Space Heat Recovery from Refrigeration

Final Performance Report

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LEGAL NOTICE

ABBREVIATIONS AND ACRONYMS

AHU	Air Handling Unit
BHP	Brake Horsepower
BTU	British Thermal Unit
Btuh	British Thermal Units per Hour
CFM	Cubic Feet per Minute
EER	Energy Efficiency Ratio
EMS	Energy Management System
HFC	Hydrofluorocarbon
HR	Heat Recovery
HRCT	Heat Recovery Condensing Temperature
HVAC	Heating, Ventilation, and Air Conditioning
Hz	Hertz
kW	KiloWatt
kWh	KiloWatt-Hours
LT	Low-Temperature
MBH	1,000 British Thermal Units per Hour
MMBtu	One Million British Thermal Units
MT	Medium-Temperature
SCT	Saturated Condensing Temperature
SF	Square Feet
TD	Temperature Difference
WC	Water Column

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EXECUTIVE SUMMARY

Pacific Gas & Electric Company (PG&E) worked with a retail grocery store chain to study heat recovery in a new supermarket in Santa Clara County, California. Energy analysis and field monitoring were conducted to better understand the natural gas savings and consequent electric energy penalty associated with a heat recovery system using the heat from the market's refrigeration systems to heat the sales area. PG&E assisted the Customer and its controls and equipment vendors during the design and construction phase of the new supermarket to evaluate multiple design options, configure the recovery system, size and select the heat exchanger and holdback valves, and prepare control sequences of operation for the heat recovery controls. PG&E also assisted during start-up and commissioning phases.

PROJECT DESCRIPTION

The subject system is a direct-condensing system using heat from four of the store's six distributed refrigeration systems, with the four systems rejecting heat via a four-circuit parallel refrigerant-to-air heat exchanger coil in the main air handling unit that serves the store's sales area. Electronic pressure regulating "holdback" valves at the outlet of the heat exchanger facilitate both sensible and latent heat exchange—significantly increasing the quantity of heat recovered (versus systems without holdback valves, where only a portion of the latent heat is available), with a consequent increase in compressor discharge pressures and consumed compressor energy (which is typical for systems with holdback pressure control, but is less than the resulting heating energy and cost savings).

The performance of the heat recovery system was evaluated versus a theoretical Base Case system consisting of the same refrigeration and HVAC systems operating with the same ambient conditions, refrigeration loads, and heating loads as the system with heat recovery, but absent of all of the components related to heat recovery. The Base Case system performance was calculated analytically.

The refrigeration systems and air handling unit were outfitted with instrumentation and data acquisition equipment to monitor electric energy and natural gas usage. An on-site monitoring panel collected the sensor data and transmitted it for processing via wireless modem. The instrumentation included sensors to monitor the refrigerant pressure and temperature inside and downstream of the recovery coil, air temperature entering and leaving each recovery coil circuit, air flowrate, and natural gas flowrate. The store's Danfoss energy management system (EMS) was also used to obtain additional refrigeration system data for the compressors, condensers, and other system operating parameters.

Several challenges were encountered, including:

- Complications with incorporating the holdback valves into the heat recovery design. Contractors, controls vendors, and equipment vendors hesitated to use the valves, instead opting to use the less sophisticated (and lower performing) design without holdback pressure control. PG&E worked closely with the vendors and contractors to incorporate the holdback valves and pump-out circuits into the construction drawings, properly size the coil and valves, and to develop functional descriptions of the valve control sequences.

- Delays in the construction schedule, related to changes in the store's corporate ownership as well as post-construction store design changes. The store was originally scheduled to open on October 18, 2013, but the grand opening did not occur until May 2, 2014. Final commissioning of the heat recovery systems was not complete until August, 2014. In response, the monitoring phase of this Emerging Technologies study was shortened from 12 months to 6 months.
- Issues with automatic data transfer from the Danfoss EMS system, requiring close cooperation between PG&E, Danfoss, and the Customer's IT department, and multiple software revisions by Danfoss to its StoreView remote access software.
- Instrument failures that occurred during the construction and monitoring phase, notably the supply air flowrate meter (installed during the construction phase, the pressure lines from the pitot arrays were disconnected from the transducer/transmitter when the ceiling tiles were hung), and the natural gas flow meter (the original diaphragm-style meters were destroyed by natural gas inrush. After two diaphragm-style meters failed, a more robust rotary-style meter was used instead. Neither PG&E or the meter vendor had experience with this failure mode before).

The heat recovery system was fully commissioned in August, 2014. The dataset for this analysis starts on September 1, 2014, and ends on February 28, 2015.

PROJECT FINDINGS/RESULTS

The performance data presented in this report represents only the components related to the heat recovery system (refrigeration system compressors and condensers for the four subject refrigeration systems, and the main air handling unit), and does not include any of the other building systems. A summary of the results of the study are presented in Table 1 below. This store is open 24 hours per day, 7 days per week.

TABLE 1: HEAT RECOVERY EVALUATION RESULTS SUMMARY

	Energy Usage				Energy Cost			
	Energy	Natural Gas	Total Energy	Peak Demand	Energy	Demand	Natural Gas	Total
	(kWh)	(Therms)	(MMBtu)	(kW)	($)	($)	($)	($)
Without Heat Recovery	172,599	9,165	1,505.6	72.6	$14,442	$5,022	$8,698	$28,162
With Heat Recovery	182,908	1,430	767.3	74.1	$15,250	$5,125	$1,369	$21,744
Savings	(10,309)	7,735	738.3	(1.5)	($808)	($103)	$7,329	$6,418
Savings (%)	(6.0%)	84.4%	49.0%	(2.1%)	(5.6%)	(2.1%)	84.3%	22.8%
Interval data for September 1, 2014 to February 28, 2015								

Detailed analysis of the results is presented in the body of this report.

PROJECT CONCLUSIONS AND RECOMMENDATIONS

The evaluation shows that refrigeration heat recovery for space heating is a viable energy efficiency measure. In this application, natural gas usage was reduced by approximately 84% and operating cost was reduced by approximately 23% during the test period for the subject AHU and refrigeration systems.

