooled condenser removed from the 2-door refrigerator and the water-cooled condenser that replaced it.

Table 25: Water-Cooled Condenser Coil Sizing

Refrigeration Unit	Manufacturer	Model #	Compressor Size (HP)	Selected Water-Cooled Condenser
1-Door Refrigerator	True Manufacturing	TG1R-1S	0.33	CX-H-033
2-Door Refrigerator	True Manufacturing	TG2R-2S	0.5	CX-H-050
1-Door Freezer	True Manufacturing	T-23F	0.5	CX-H-050
Vertical Food Case	Structural Concepts	SBB45	1	CX-H-100
Horizontal Food Case	Structural Concepts	SB5766.3923A	1.5	CX-H-150
Ice Machine	Ice-O-Matic	ICE1006HA	1.5	CX-H-150

A. B.

Figure 26: (A) Original Fan-Cooled Condenser (B) Water-Cooled Condenser

In order to verify the functionality of each water-cooled condenser for each refrigerator, a comparison was made between the power consumption of each unit operating under different water loop conditions versus that same unit operating with an air-cooled condenser. For each unit, power consumption was data logged for a period of time between three to five hours at each given water loop temperature interval, as well as for the air-cooled baseline. For the three smaller refrigerators (1-door freezer, 1-door refrigerator, 2-door refrigerator), power was data logged using Onset's HOBO plug load data loggers. For the three larger units (horizontal food case, vertical food case, ice machine), power was data logged using an ELITEpro XC Dent Instruments Energy Logger. Room temperature was determined by the laboratory thermostat to be to be 70°F, and verified using a HOBO Tidbit v2 temperature data logger. The following figures demonstrate a duty cycle comparison for each unit:

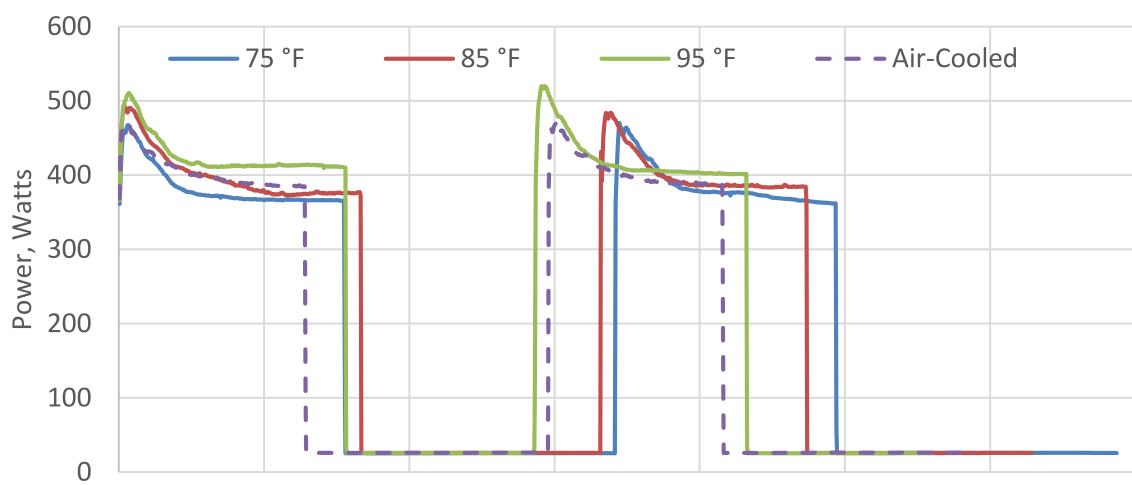


Figure 27: 1-Door Freezer Duty Cycle Comparison

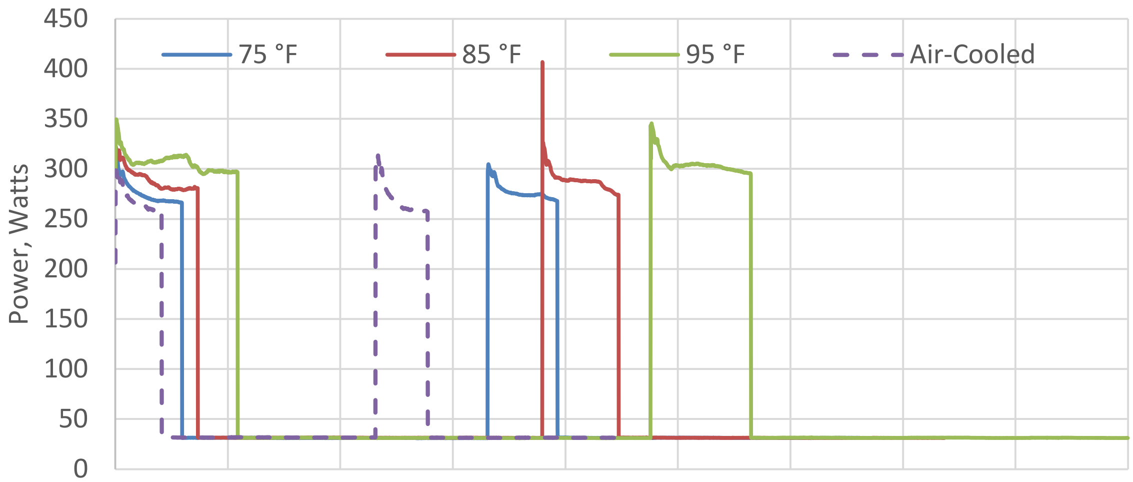


Figure 28: 1-Door Refrigerator Duty Cycle Comparison

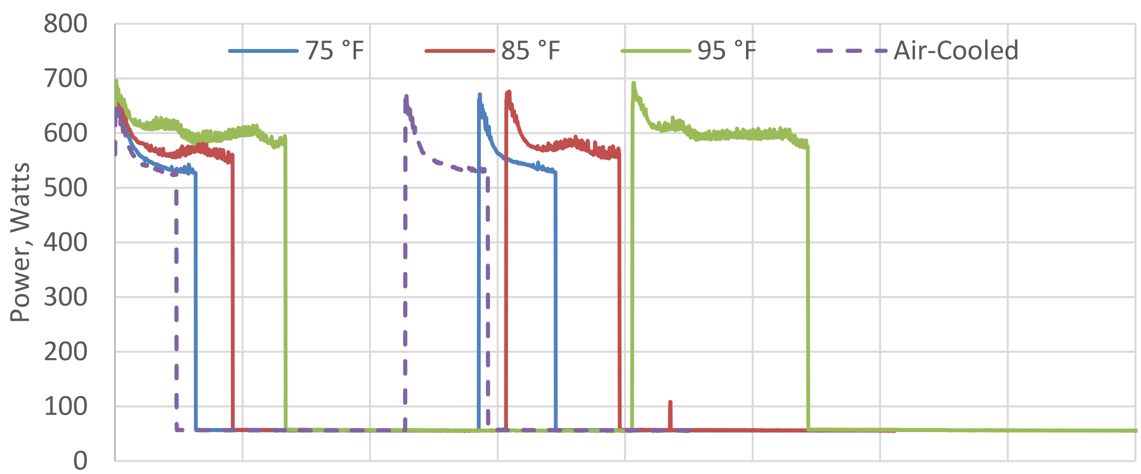


Figure 29: 2-Door Refrigerator Duty Cycle Comparison

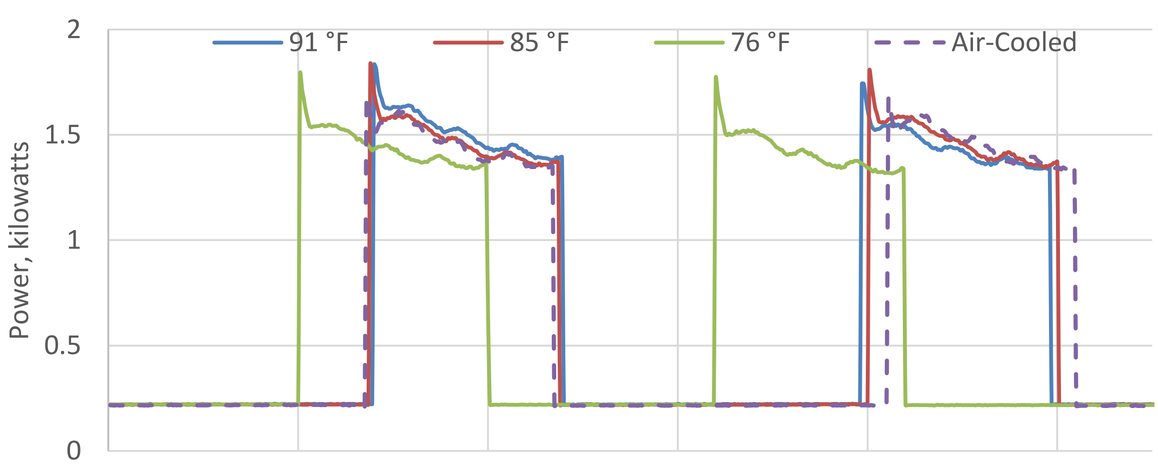


Figure 30: Horizontal Food Case Duty Cycle Comparison

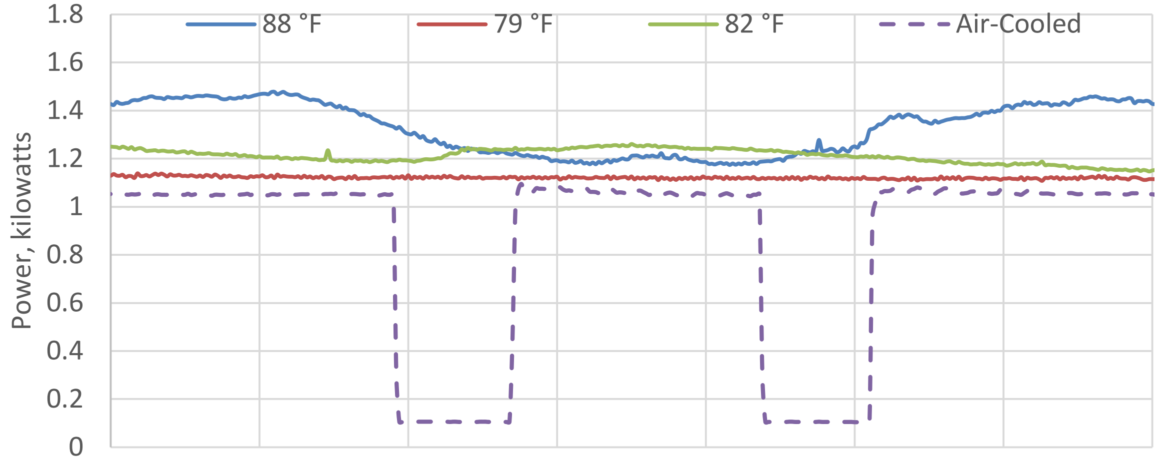


Figure 31: Vertical Food Case Duty Cycle Comparison

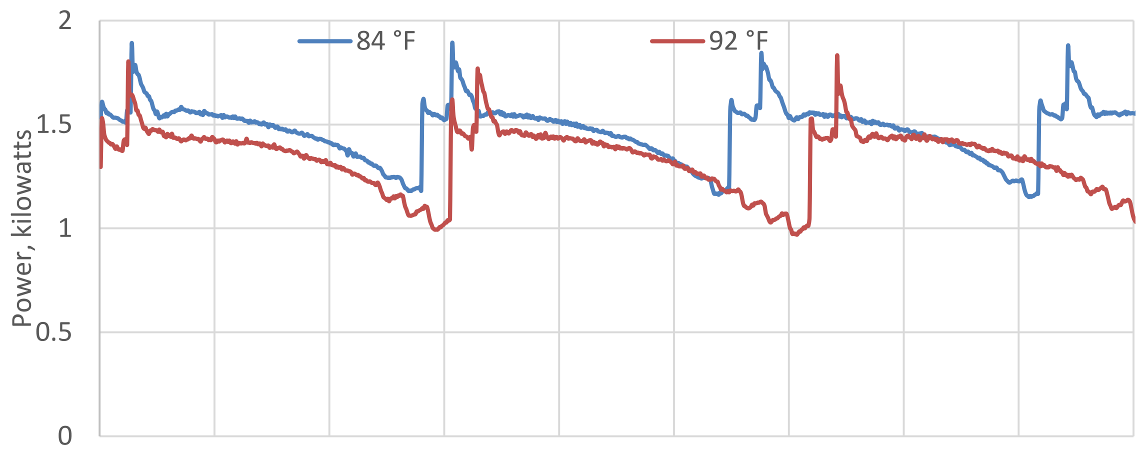


Figure 32: Ice Machine Duty Cycle Comparison

For the 1-door freezer, the original water-cooled condenser was undersized. It was tested with the undersized condenser, then retrofitted with a properly sized condenser and retested. Figure 32 demonstrates the duty cycle of the 1-door freezer at different water loop temperatures for both condenser coil sizes.

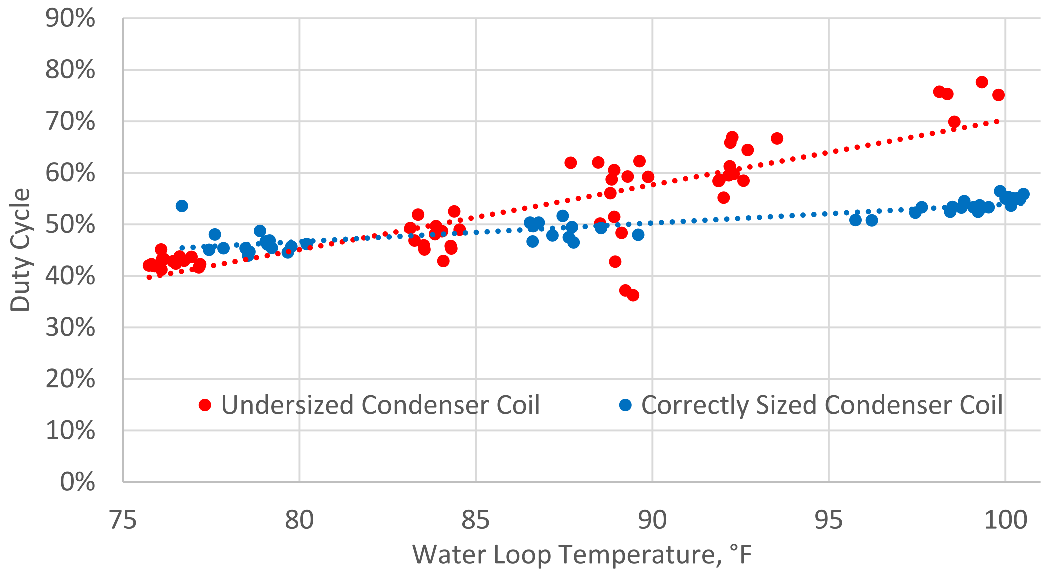
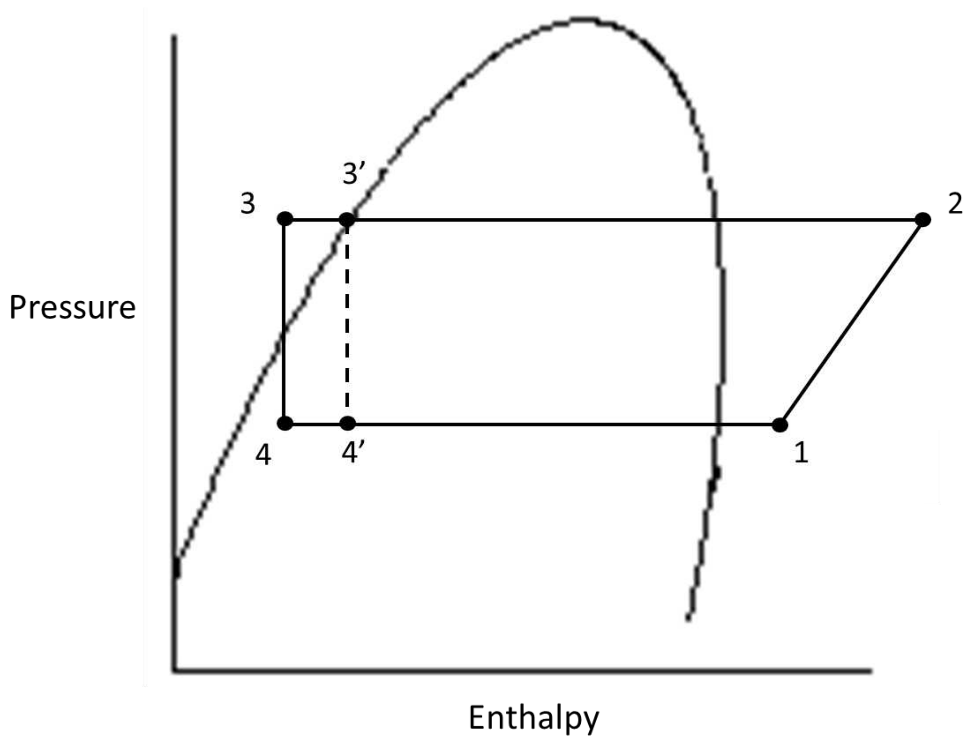


Figure 33: Effect of Improperly Sized Condenser Coil for 1-Door Freezer

The correctly sized condenser coils shows a significantly less steep increase in duty cycle with increasing water loop temperature. The horizontal food case and the ice machine performed as intended, but analysis of data logging for the vertical food case indicated similar performance characteristics to the freezer operating with the undersized condenser coil. At any water loop temperature, the vertical food case did not cycle, instead staying in the loaded state. Data for the same unit operated with an air-cooled condenser demonstrated an expected duty cycle. This strongly suggests the water-cooled condenser coil is undersized. However, due to time constraints, this condenser coil was not able to be resized.

Although the air-cooled vertical food case in the lab exhibited an expected duty cycle, the same model air-cooled food case operating in the store operated with a 100% duty cycle. This observation is most likely related to the heat transfer that occurs in the condenser. The condenser fans for the laboratory air-cooled unit utilize 70°F air. By contrast, the in-store back-of-house is typically over 80°F, and can be significantly warmer still behind a refrigerator. Therefore, the condenser fans for the in-store air-cooled unit are providing cooling with significantly warmer air.

This warmer air limits the condenser's effectiveness. In terms of the refrigeration cycle for the refrigerant, the condenser is likely unable to cool the refrigerant to the same sub-cooled level. This in turn impacts the evaporator, since the saturated refrigerant entering the evaporator will have a higher enthalpy, and therefore less cooling capacity. Therefore, in order to achieve the same cooling effect for a warmer air supply to the condenser, the refrigerator's duty cycle is increased until it can no longer increase, at which point the temperature set point in the refrigerator can no longer be maintained. Figure 34 shows a comparison of a normal refrigeration cycle to a refrigeration cycle operating with a higher air supply temperature to the condenser (denoted using ').



1—2	Compressor
2—3	Condenser
3—4	Expansion Valve
4—1	Evaporator

Figure 34: Effects of Hotter Air Supply for Condenser on Refrigeration Cycle

<u>Water-Water Heat Exchangers</u>

The following values are taken directly from APPENDIX I: INITIAL MODELING:

- Average flow rate to coffee machines—2.4 gallons per hour (1/2 of filtered water use)
- Average flow rate for domestic hot water heater—6 gallons per hour
- Hot water heater efficiency—0.92
- Coffee warmer efficiency—0.80

As a preliminary estimate, the flow rate of the water loop was predicted to be about 5 gallons per minute, or 300 gallons per hour. Because the flow rates for each hot water source are small relative to the flow of the water loop flow, the assumption was made that for city water entering a counter-flow heat exchanger with the water loop, the exiting temperature of the city water would be approximately equal to the temperature of the water loop, and the water loop, in turn, would be relatively unaffected by the heat exchange.