Significant statewide energy savings are possible with widespread adoption of this measure. While California's energy efficiency standards (Title 24) require heat recovery for newly constructed commercial refrigeration (e.g. supermarket) applications, financial incentives are recommended for supermarket retrofit applications, and new-construction applications that are exempt from the Title 24 requirements but still have space heating loads and refrigeration capacity. However, substantial market support and training by the California utility companies (beyond just incentives) is needed to achieve the intended savings levels and market penetration, while balancing electric energy penalty and refrigerant charge increase. Specifically, the industry seems to have lost much of the technical understanding related to holdback valve utilization and control.

The focus of this project was a direct-condensing heat recovery system, where heat is exchanged from refrigerant directly to air. Other heat recovery configurations are viable, with comparable (or even higher) savings expectations, which would be more suitable for retrofit applications. One configuration in particular is the indirect design, where recovered heat is transferred from the refrigerant to an intermediate fluid (normally water or water-glycol) which is circulated through a fluid-to-air heat exchanger located in the air handling unit airstream.

HISTORY

Use of heat from refrigeration systems to provide space heating in supermarkets has a long history and at one time was used extensively and provided all or most of the heat in many stores, both in California and across the US. However, heat recovery became less common in recent decades, largely because of the perception that heat recovery systems significantly increase refrigerant charge and leakage. Many supermarket chains are again considering low-charge heat recovery systems as a way to reduce natural gas usage and operating costs, and in order to meet sustainability objectives. In addition, the 2013 California Title 24 Building Energy Efficiency Standards include requirements for heat recovery systems in new-construction projects. Title 24 mandates that at least 25% of the heat of rejection (THR) from refrigeration systems shall be used for space heating in new supermarkets.

ASSESSMENT OBJECTIVES

This project assessed the natural gas energy savings from a direct-condensing refrigerant-to-air heat recovery system for space heating in a supermarket in Santa Clara County, California. The project compared the performance of the heat recovery system to a "Base Case" system with no refrigeration heat recovery, a standard natural-gas furnace for space heating, and typical distributed single-stage R-507A parallel commercial refrigeration systems.

This project provides the necessary instrumentation, data acquisition equipment, and analysis required to monitor and evaluate the electric energy and natural gas usage of the refrigeration systems and air handling unit. The data was processed and compared to a theoretical scenario consisting of refrigeration and HVAC systems with no connected heat recovery capacity.

TECHNOLOGY EVALUATION

The subject heat recovery system consists of a four-circuit direct-condensing heat recovery coil installed inside the main air handling unit (AHU) that serves the supermarket's main sales area. Within the AHU, the coil is located upstream of the natural gas furnace, with each coil circuit connected to the discharge of one of four refrigeration systems (designations B, C, D, and E). The heat recovery coil is connected in series with the air-cooled refrigeration condensers for each refrigeration system. Three-way control valves divert refrigerant from the refrigeration compressors to the heat recovery coil and then to the refrigeration condensers when the system is in heat recovery mode. When the system is not in heat recovery mode, the three-way valve diverts refrigerant directly to the refrigeration condensers. Because the system is designed to operate with or without the reclaim activated, the sizing and specification of the condensers is equivalent to a design without reclaim. Pump-out circuits are included to evacuate the heat recovery coils to the refrigeration suction header when the system is not in heat recovery mode.

Four electronic holdback valves located immediately downstream of each of the heat recovery coil circuits control the refrigerant pressure inside the coil. Holding the refrigerant inside the coil at a higher pressure induces condensation of the refrigerant from a vapor state to mostly liquid, recovering much of the latent heat that would otherwise not be available without holdback valves. The valves are controlled so that the temperature difference (TD) between the mixed return/outside air and the refrigerant condensing temperature inside the heat recovery coil is held constant (subject to programmed maximum and minimum condensing temperatures). Figure 1 below is a schematic drawing of a refrigeration heat recovery system. For simplicity, only one of four circuits is shown.

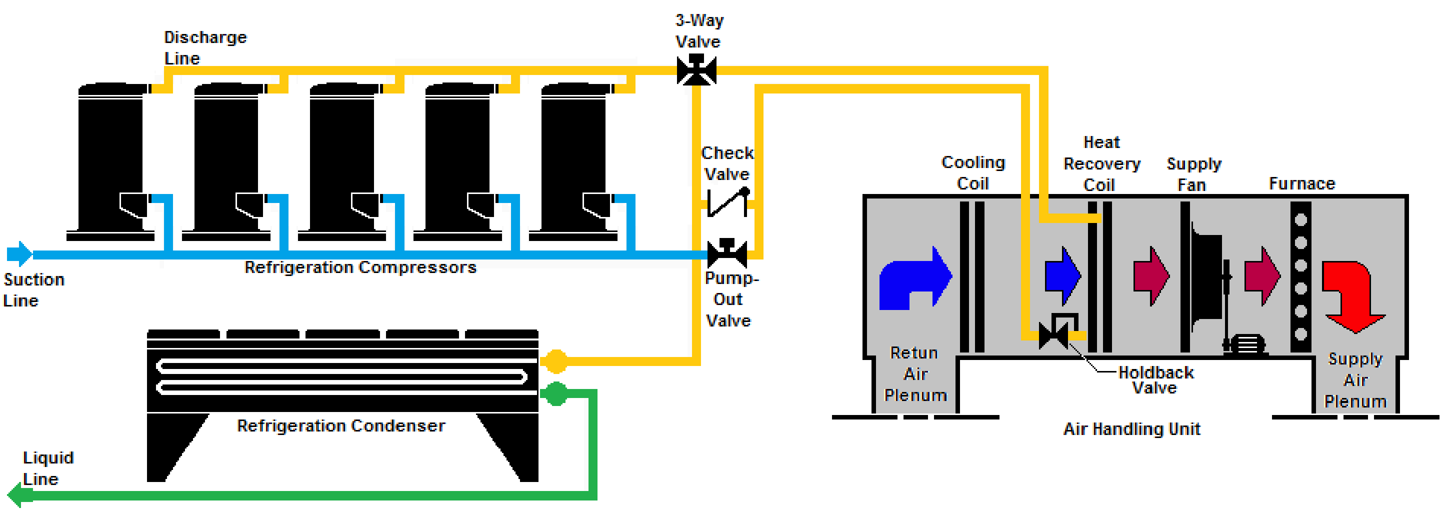


FIGURE 1: SCHEMATIC DRAWING OF HEAT RECOVERY SYSTEM

Heat recovery is the primary source for space heating. The AHU natural gas furnace is used only when heat recovery from all four refrigeration systems are already active and additional heating capacity is required. Staging of this process consists of:

- Stage 1: Heat recovery from Refrigeration Systems B and C
- Stage 2: Includes Stage 1, and adds the heat recovery from Refrigeration Systems D and E
- Stage 3: Includes Stages 1 and 2, and adds the AHU natural gas furnace

TEST METHODOLOGY

Pressure and temperature sensors were added to the four subject refrigeration systems. Airflow, natural gas flow, and air temperature sensors were added to the air handling unit where the heat recovery condensing coil is installed. A monitoring panel located in the condenser section of the air handling unit collected the sensor data and transmitted it for processing via wireless modem. The store's Danfoss energy management system was also used to obtain additional refrigeration system data for the compressors, condensers, air handling unit, space temperature and relative humidity, outside ambient temperature, and other system operating parameters.