A reasonable estimate for city water temperature in Fort Collins is 60°F. A water loop temperature of 100°F would therefore mean the city water would enter the heat exchanger at 60°F and exit at approximately 100°F. This results in the following heat transfer requirements for a heat exchanger:

Table 26: Water-Water Heat Exchanger Sizing

Hot Water Source	Flow Rate, gph	Heat Transfer, Btu/hr
Domestic Hot Water Heater	6	2,002
Filtered Water to Coffee Brewers	2.4	801

Two Bell & Gosset brazed plate heat exchangers were selected to meet these requirements. Additional requirements of the heat exchangers included a small footprint, 3/4" thread sizes, and suitability for drinking water. The following figure illustrates the selected heat exchanger:

Figure 35: Water-Water Heat Exchanger

Heat Rejection System

The heat rejection system consists of two fans, each mounted on an air-to-water heat exchanger, located outside the building. For laboratory testing purposes, only one fan and one heat exchanger were required. The methods used to size the system were also used in the original system model, and are described below. In addition, experimentation revealed the need for an improved model. The testing methodology used to modify the model is also outlined below. Finally, the power consumption of the fan was measured using a plug-in power logger.

Sizing and Original Model

Several initial attempts were made at designing a heat rejection system. However, complications with aligning these modeled values with various manufacturer-provided data specs proved to be an excessively complicated task. Instead, a manufacturer was consulted, and provided with the performance requirements of the heat rejection system. The performance requirements were determined by assuming a worst case scenario, which was determined to be a 100°F day. The requirements included:

- 5 gallons per minute of water flow
- Water inlet and outlet temperatures 120°F and 100°F, respectively

- Approximately 19,200 Btu/hr heat rejection

The manufacturer selected was Valutech Mechanical & Thermal Solutions. After reviewing the requirements, Valutech suggested two 24X24 hydronic coil air to water heat exchangers, each operating with fan delivering 3,000 cfm. The fan was mounted to the radiator, as shown in Figure 36:

Figure 36: Heat Rejection System

In order to predict the performance of the heat rejection system under different environmental conditions, a model was created. This model relied on the Effectiveness-NTU Method, summarized as:

$$\dot{q} = \varepsilon C_{min} ETD$$

Where $\dot{q}$ refers to the heat transfer rate, ε refers to the heat exchanger effectiveness, C_{min} refers to the minimum value between C_{air} and C_{water}, and ETD refers to the entering temperature difference of the two fluids. C_{air} and C_{water} refer to the heat capacity rate of either fluid, and are calculated as the product of the mass flow rate of the fluid and its specific heat capacity.

The model relied on a combination of manufacturer performance data and fluid properties of both air and water. Specific data for the 24X24 unit was unavailable. Instead, data for the 22X22 unit was used as a conservative estimate. The following tables contain the manufacturer specifications used in the model (MS), the fluid properties (FP), assumed values (A), and the calculated values (C):

Table 27: (1) Water-Side Properties. (2) Air-Side Properties. (3) Heat Exchanger Performance Data

(1) Water Side	
Flow Rate, gpm	20 (MS)
Heat Capacity, Btu/lbm-°F	1 (FP)
Entering Temperature, °F	180 (MS)
Heat Capacity Rate, Btu/s-°F	2.78 (C)

(2) Air Side	
Heat Capacity, Btu/lbm-°F	0.24 (FP)
Entering Temperature, °F	60 (A)

(3) Heat Exchanger Performance			
Flow Rate, cfm (MS)	Heat Transfer, Btu/hr (MS)	C_{min} (C)	ε (C)
2,400	178,200	0.63	0.65
2,600	185,152	0.69	0.62
2,800	190,890	0.74	0.60
3,000	195,412	0.79	0.57
3,200	198,720	0.84	0.54
3,386	200,000	0.89	0.52

Since the performance data provided by the manufacturer did not include an ambient operating temperature for the unit, a value of 60°F was assumed for the model. It should be noted that model outputs show variation when this value is changed. With the information from Table 27, a relationship between ε and C_{min} could be defined. This was done by plotting ε on the y-axis versus C_{min} on the x-axis, as shown in Figure 37:

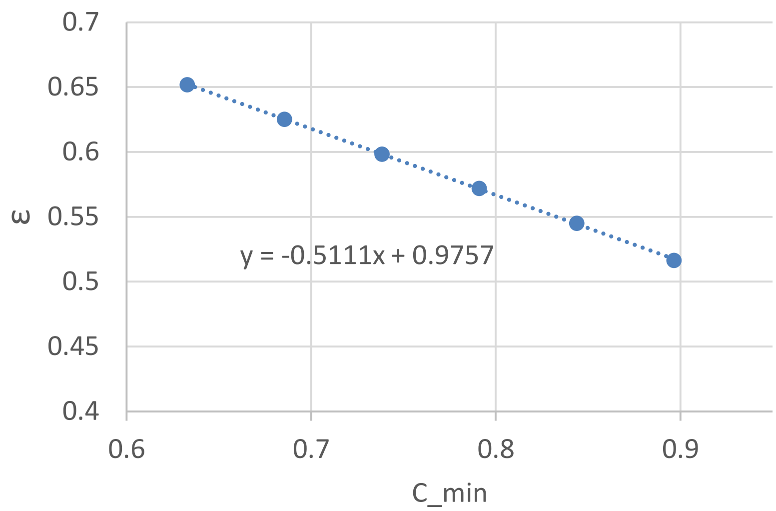


Figure 37: Outside Heat Rejection Heat Exchanger Performance Equation

For the water loop heat recovery system, the preliminary estimate for the water flow rate was 5 gallons per minute. This equates to a heat capacity rate, C_{wat}, of 0.695 Btu/s-°F. At the manufacturer specified 3,000 cfm, this means that the limiting heat transfer fluid, C_{min}, in the case of the water loop heat recovery system is the water, as opposed to the air in the heat exchanger performance modeling.

Using the heat exchanger performance equation to estimate heat exchanger effectiveness and a combined added heat load of 19,200 Btu/hr, the model predicted a high water temperature of 109°F on a 100°F day for two heat rejection units mounted on the roof top and connected in series.

A maximum water loop temperature of 110°F was determined to be the safety ceiling for the refrigeration units. Since the heat rejection system modeling predicted a maximum temperature lower than this even on the hottest day, the system was purchased and installed. However, laboratory testing of the system suggested that the initial model required substantial modification.

Modifications to Original Model

The original model for the heat rejection system relied on manufacturer performance specifications and several assumptions. During laboratory testing of all six refrigeration units, the heat rejection system was unable to maintain temperatures below 90°F. This was a substantially higher levelized temperature than the model predicted.

To improve the model, an experiment was designed to determine the heat transfer coefficient of the outside condenser, both for the fan on and fan off states. The set up for this experiment involved disconnecting the closed water loop, allowing the water to exit the system immediately after passing through the outside condenser. Cold city water was run through a hose and connected to the overflow tank. To ensure the flow rate was constant, the circulation pump was used, which delivers 5.5 gallons per minute. Great care was taken to ensure the flow delivered by the hose was equal to that delivered by the pump. Temperature loggers were placed on either side of the condenser. A functional representation of the experimental setup is shown in Figure 38:

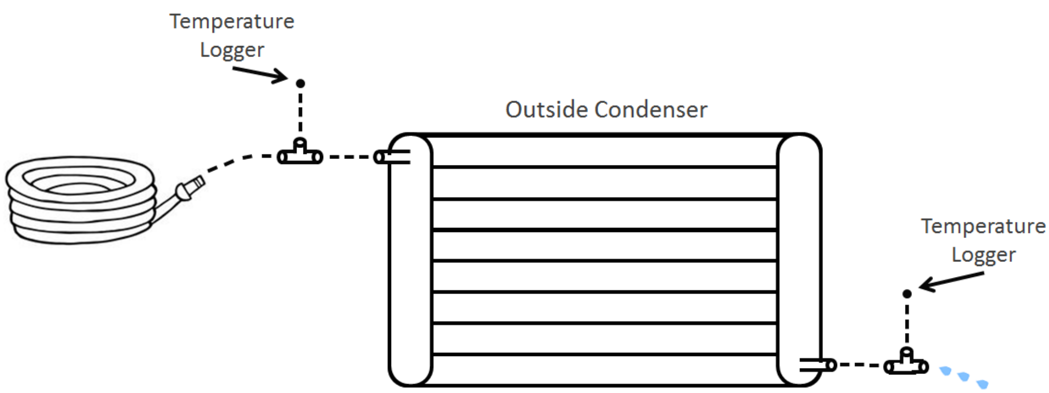


Figure 38: Outside Condenser Heat Transfer Coefficient Testing

The system was allowed to run for a 30-minute period. Ten minutes were given each for the setup, the fan on condition, and the fan off condition. With the flow rate known, the inlet and outlet temperatures of the water logged, the fluid properties of both the air and the water known, and the ambient air temperature at 70°F, the following equations could be used to determine the heat transfer coefficient of the condenser in either scenario:

$$\dot{q} = \dot{m}_{water} C_{p_{water}} (T_{o,water} - T_{i,water})$$

$$\dot{q} = \dot{m}_{air} C_{p_{air}} (T_{i,air} - T_{o,air})$$

$$\dot{q} = hA(T_{average,air} - T_{average,water})$$

This setup provides three equations and four unknowns. However, *A* represents the surface area of the condenser coils, a value provided by the manufacturer. Therefore, with three equations and three unknowns, the value for the heat transfer coefficient for the outside condenser was easily determined.