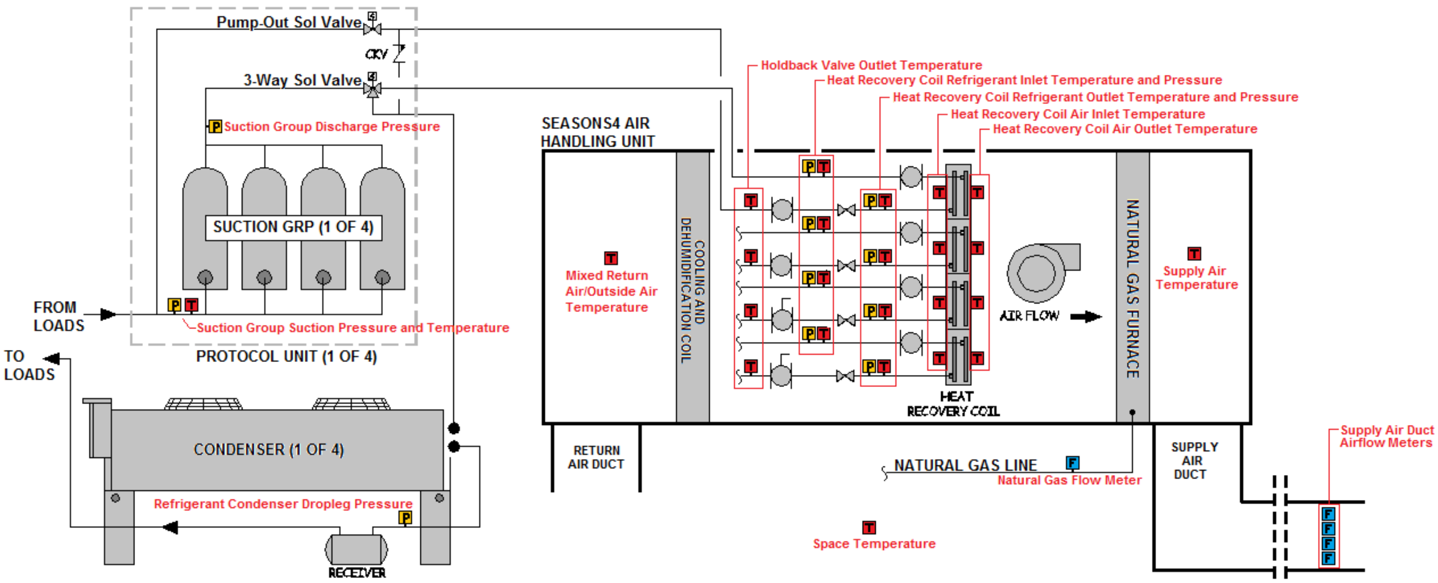


FIGURE 2: INSTRUMENTATION DIAGRAM FOR HEAT RECOVERY EVALUATION

For a detailed description of the evaluation methodology, refer to Appendix A: Analysis Plan.

RESULTS

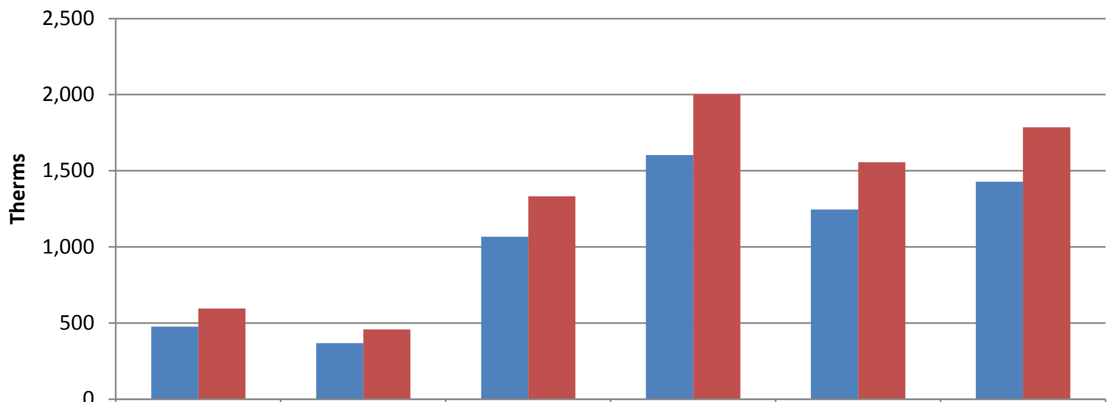
Presented below are the results for the heat recovery ET project. The recovery system is expected to save gas that would have been used in the furnace to heat the supply air, while using more electricity because of an increase in load on the AHU fan and an elevated energy usage in the refrigeration. The results are presented in the following sections:

- Natural Gas Usage and Savings
- Electric Energy Usage and Penalty
- Cost Results

The dataset for this analysis starts on September 1, 2014, and ends on February 28, 2015

NATURAL GAS USAGE AND SAVINGS

Figure 3 shows the total heat recovered, as well as the resultant amount of natural gas that was offset by the heat recovery system, for each month of the six-month test period.



	September	October	November	December	January	February
Heat Recovered	476	367	1,067	1,604	1,246	1,428
Natual Gas Savings	595	459	1,333	2,005	1,557	1,786

FIGURE 3: HEAT RECOVERED AND CONSEQUENT NATURAL GAS SAVINGS PER MONTH

In total, 6,188 Therms of heating load was served via the heat recovery system, offsetting 7,735 Therms of natural gas over the six months of the subject test period (based on a measured furnace combustion efficiency of 80%).

ELECTRIC ENERGY USAGE AND PENALTY

The amount of electric energy required by the refrigeration system and air handling unit is different for systems with heat recovery. Components that have different electric energy requirements with heat recovery include:

- AHU Supply Fan Energy (expected to increase)
- Condenser Fan Energy (expected to decrease)
- Refrigeration System Compressor Energy (expected to increase)

Because different elements of the heat recovery system may increase or decrease electricity consumption, the consumption for each of these subsystems is analyzed separately and discussed below.

AHU SUPPLY FAN ENERGY

All other factors being equal, the electric energy required by an AHU supply fan is slightly higher for systems with heat recovery because of the added air pressure drop across the heat recovery coil. For this study, the supply fan energy was not measured directly but was calculated based on measured airflow in concert with fan power data provided by the AHU manufacturer. Third-power fan affinity laws were used to calculate supply fan power at reduced speed, with the ratio of supply fan energy use with heat recovery to Base Case assumed to be linearly proportional to the ratio of total static pressure with heat recovery to without. The Table 2 below shows the static pressure and power assumptions that were used to calculate the AHU supply fan power.

TABLE 2: MAIN AHU SUPPLY FAN STATIC PRESSURE, POWER, AND ENERGY USAGE

	Base Case	With Heat Recovery
AHU Static Pressure seen by Supply Fan	4.01 In. WC	4.49 In. WC
Supply Fan Brake Horsepower at 100% Speed	17.33 BHP	19.4 BHP
Supply Fan Electric Power Usage at 100% Speed	14.1 kW	15.8 kW

The AHU supply fan speed is controlled with a variable speed drive. The fan is controlled at one of three fixed speeds, depending on the status of unit:

- The fan operates at 100% speed (60 Hz.) when the AHU is in the second stage of cooling or if both stages of heating and the gas furnace is on
- The fan operates at 90% speed (54 Hz.) if one or both stages of heat recovery are activated, but the furnace is not required
- The fan operates at 35% speed (21 Hz.) any time the AHU is neither in any stage of cooling or is in any stage of heating

CONDENSER FAN ENERGY

The condensing temperature control strategy for the subject refrigeration condensers is an ambient-reset (e.g. drybulb-following) strategy with fans cycling off and on to maintain a target saturated condensing temperature (SCT). The target SCT is calculated by adding a fixed control TD to the measured ambient drybulb temperature. For condensers with fan cycling control, the condenser heat rejection capacity is a function of the number of fans running as well as the temperature difference (TD) between the actual refrigerant saturated condensing temperature (SCT) and the ambient drybulb temperature. Since the SCT control strategy works to maintain a fixed TD, the number of condenser fans running (and therefore also electric energy consumed) is directly proportional to the heat rejection capacity. Consequently, the condenser electric fan power is lower when heat recovery is on because a portion of the heat rejection load is served by the heat recovery coil.

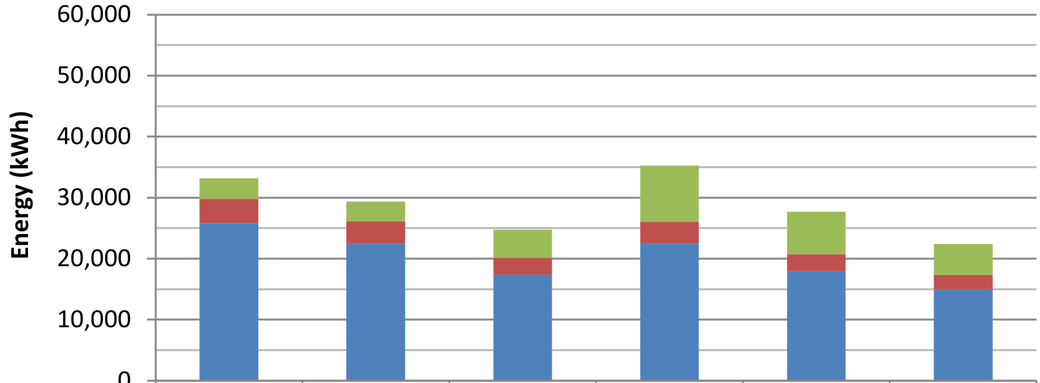
The control TD for all four of the subject refrigeration systems for this test was 10°F.

COMPRESSOR ENERGY

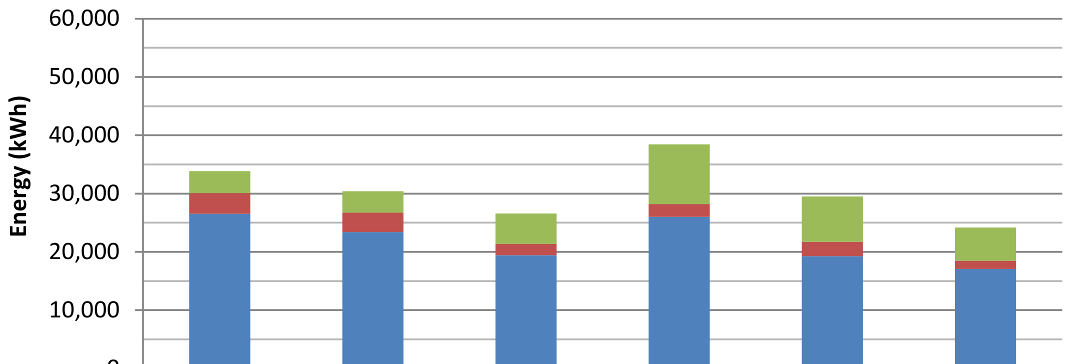
The subject heat recovery system employs condensing temperature control in the heat recovery coil via electronic holdback valves in order to reclaim both the sensible heat as well as, and more importantly, the majority of the latent heat of rejection from the refrigerant, which would not otherwise be recovered without control (the operation of the holdback valves is discussed later in this report). The valves maintain a heat recovery condensing temperature (HRCT) in the recovery coil that is nearly always higher than the SCT that could be maintained in the refrigeration condenser. Consequently, the discharge pressure "seen" by the refrigeration compressors is higher with heat recovery. Higher discharge pressures require more work from the compressors, increasing electric energy and demand.

Electric Energy Usage and Penalty Results

Figure 4 illustrates the calculated electric energy use by month for the refrigeration system compressors and condensers, and for the air handling unit supply fan. The electric energy use was calculated based on performance data provided by the component manufacturer, and measured capacity and run-times from the test data. The top figure shows the calculated Base Case energy usage, while the bottom figure shows the energy usage with the heat recovery system.