Cold water was pumped through the condenser, and the temperature of the water entering and exiting the condenser was logged. With ambient air at 70°F, the water should exit at a warmer temperature. Fan on and fan off scenarios were simulated. The results are shown in Figure 39:

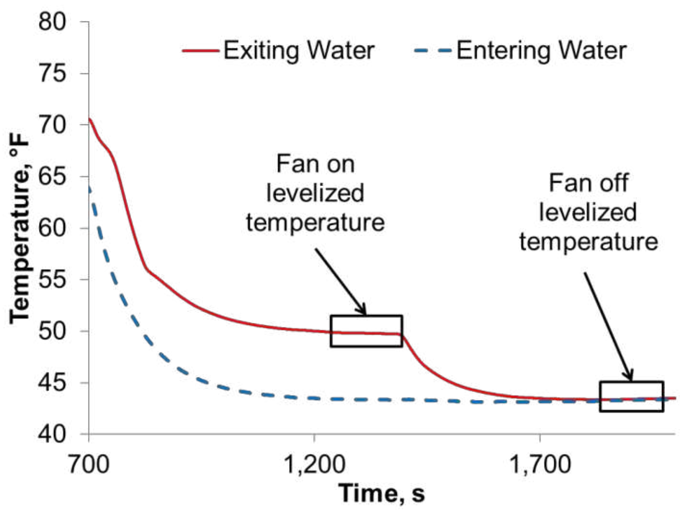


Figure 39: Outside Condenser Testing

The fan on test case began at about 700 seconds. The levelized temperature was taken as the final value before the fan was turned off. The levelized temperature for the fan off test case was similarly determined. In both cases, the entering water temperature was subtracted from the levelized exiting water temperature to acquire a temperature difference through the condenser.

This temperature difference was applied to the general heat transfer equation for fluid flow, along with the known value for the flow rate of the pump (5.5 gallons per minute, or 0.7645 lbm/s) and the heat capacity of water. After solving for the heat transfer rate of the condenser in either case, the same heat transfer equation was reapplied to the air side of the condenser. The fan delivers 3,000 cfm (3.75 lbm/s), and the heat capacity of air is 0.24 Btu/lbm°F.

With the entering and exiting temperatures of both fluids known, the general equation for convective heat transfer was used to determine the heat transfer coefficient of the condenser in both cases. This process is outlined in Figure 40:

A.

Water Side Calculations			
Entering Temperature, °F	43.2	Mass Flow Rate, lbm/s	0.7645
Exiting Temperature, °F	49.7	Specific Heat Capacity, Btu/lbm-°F	1

$$\dot{q} = 0.7645\frac{lbm}{s}\times 1\frac{Btu}{lb_m{}^\circ F}\times(49.7^\circ F - 43.2^\circ F) = \mathbf{17{,}916\ Btu/hr}$$

Air Side Calculations			
Entering Temperature, °F	70	Mass Flow Rate, lbm/s	3.75
Heat Transfer Rate, Btu/hr	17,916	Specific Heat Capacity, Btu/lbm-°F	0.24

$$T_{air,exiting} = 70^\circ F - \frac{17{,}916\frac{Btu}{hr}}{3.75\frac{lbm}{s}\times 0.24\frac{Btu}{lb_m{}^\circ F}} = \mathbf{64.5^\circ F}$$

Heat Transfer Coefficient Calculations			
Average Water Temperature, °F	46.4	Heat Transfer Rate, Btu/hr	17,916
Average Air Temperature, °F	67.2	Condenser Surface Area, ft^2	15.18

$$h = \frac{17{,}916\frac{Btu}{hr}}{15.18\ ft^2\times(67.2^\circ F - 46.4^\circ F)} = \mathbf{56.75\ \frac{Btu}{hr\ ft^2{}^\circ f}}$$

B.

Water Side Calculations			
Entering Temperature, °F	43.2	Mass Flow Rate, lbm/s	0.7645
Exiting Temperature, °F	43.4	Specific Heat Capacity, Btu/lbm-°F	1

$$\dot{q} = 0.7645\frac{lbm}{s}\times 1\frac{Btu}{lb_m{}^\circ F}\times(49.7^\circ F - 43.2^\circ F) = \mathbf{623\ Btu/hr}$$

Air Side Calculations			
Entering Temperature, °F	70	Mass Flow Rate, lbm/s	3.75
Heat Transfer Rate, Btu/hr	623	Specific Heat Capacity, Btu/lbm-°F	0.24

$$T_{air,exiting} = 70^\circ F - \frac{17{,}916\frac{Btu}{hr}}{3.75\frac{lbm}{s}\times 0.24\frac{Btu}{lb_m{}^\circ F}} = \mathbf{69.8^\circ F}$$

Heat Transfer Coefficient Calculations			
Average Water Temperature, °F	43.3	Heat Transfer Rate, Btu/hr	623
Average Air Temperature, °F	69.9	Condenser Surface Area, ft2	15.18

$$h = \frac{17{,}916\frac{Btu}{hr}}{15.18\ ft^2\times(67.2^\circ F - 46.4^\circ F)} = \mathbf{1.54\frac{Btu}{hr\ ft^2{}^\circ f}}$$

Figure 40: Outside Condenser Heat Transfer Coefficient Calculation Process for (A) Fan On Condition, (B) Fan Off Condition

The computed values of the heat transfer coefficient of the condenser for the fan on and fan off states were 56.75 Btu/hr-ft^2°F and 1.54 Btu/hr-ft^2°F, respectively. In addition, the measured value for the power consumption of the fan was 160 Watts.

Circulation Pump

Although the arrangement of the equipment sizing section might tend to suggest a chronological order in which each component was sized, the reality was that the sizing occurred simultaneously. The sizing of all components was dependent on the flow rate of the circulation pump. Both the flow rate and the required pressure of the pump were, in turn, dependent on the other system components.

Using the outside heat rejection model, the following curve was developed to help better understand the system's response to variations in water loop flow:

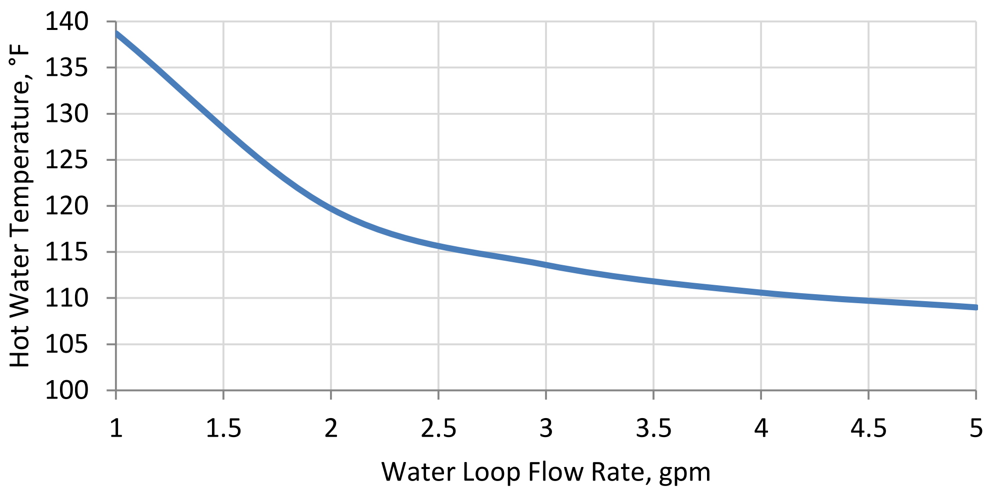


Figure 41: Water Loop Response to Variations in Flow Rate as Simulated at 100°F Ambient Temperature

As a safety precaution for the refrigeration units, a ceiling temperature of 110°F was suggested. According to Figure 41, the flow rate of the circulation pump should therefore be at least 5 gallons per minute. In addition, the suggested flow rate for the largest water-cooled condenser coils in the system is 4.5 gallons per minute. For these reasons, it was determined that the circulation pump should be able to deliver at least 5 gallons per minute.

In addition to the flow rate, the pressure requirements of the pump had to be determined. The manufacturer of each component in the water loop system was consulted to determine the pressure drop through the component as a function of flow rate, and the following table and figure were created:

Table 28: Pressure Drop across System Components

Water Loop Flow Rate, gpm	CX-H-033	CX-H-050	CX-H-100	CX-H-150	Outside Radiator	Water-Water Heat Exchanger	Total Pressure Drop, psi
2	0.3	1.3	2.3	1.5	0.6	1.0	**11.0**
3	0.5	3.2	4.1	3.2	0.7	2.2	**21.6**
4	0.6	5.2	6.0	4.8	0.8	3.3	**32.2**
5	0.8	7.1	7.8	6.4	0.9	4.5	**42.8**
6	1.0	9.1	9.7	8.1	1.0	5.7	**53.3**
7	1.1	11.0	11.5	9.7	1.1	6.8	**63.9**
8	1.3	13.0	13.4	11.3	1.2	8.0	**74.5**

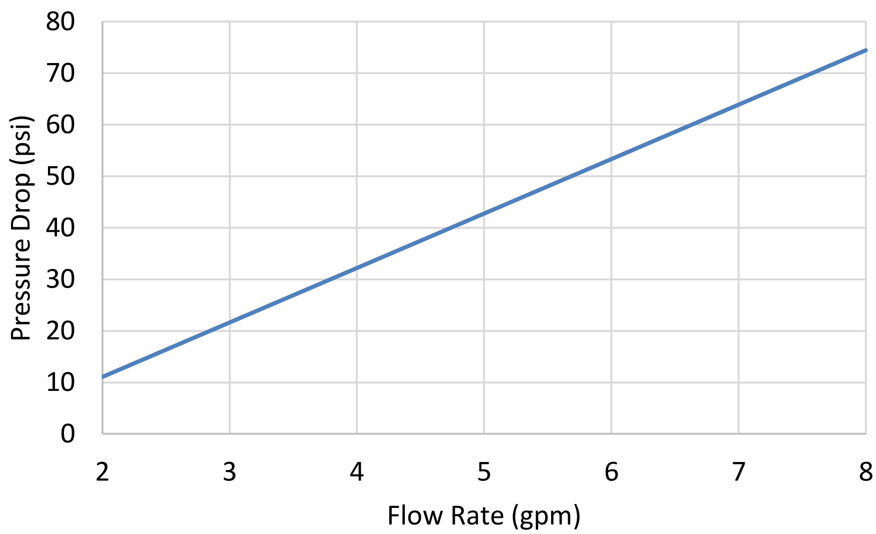


Figure 42: Water Loop System Curve

From the system curve, the pressure requirement of a pump delivering 5 gallons per minute is about 42 psi. The pump selected was a Procon rotary vane pump powered by a 1/3 hp Dayton motor. A picture of the pump mounted on the motor as well as the performance curve for this pump is shown below:

Figure 43: Circulation Pump and Motor

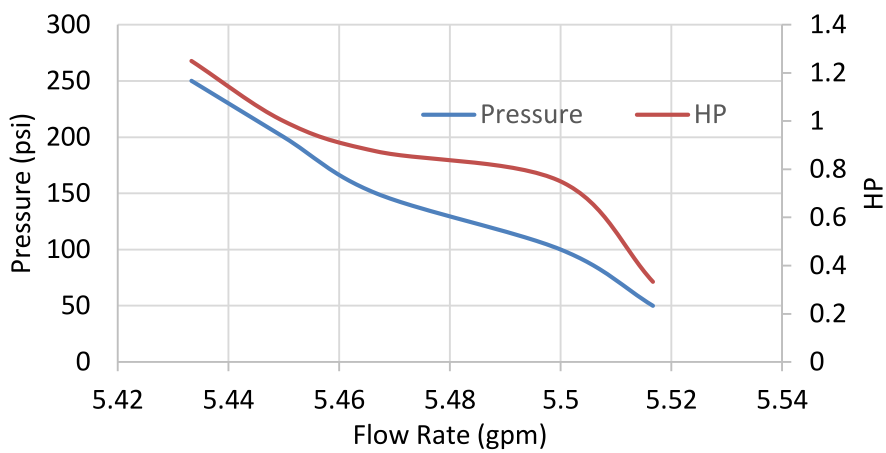


Figure 44: Circulation Pump Performance Curve

From the curve, it is evident a flow rate of 5 gallons per minute is not achievable. The flow rate of the circulation pump was verified by pumping water through the system, including all heat exchangers and condenser coils, and then disconnecting the end of the line to allow the water to instead fill a five gallon bucket. The amount of time the pump required to fill the five gallon bucket was measured five times, and the average of these flow rate values, determined as 5.5 gallons per minute, is used in the final model. Power consumption was measured directly using a DENT power logger to be 400 Watts.

An important design consideration when dealing with a variable temperature water loop is the effects of fluid properties on system parameters. Specifically, the impacts of density and viscosity on the flow rate and power consumption of the pump should be understood. The water loop is expected to remain within a temperature range between 60°F-120°F. Table 29 shows water properties in this range.

Table 29: Water Properties

Temperature, °F	Density, slugs/ ft^3	Kinematic Viscosity, $(ft^2/s) \times 10^{-5}$
60	1.938	1.21
70	1.936	1.052
80	1.934	0.926
90	1.931	0.823
100	1.927	0.738
120	1.918	0.607

From Table 29, it is evident the density of water remains essentially constant for the given temperature range. However, the viscosity of the water at 60°F is double the viscosity at 120°F. The relationship between viscosity and flow rate begins with the following equation:

Equation 12

$$\Delta P = f\frac{\rho L}{D}\frac{v^2}{2}$$

Where,

ΔP = Pressure loss through the system

ρ = Density of water

L = Length of water flow network

D = Diameter of piping

v = velocity of water

f = friction factor

Then, the pressure loss through the system can applied to the pump curve to determine the flow rate delivered by the pump. If a flow rate of 5.5 gallons per minute is assumed, a Reynolds number can be computed for each viscosity increment between water temperatures of 60°F to 120°F. The Reynolds number is defined as:

$$Re = \frac{vD}{\nu}$$

Where,

Re = Reynolds number

v = velocity of water, 4.02 ft/s (5.5 gal/min through a ¾ inch diameter pipe)

D = Diameter of pipe, 0.75 inches

ν = Kinematic viscosity, see Table 29

With the Reynolds number known, the Moody diagram can be consulted to determine a friction factor. The water loop heat recovery system uses plastic PEX piping, which can be approximated as smooth piping. A sample Moody diagram is shown Figure 45. For each temperature increment, the viscosity and subsequent Reynolds number and friction factor are shown in Table 30.

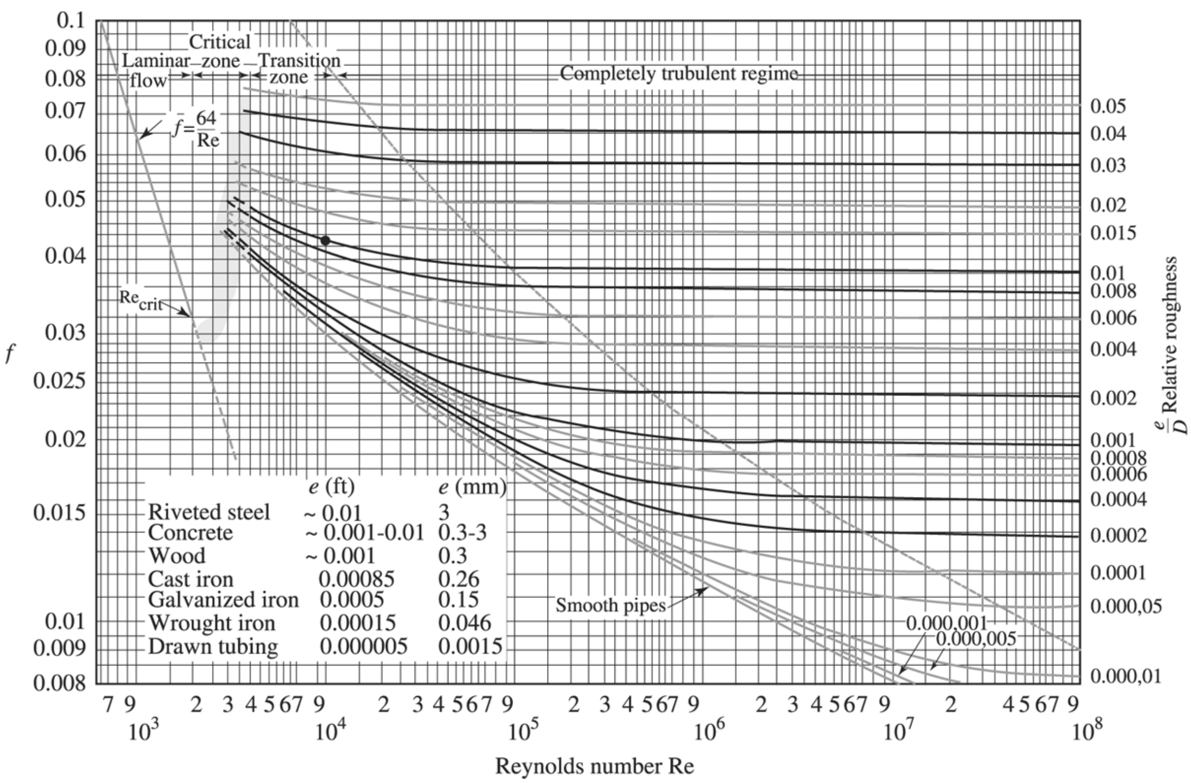


Figure 45: Moody Diagram

Table 30: Reynolds Number and Friction Factor for Flow Rate of 5.5 gal/min

Temperature, °F	Density, slugs/ ft^3	Kinematic Viscosity, $(ft^2/s) \times 10^{-5}$	Reynold's Number	Friction Factor, f
60	1.938	1.21	20,757	0.155
70	1.936	1.052	23,875	0.153
80	1.934	0.926	27,123	0.15
90	1.931	0.823	30,518	0.144
100	1.927	0.738	34,033	0.14
120	1.918	0.607	41,377	0.137

The friction factor changes from 0.155 to 0.137 over the entire temperature range the water loop could see. This represents an 11% reduction in friction factor for a 60°F increase in water temperature. From Equation 12, the pressure loss in the system is proportional to the friction factor, so an 11% reduction in friction factor would equal an 11% reduction of pressure losses in the system. The change in system pressure losses in turn results in a change in the required pressure delivered by the pump, which will in turn affect the flow rate according to the pump curve. However, by inspection of the pump curve in Figure 44, even a significant change in pressure (>200%) results in negligible change in flow rate (<0.1 gpm). Therefore, the effects of changing viscosity are not expected to significantly impact the flow rate of the system. However, as water temperature increases and viscosity subsequently increases, the resulting reduction in the friction factor of the water will in turn result in a reduced pressure drop through the system and therefore a reduced power consumption of the pump. In addition, the effects are expected to become less substantial for smaller flow rates.

APPENDIX III: DETAILED TESTING RESULTS

1-Door Freezer

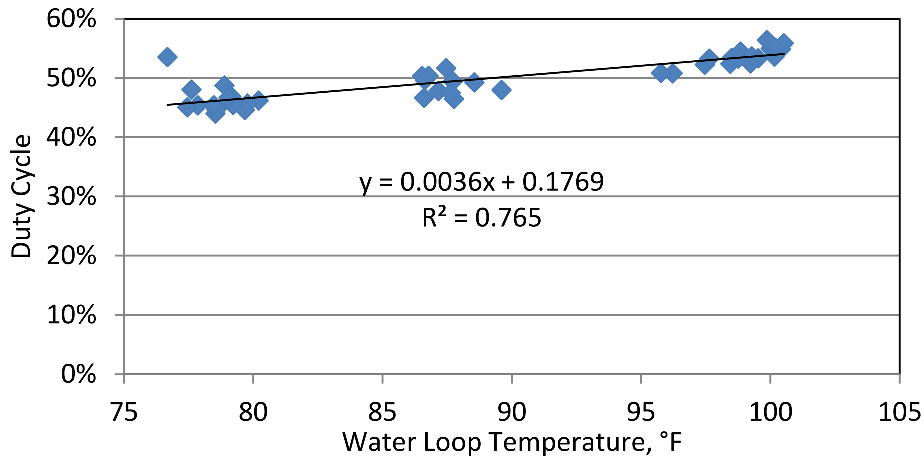


Figure 46: 1-Door Freezer Duty Cycle vs. Water Loop Temperature

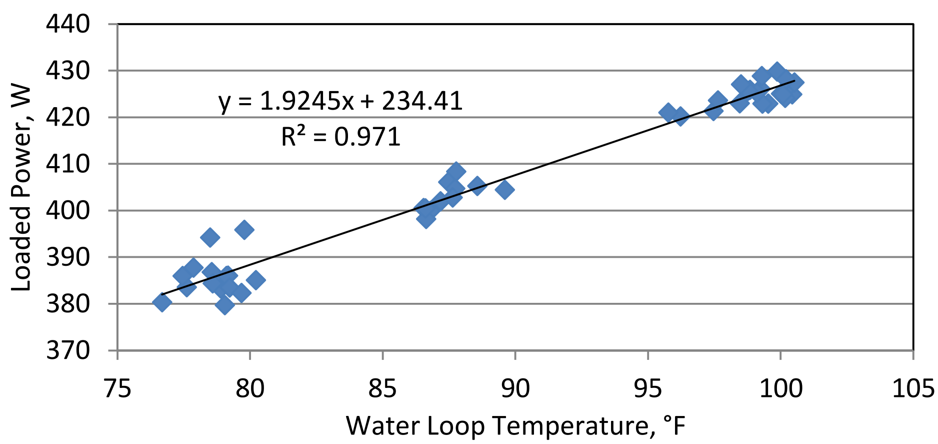


Figure 47: 1-Door Freezer Loaded Power vs. Water Loop Temperature

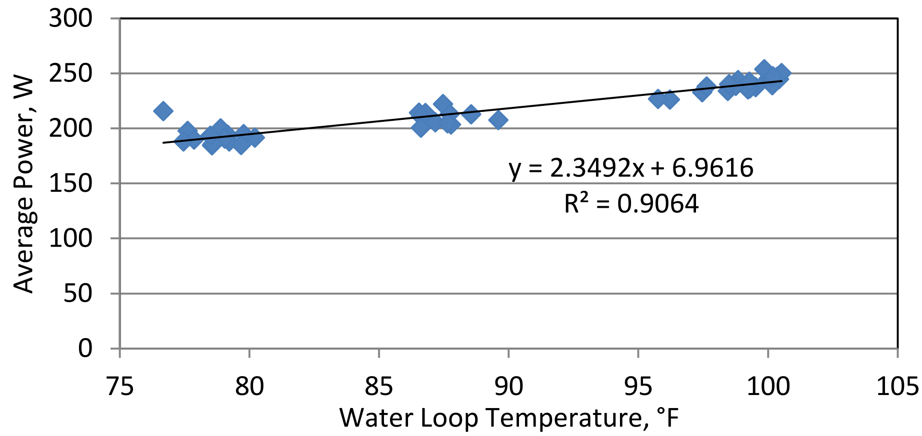


Figure 48: 1-Door Freezer Average Power Draw vs. Water Loop Temperature

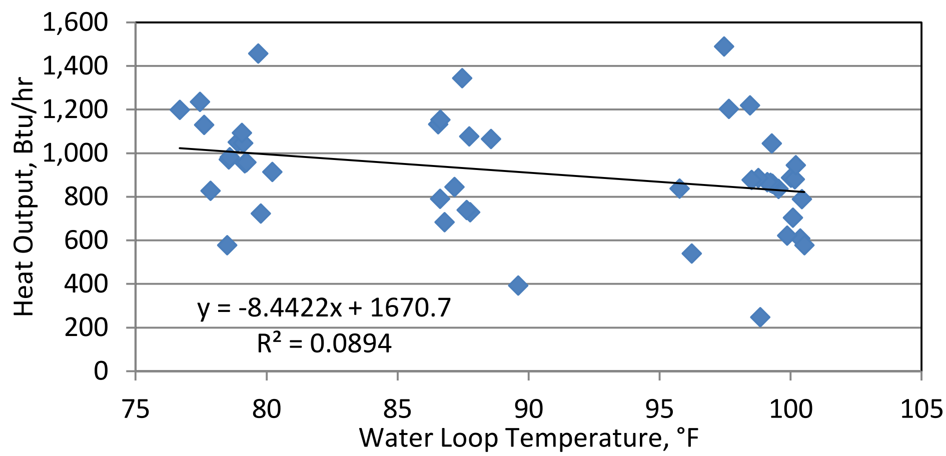


Figure 49: 1-Door Freezer Heat Output vs. Water Loop Temperature

Table 31: 1-Door Freezer Testing Results Summary

Water Loop Temperature, °F	79	87	99
Loaded Power, W	386	403	425
Duty Cycle	46%	49%	54%
Average Power, W	193	210	240
Heat Output, Btu/hr	1,008	956	847

1-Door Refrigerator

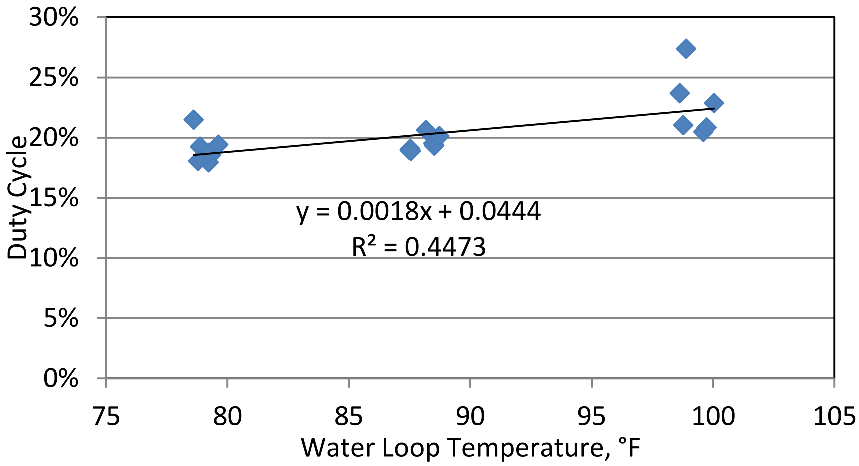


Figure 50: 1-Door Refrigerator Duty Cycle vs. Water Loop Temperature

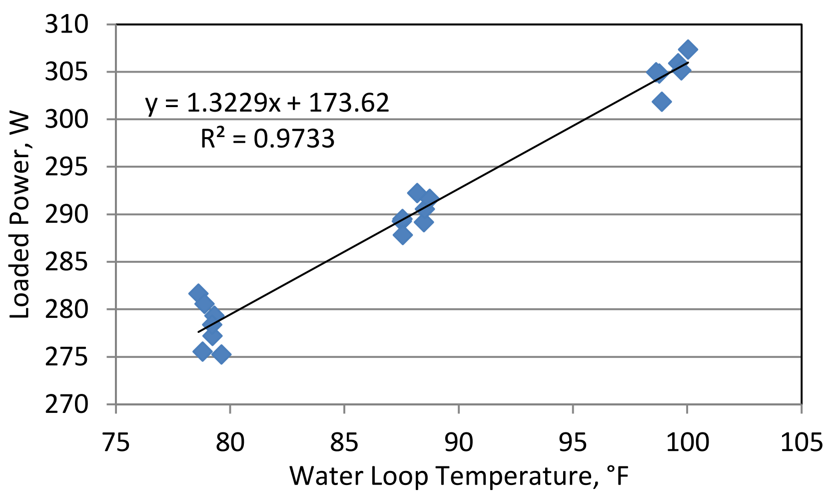


Figure 51: 1-Door Refrigerator Loaded Power vs. Water Loop Temperature

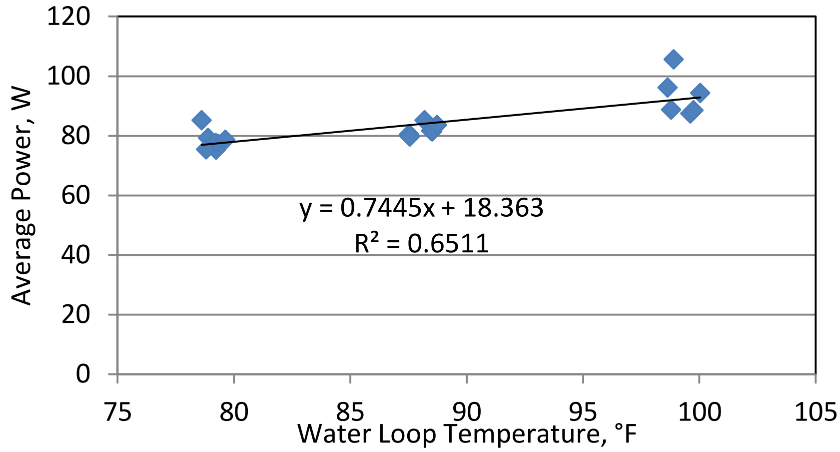


Figure 52: 1-Door Refrigerator Average Power Draw vs. Water Loop Temperature

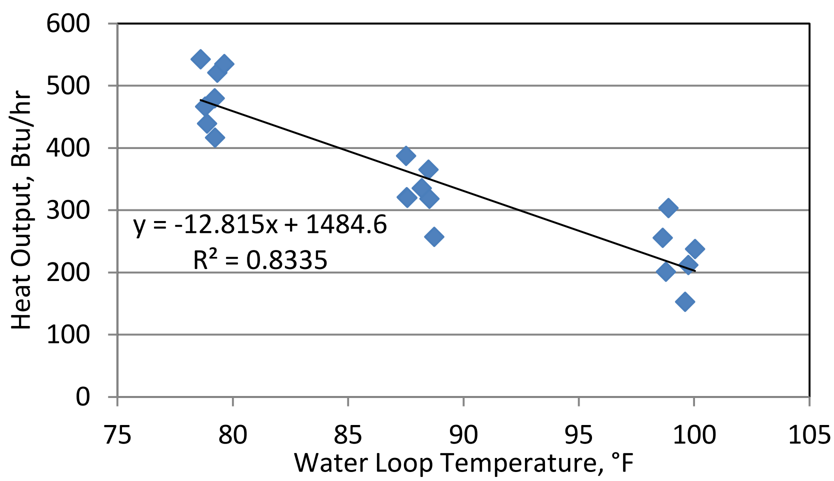


Figure 53: 1-Door Refrigerator Heat Output vs. Water Loop Temperature

Table 32: 1-Door Refrigerator Testing Results Summary

Water Loop Temperature, °F	79	88	99
Loaded Power, W	278	290	305
Duty Cycle	19%	20%	23%
Average Power, W	78	82	93
Heat Output, Btu/hr	486	329	227