	September	October	November	December	January	February
Base Case Supply Fan	3,389	3,260	4,660	9,173	6,977	5,081
Base Case Condensers	3,959	3,581	2,711	3,555	2,782	2,371
Base Case Compressors	25,823	22,517	17,398	22,514	17,924	14,959



	September	October	November	December	January	February
Supply Fan with HR	3,795	3,650	5,218	10,271	7,812	5,689
Condenser with HR	3,528	3,349	1,942	2,178	2,454	1,421
Compressors with HR	26,530	23,384	19,407	26,010	19,246	17,063

Figure 4: Electric Energy Usage for AHU Supply Fan, Compressors, and Condenser Fans Per Month

Figure 5 below shows the total energy penalty (calculated as energy usage with heat recovery minus Base Case energy usage) for the entire test period, for each of the components analyzed in this study.

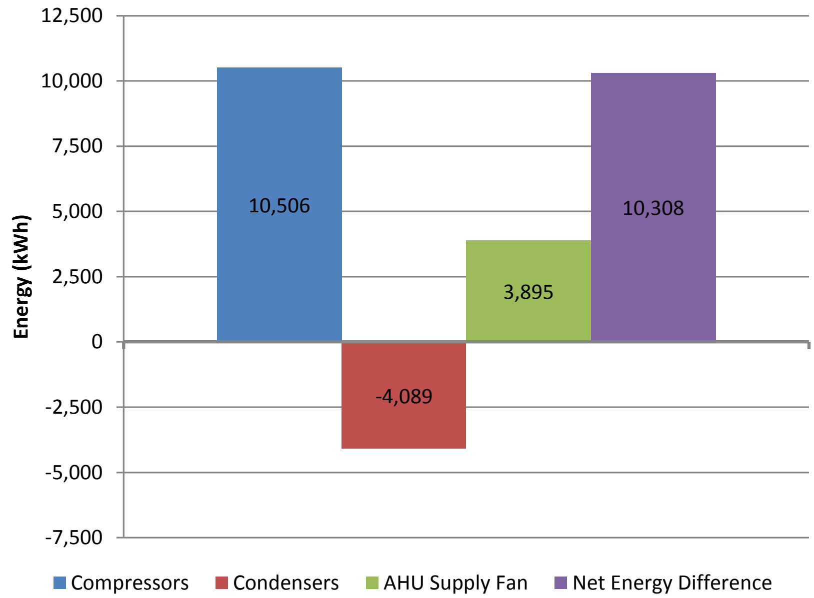


FIGURE 5: NET ENERGY PENALTY PER COMPONENT FOR FULL TEST PERIOD

Figure 5 shows that the refrigeration compressors and AHU supply fan required approximately 10,500 kWh and 3,900 kWh more energy respectively with the heat recovery unit installed over the course of the (6 month) test period, while the condenser fans used approximately 4,100 kWh less. The net energy penalty for the test period was approximately 10,300 kWh.

COST RESULTS

To obtain figures on the changes in energy costs, the raw data for energy consumption from the refrigeration system and the AHU gas meter were used in conjunction with rate tables to calculate monthly energy costs. PG&E's E-19 time-of-use rate schedule for secondary service, and PG&E's G-NR1 rate schedule were assumed for electric energy cost and natural gas cost, respectively, which are the appropriate schedule for this building size and type. Table 3 and Table 4 below show the energy costs that were assumed for this study.

TABLE 3: SUMMARY OF E-19 ELECTRIC ENERGY SCHEDULE

	Electric Demand		Electric Energy	
Season	**Time-of-Use Period**	**Demand Charge** (per kW)	**Time-of-Use Period**	**Energy Charge** (per kWh)
Summer	Max. Peak	$12.77	Peak	$0.14209
	Part Peak	$2.91	Part Peak	$0.09532
	Maximum	$7.26	Off Peak	$0.07618
Winter	Part Peak	$1.04	Part Peak	$0.08409
	Maximum	$7.26	Off Peak	$0.07327

TABLE 4: SUMMARY OF G-NR1 NATURAL GAS SCHEDULE

	Summer		Winter	
	First 4,000 Therms	Excess Therms	First 4,000 Therms	Excess Therms
Procurement	$0.56306	$0.56306	$0.56306	$0.56306
Transportation	$0.32278	$0.16637	$0.39418	$0.20318
Total Charge	$0.88584	$0.72943	$0.95724	$0.76624

Table 5 below summarizes the electric energy, electric demand, and natural gas cost by month, as well as the aggregate cost results for the test period.

TABLE 5: ELECTRIC ENERGY AND DEMAND COST, AND NATURAL GAS COST, FOR THE TEST PERIOD

	Base Case			With Heat Recovery			Difference		
Month	**Energy Cost ($)**	**Demand Cost ($)**	**Total Energy Cost ($)**	**Energy Cost ($)**	**Demand Cost ($)**	**Total Energy Cost ($)**	**Energy Cost ($)**	**Demand Cost ($)**	**Total Energy Cost ($)**
September	$2,964	$1,650	$4,615	$3,019	$1,686	$4,704	$54	$35	$90
October	$2,842	$1,535	$4,376	$2,934	$1,570	$4,505	$93	$35	$128
November	$2,080	$439	$2,519	$2,229	$449	$2,678	$149	$10	$159
December	$2,283	$467	$2,750	$2,490	$472	$2,962	$207	$5	$212
January	$2,168	$451	$2,618	$2,309	$465	$2,774	$141	$15	$156
February	$2,105	$480	$2,585	$2,269	$483	$2,753	$164	$3	$167
TOTAL	$14,442	$5,022	$19,463	$15,250	$5,125	$20,376	$809	$104	$912

	Natural Gas Usage (Therms)		Cost ($)		
Month	**Base Case**	**With Heat Recovery**	**Base Case**	**With Heat Recovery**	**Savings**
September	595	0	$527	$0	$527
October	459	0	$407	$0	$407
November	2,061	728	$1,973	$697	$1,276
December	2,005	0	$1,919	$0	$1,919
January	1,658	101	$1,587	$97	$1,490
February	2,387	602	$2,285	$576	$1,709
Total:	9,165	1,430	$8,698	$1,369	$7,329

	Energy Cost				
	Energy ($)	**Demand ($)**	**Total Electric Energy ($)**	**Natural Gas ($)**	**Total ($)**
Without Heat Recovery	$14,442	$5,022	$19,464	$8,698	$28,162
With Heat Recovery	$15,250	$5,125	$20,375	$1,369	$21,744
Savings	($808)	($103)	($911)	$7,329	**$6,418**
Savings (%)	(5.6%)	(2.1%)	(4.7%)	84.3%	**22.8%**

Table 5 shows that heat recovery offset $7,300, or 84%, of natural gas cost for heating energy. The consequent increase in electric energy and demand cost was $911, or 5%. Overall, the heat recovery system saved approximately $6,400, or 23% of the total energy cost for these 6 months.

ANALYSIS

Detailed analysis of the test results are presented in the following sections, including:

- Holdback Valve Operation
- Demand Analysis
- Loads and Energy Usage versus DOE2 Analysis
- System Performance versus Title 24 Requirements

HOLDBACK VALVE OPERATION

The heat recovery system uses four Sporlan-brand CDS electronic pressure regulating holdback valves, one per refrigeration system, at the refrigerant outlet of each of the heat recovery coil circuits. The valves' flow capacities are continuously modulated by the Danfoss controller to control the pressure in the heat recovery coils when the system is in heat recovery mode. Holding the refrigerant inside the coil at a higher pressure increases the saturated condensing temperature of the refrigerant, inducing condensation of the refrigerant from a vapor state to mostly liquid inside the recovery coil. This method recovers much of the latent heat from the refrigerant that would otherwise not be available without holdback valves. The use of electronic holdback valves with adjustable holdback setpoints in commercial refrigeration systems is a relatively new concept; part of the project included investigating the efficacy of this approach to increasing the performance of the recovery system.

The valve modulation of each of the four holdback valves is controlled independently by each of the respective refrigeration system controllers. The valves are modulated to maintain a target heat recovery condensing temperature (HRCT). HRCT is determined by adding the current AHU mixed air temperature (e.g. mixed return air and outside air) plus an adjustable heat recovery temperature difference (HRTD) of 18°F, subject to minimum and maximum limits.

In addition, the valve control strategy includes a minimum flow allowance of 10% which minimizes the risk of high pressure events due to valve hunting or fast changes in refrigeration load or discharge pressure.

Figure 6 below shows the HRCT and mixed return/outside air temperature for a sample 48-hour period, showing the HRCT varying by a fixed TD versus the mixed air temperature. For clarity, only System B is shown.

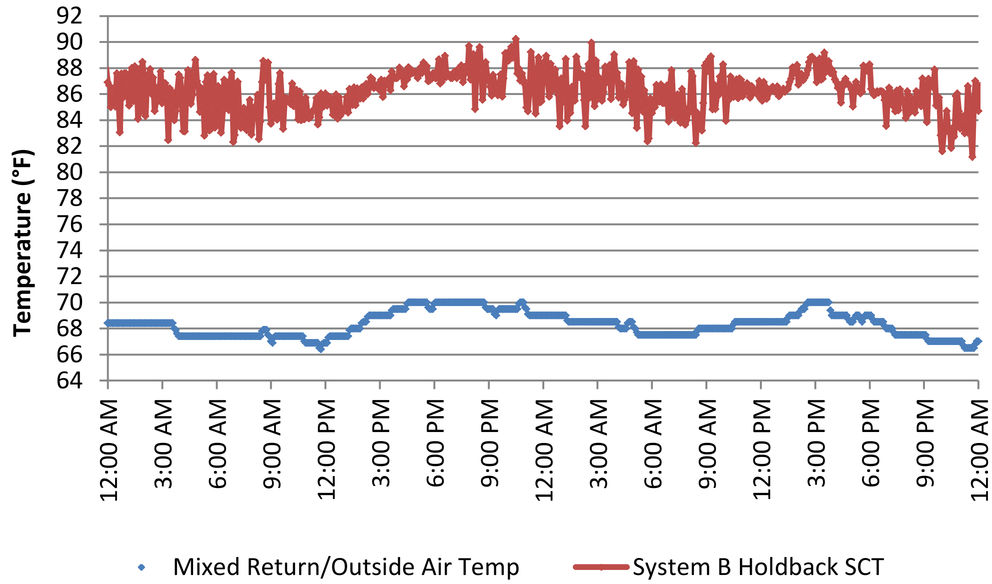


FIGURE 6: HOLDBACK SCT AND MIXED RETURN/OUTSIDE AIR TEMPERATURE FOR ONE RECOVERY SYSTEM FOR 48 HOUR SAMPLE PERIOD

VALVE CONTROL AND PRESSURE DROP ANALYSIS

The flowrate capacity of the holdback valve is a function of both the pressure drop across the valve and the position of valve piston from 0% (valve fully closed) to 100% (valve fully open). The position of the valve piston is controlled, while the pressure drop across the valve is a function of both the holdback pressure in the recovery coil and the condensing pressure in the refrigeration condenser downstream of the coil. Figure 7 below shows the approximate[1] pressure drop across the holdback valve and the valve position over one sample week of operation. The holdback valve position is represented by the red data points which reference the right vertical axis. The pressure drop across the valve is shown by the blue data points which reference the left vertical axis. For clarity, only one system (B) is shown.

[1] Pressure drop across holdback valve calculated as heat recovery coil pressure minus refrigeration condenser dropleg pressure. Pressure drop in the refrigeration condenser and the piping from the recovery coil to the condenser are assumed negligible for this analysis since only a qualitative understanding of pressure drop across the holdback valve is necessary to see the relationship versus valve position.