2-Door Refrigerator

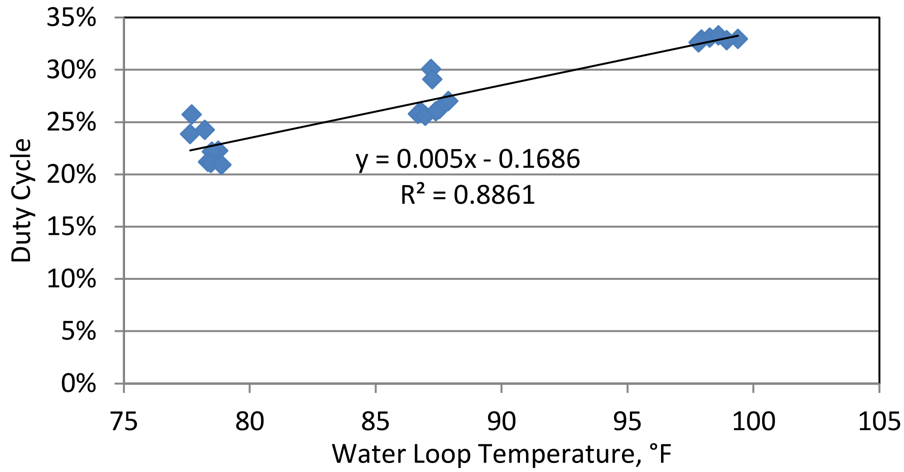


Figure 54: 2-Door Refrigerator Duty Cycle vs. Water Loop Temperature

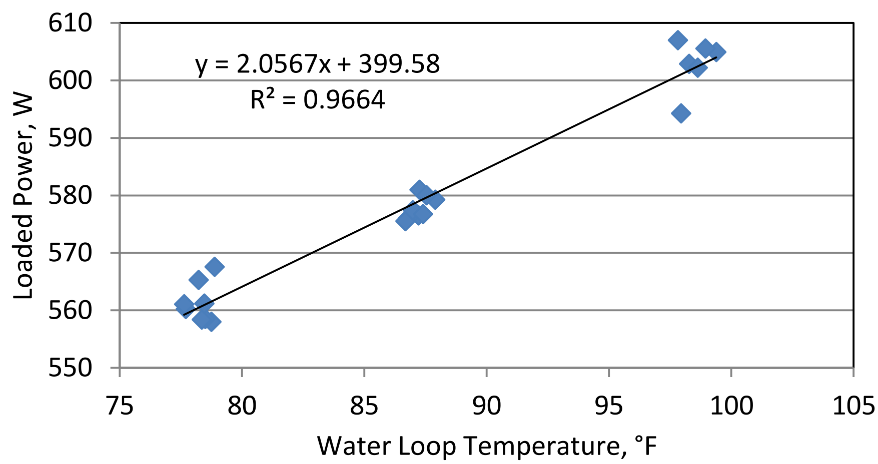


Figure 55: 2-Door Refrigerator Loaded Power vs. Water Loop Temperature

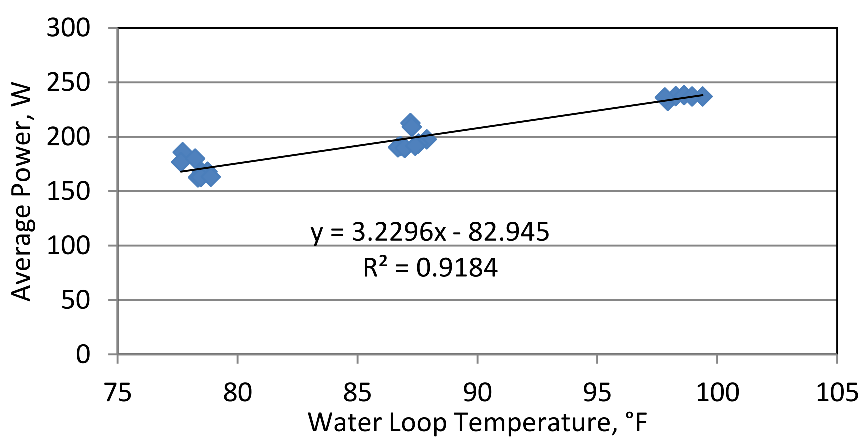


Figure 56: 2-Door Refrigerator Average Power Draw vs. Water Loop Temperature

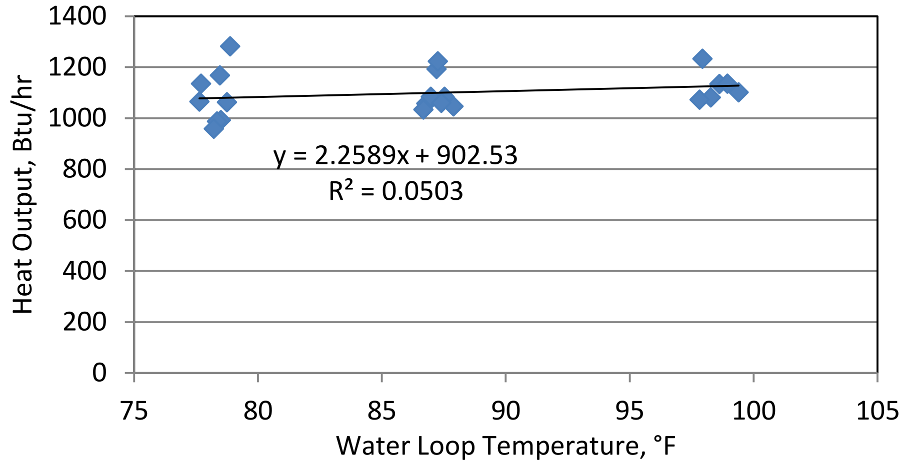


Figure 57: 2-Door Refrigerator Heat Output vs. Water Loop Temperature

Table 33: 2-Door Refrigerator Testing Results Summary

Water Loop Temperature, °F	78	87	97
Loaded Power, W	561	578	599
Duty Cycle	23%	27%	32%
Average Power, W	171	198	230
Heat Output, Btu/hr	1,081	1,103	1,116

Horizontal Food Case

Table 34: Horizontal Food Case Testing Results Summary

Water Loop Temperature, °F	76	85	91
Loaded Power, W	1,445	1,496	1,500
Duty Cycle	35%	35%	35%
Average Power, W	616	661	665
Heat Output, Btu/hr	4,768	4,528	4,006

Vertical Food Case

Table 35: Vertical Food Case Testing Results Summary

Water Loop Temperature, °F	79	83	89
Loaded Power, W	1,116	1,144	1,263
Duty Cycle	100%	100%	100%
Average Power, W	1,116	1,144	1,263
Heat Output, Btu/hr	7,772	8,336	8,465

APPENDIX IV: ADDITIONAL EXPERIMENTATION

COP Testing

In order to understand the relationship between a refrigerator's coefficient of performance and the thermal loading on the unit, a preliminary experiment was conducted. The set up for this experiment involved placing a temperature logger in a warm bucket of water. Then the bucket of water was placed inside of the single door refrigerator. The power consumption and heat rejection of the refrigerator were data logged for six refrigeration cycles. The objective was to see the response of refrigerator power consumption and heat output to the changing load conditions, represented as the changing bucket temperature.

A five-gallon bucket was placed in the single door refrigerator for a period of two hours. Six complete refrigeration cycles brought the bucket temperature from 59°F to 47°F. The average water temperature entering the condenser coil for the refrigerator was a consistent 72.2°F. These results are shown in Table 36.

Table 36: 1-Door Refrigerator Response to Added Heat Load

Variable	Cycle 1	Cycle 2	Cycle 3	Cycle 4	Cycle 5	Cycle 6	Predicted Value for 72°F Water Loop
Duty Cycle	34%	31%	30%	27%	26%	25%	17%
Loaded Power Draw, W	267	266	267	269	267	269	269
Average Power, W	111	106	101	95	93	92	72
Heat Output, Btu/hr	1,068	1,011	964	921	889	894	559
COP	2.83	2.81	2.80	2.83	2.81	2.86	2.27
Additional Heat Load from Bucket, Btu/hr	333	288	258	225	195	178	0

The graphs in APPENDIX III: DETAILED TESTING RESULTS demonstrate the experimentally derived predictive equations for the unit. These equations were used to predict what the conditions for the refrigerator should be at 72.2°. The predicted values were then compared to each of the six cycles from the COP experiment. The key observation is that the experimental characteristics of the refrigerator are close to their predicted values.