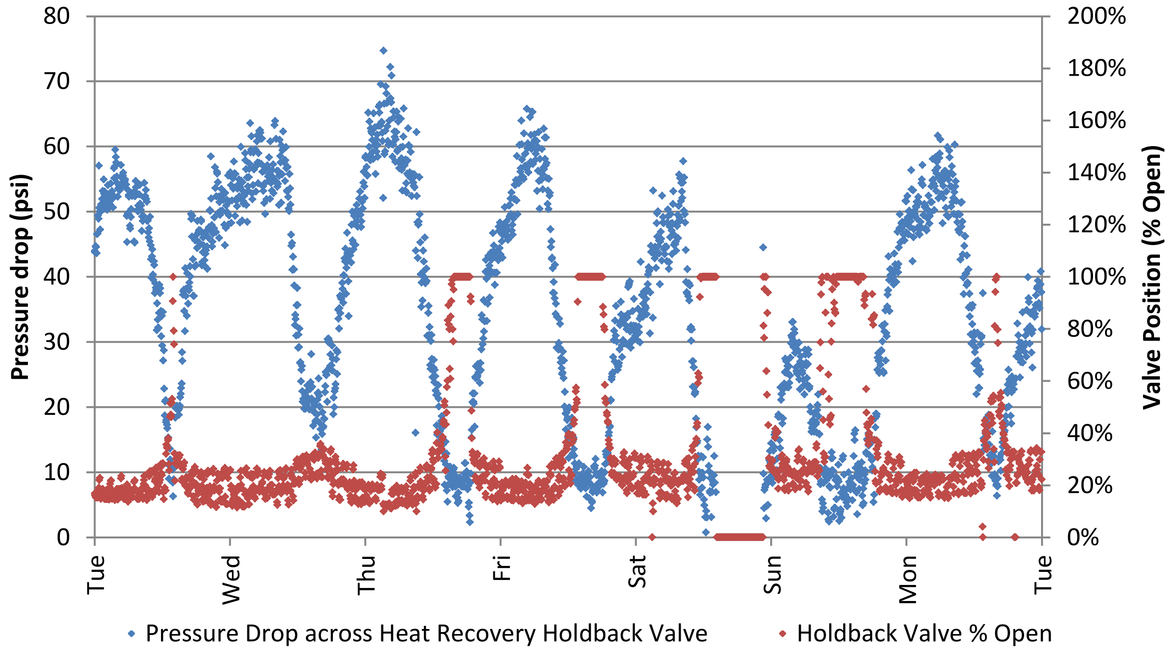


FIGURE 7: PRESSURE DROP ACROSS HOLDBACK VALVE AND VALVE POSITION FOR ONE SAMPLE WEEK

Figure 7 shows that the relationship between the valve position and the pressure drop are inversely related; the valve is only partially open when pressure drop is high, and is nearly (and at times fully) open when pressure drop is low, which matches expectations.

The quantity of heat recovered for the same test period as above is shown in Figure 8. The heat recovery coil holdback pressure and condenser dropleg pressure are shown in blue and red, respectively, and reference the left vertical axis, while the heat recovered is shown in green and references the right vertical axis.

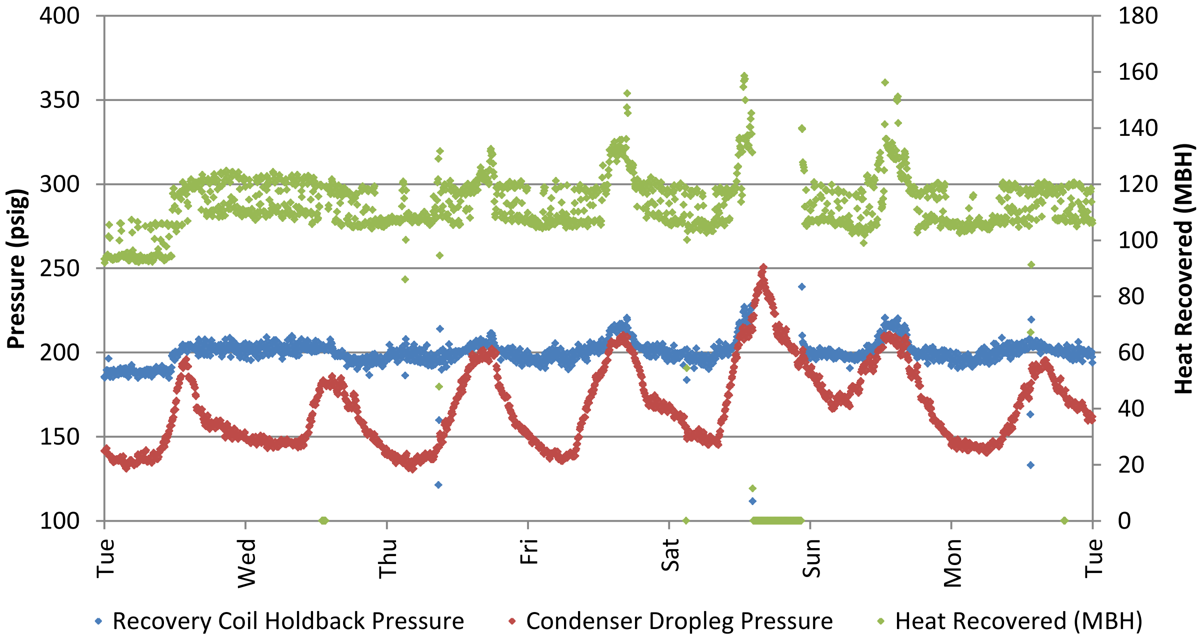


FIGURE 8: HEAT RECOVERY HOLDBACK PRESSURE, CONDENSER DROPLEG PRESSURE, AND HEAT RECOVERED FOR SAMPLE WEEK

Note that for the sample period shown, heat recovery is off during the late-Saturday/early-Sunday hours.

In general, the holdback valves facilitate the reclamation of both the sensible "superheat" from the refrigerant, and also the majority of the latent heat of rejection from the refrigerant. Without holdback, only a portion the superheat component is available for recovery, which is significantly less. However, Figure 8 shows a special case; when the saturated condensing temperature (SCT) in the refrigeration condenser is nearly equal to the heat recovery condensing temperature (HRCT), the quantity of heat recovered increases by 5-10%. These conditions occur when the outside ambient temperature is relatively warm (and therefore the SCT in the refrigeration condenser is relatively high), but there is still a heating load in the sales area. An analysis of the data from these periods shows that the increase in recovered heat can be attributed to a rise in the THR from both an increase in refrigeration load during warmer ambient mid-day temperatures, as well as an increase in heat of compression due to higher discharge pressures during these periods.

DEMAND ANALYSIS

Figure 9 below shows the demand penalty with heat recovery (e.g. the sum of all component demands with heat recovery minus the Base Case) over 24 hours for every day of the test period. The thick red line represents the overall average electric demand penalty for the entire test period.