As mentioned above, the values for average power and heat output, the key values in determining COP, seem to be approaching the predicted value. The following two figures demonstrate the results of projecting the average power and heat output to the zero heat load condition:

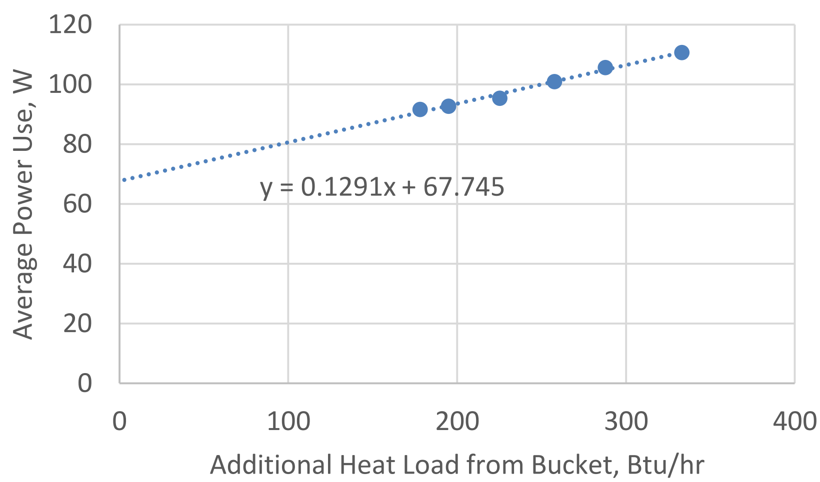


Figure 58: 1-Door Refrigerator Average Power Projected Response to Diminishing Heat Load

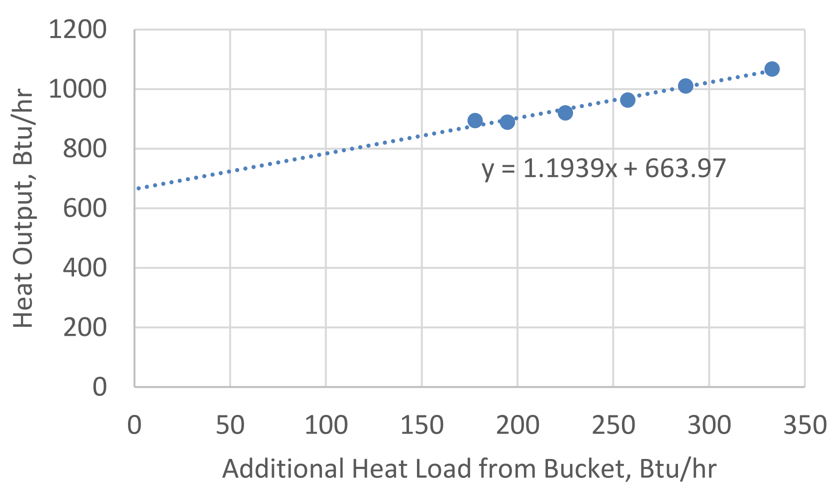


Figure 59: 1-Door Refrigerator Heat Output Projected Response to Diminishing Heat Load

The values projected by Figure 58 and Figure 59 are an average power of 67.7 Watts and a heat output of 664 Btu/hr. These differ from the values in Table 36. It is also notable that the COP seems to remains approximately constant at about 2.82, differing from the experimentally derived predictive equations which indicate the COP should be about 2.27. The reason for these differences is likely found in the missing data points. Eventually the additional heat load from the bucket would have reached zero, but the experiment was stopped when it had only reached about 178 Btu/hr. It is likely that if the experiment had run longer, the behavior of the data points between 178 Btu/hr and 0 Btu/hr would have shown a reduction in the COP over time, with the heat output dropping more and more sharply and the average power dropping slightly less sharply. In other words, a linear best fit line is reasonably representative of the existing data, but likely does not offer the same accuracy when projected beyond the data points.

An additional limitation to the data is that it only shows the response of the single door refrigerator to a changing heat load for the particular water loop temperature of 72.2°F. It is likely that the response will be different for different water loop conditions.

Store Simulation Testing

In addition to the testing above, another set of experiments was conducted to attempt to mitigate the loading discrepancy between in-store units and laboratory units. The goal of these experiments was to simulate the in-store loading conditions in the laboratory. The expectation was that by subjecting the laboratory refrigerators to similar loading conditions as the in-store refrigerators, the power consumption would be similar.

The experimental procedure involved six 1-hour visits to the 14944 Starbucks store. During these visits, the number of times each refrigerator was opened was tallied, and these tallies were averaged for each unit to acquire an approximate number of hourly openings of each unit. With the number of hourly openings known, the original laboratory testing of each refrigerator was repeated for a water loop temperature of 75°F, but with the inclusion of the refrigerator door openings. Table 37 contains the observations of the six visits to Starbucks store # 14944. Table 38 summarizes these observations.

Table 37: Results of Monitoring Starbucks Store # 14944 for Five 1-Hour Periods

Date	Start Time	End Time	Refrigerator	# of Openings
15-Mar	5:55 AM	6:55 AM	1-Door Freezer	3
			1-Door Refrigerator	0
			2-Door Refrigerator	1
15-Mar	6:55 AM	7:55 AM	1-Door Freezer	1
			1-Door Refrigerator	0
			2-Door Refrigerator	6
16-Mar	12:10 PM	1:10 PM	1-Door Freezer	0
			1-Door Refrigerator	0
			2-Door Refrigerator	1
17-Mar	5:00 PM	6:00 PM	1-Door Freezer	2
			1-Door Refrigerator	1
			2-Door Refrigerator	8
17-Mar	12:10 PM	1:10 PM	1-Door Freezer	3
			1-Door Refrigerator	0
			2-Door Refrigerator	7

Table 38: Summary of Store Monitoring

Unit	Average Number of Openings per Hour
1-Door Freezer	4.6
1-Door Refrigerator	1.8
2-Door Refrigerator	0.33

Next the laboratory units were tested again, this time incorporating the respective number of refrigerator door openings per hour. Table 39 shows a comparison of the laboratory air-cooled baseline to the in-store baseline, as well as the resulting average power of each refrigerator tested in the lab and including regular door openings. Finally, the water loop temperature was set to 75°F for this experiment. Table 39 also includes the results of the testing of the water-cooled refrigerators at 75°F.

Table 39: Laboratory Simulation of Store Operation versus Baselines

Unit	In-Store Air-Cooled Baseline Average Power Draw, W	Laboratory Air-Cooled Baseline Average Power Draw, W	Normal Testing Power Draw, Water Loop 75°F, W	Laboratory Simulation of In-Store Operations Power Draw, Water Loop 75°F, W
1-Door Freezer	268.5	195	194	209
1-Door Refrigerator	126.3	100	82	75
2-Door Refrigerator	460.2	185	182	177

By comparison, the in-store baseline average power draw is considerably higher for all three refrigerators than the laboratory baseline. However, the laboratory simulation did not demonstrate the expected increase in average power compared to the normal testing. It was determined that further experimentation was required to understand the difference between the laboratory units and the in-store units.

APPENDIX V: AC RUN TIME ALGORITHM

The algorithm used to estimate the run time of the air conditioning units operated in each of the 18 cities simulated using the final model is shown below:

$$ACRT_{high} = 16 + (1.5 + 1.5x) + (2.5 + 2.5y)$$

Where,

$ACRT_{high}$	=	Average daily air conditioning run time, hours
x	=	adjustment factor, derived from Starbucks logged data for 9,311 stores
y	=	Adjustment factor, derived from EIA's CBECS data

The base value of 16 was selected based on a high run time scenario to achieve run times between 18 and 24 hours. It should be pointed out that the high run time scenario is uncommon, accounting for only about 1.5% of all stores. The development of each adjustment factor is described below.

x-Adjustment Factor

Air conditioning run time data for 9,311 Starbucks stores was used to develop the x adjustment factor. A table was created aggregating the number of stores operating air conditioning within defined run time ranges (e.g., 20-24 hours) for each state. A sample of this table is shown in Table 40: Starbucks Stores AC Run Time by State

Table 40: Starbucks Stores AC Run Time by State

	Average Daily AC Run Time, Hours			
	20-24	15-20	8-15	0-8
AL	0	9	32	36
AK	0	0	1	8
AZ	2	43	169	153
AR	0	7	26	26
CA	28	214	722	902
CO	1	14	97	204

For each state, the percentage of stores within that state for which data was available that operate air conditioning units between 20-24 hours was computed. For example, for California, the percentage of high run time AC units is given as:

$$\frac{18}{18 + 214 + 722 + 902} = 1.5\%$$

The mean value was computed by taking the total number of stores in the 20-24 hour run time range and dividing by the total number of stores all together. The standard deviation was computed using the 50 high run time percentage values from the 50 states. Then for each state, the following equation was used to determine the adjustment factor, x:

Equation 13

$$\frac{\alpha - \mu}{\sigma}$$

Where,

α = Percentage of stores with AC run time between 20-24 hours for a given state

μ = Mean value for stores with AC run time between 20-24 hours, 0.016

σ = Standard deviation, 0.0232

For any states with zero Starbucks stores within the high run time range (20-24 hours), a value of (-1) was used for the x-adjustment factor. For all values greater that a standard deviation from the mean, a value of 1 was used for the x-adjustment factor.

<u>y-adjustment factor</u>

The y-adjustment factor was developed using the EIA's CBECS data. This survey contains information such as commercial building electricity usage for cooling for each of the 9 census regions. Furthermore, it also contains the number of commercial buildings per census region, as well as the square footage of each building. A simple analysis was used to determine the electricity usage per square footage of commercial buildings within each census region. These results are shown in

Table 41: Commercial Building Electricity Usage for Cooling by Census Region

Census Region	Electricity Usage for Cooling, Btu/ft^2
New England	3,766
Middle Atlantic	5,976
East North Central	5,312
West North Central	5,282
South Atlantic	14,072
East South Central	10,487
West South Central	14,742
Mountain	7,131
Pacific	7,198

The mean (8,220) and standard (3,970) deviation were determined for the 9 census regions. Equation 13 was then applied to each census region, with α representing the respective electricity usage for cooling. As an example, for the highest value for electricity usage is found in the South Atlantic Census region, and the computed value is given as:

$$\frac{14{,}072 - 8{,}220}{3{,}970} = 1.64$$

This value represents the highest computed value. To standardize, each subsequent value was then divided by this value, as shown in Equation 14.

Equation 14

$$\frac{\alpha - \mu}{\sigma} \times \frac{1}{1.64}$$

The value obtained by performing this calculation represents the y-adjustment factor. Phoenix used the y-adjustment factor for the West South Central Census region. For cities in California, the adjustment factor was further obtained by averaging the computed value with the computed value for the West South Central.