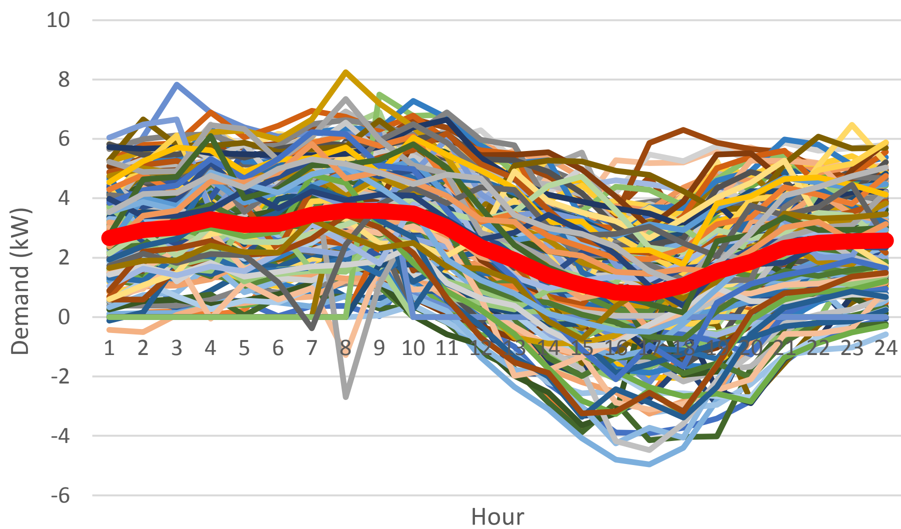


FIGURE 9: ENERGY PENALTY (HEAT RECOVERY MINUS BASE CASE) FOR 24-HOUR PERIODS

Figure 9 shows an unexpected trend; the electric demand penalty with heat recovery is less during the mid-day hours, typically from approximately noon to 6 PM. In some instances, there is an electric demand *reduction* with heat recovery (in other words, the electric energy penalty is *negative*).

To understand why this happened, the electric demand for each component was individually analyzed for an example day when the electric demand penalty was negative. Figure 10 below shows the electric demand disaggregated by component for a sample 24-hour period (February 16th).

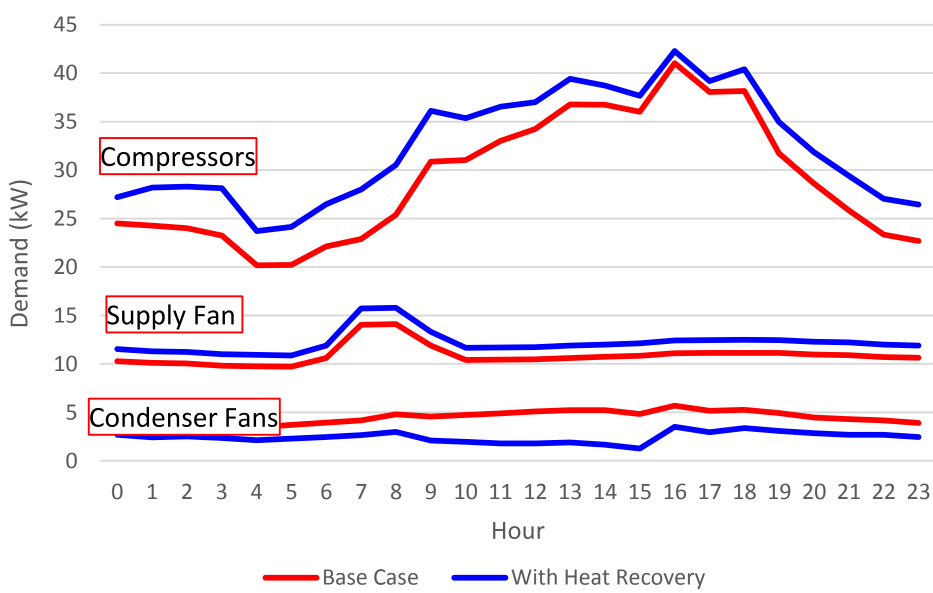


FIGURE 10: ELECTRIC DEMAND BY COMPONENT FOR SAMPLE 24-HOUR PERIOD

Figure 10 explains why the electric demand penalty is reduced (and, in the sample period shown, is actually negative) during mid-day hours. Between approximately noon and six PM, the compressor demand with heat recovery is nearly equal to the Base Case demand (e.g. near-zero penalty), and the condenser fan demand *reduction* with heat recovery is larger than the demand *penalty* from the AHU supply fan, resulting in a net reduction in electric demand with heat recovery. For the example period, the maximum demand reduction is approximately 3 kW.

During mid-day, the ambient temperature (and therefore the SCT in the refrigeration condenser) is higher, and in some cases, is nearly equal to the holdback condensing pressure in the heat recovery coil. In this situation the compressor demand with heat recovery is nearly equal to the Base Case demand since, all other factors being equal, the demand penalty with heat recovery is due to the increase in discharge pressure from the holdback valves. Concurrently, the condenser fan demand is reduced because heat recovery reduces the THR load on the condenser. The net effect is a reduction in energy penalty.

This scenario only occurs when the ambient drybulb temperature is relatively high *and* there is a need for heating capacity. During peak demand periods when energy prices are high (during the hot summer months), the heating load will be zero. Heat recovery will not reduce electric demand during peak demand periods, and static pressure penalty of the heat recovery coil in the air handling unit will actually be a slight demand penalty.

LOADS AND ENERGY USAGE VERSUS DOE2 ANALYSIS

Before the heat recovery system was designed, the supermarket used for this analysis was evaluated using DOE-2.2R energy simulation software. The DOE-2.2R energy model was calibrated using metered energy data from comparable stores from the same national chain in comparable climates. The energy model was used to predict energy cost and savings for the proposed heat recovery system, and was also used as a sizing tool during the design phase of the heat recovery system.

DOE2 has the capability to explicitly model direct-condensing heat recovery systems for space heating, including the heat recovery holdback valve, heat recovery supply line pressure losses, holdback valve pressure losses, and refrigeration compressor energy penalty. This analysis also considers the air handling unit (AHU) airside pressure penalty associated with the heat recovery coil.

For more information about the DOE-2.2R energy model, see Appendix B: DOE-2.2R Simulation.

This section compares the data collected during the test phase of the project to the DOE-2.2R energy simulation results, and includes the following sections:

- Comparison of Ambient Temperature Data
- Comparison of Natural Gas Usage
- Comparison of Refrigeration System Performance

COMPARISON OF AMBIENT TEMPERATURE DATA

Figure 11 below shows the ambient drybulb temperature (DBT) over the subject test period for both the actual test data as well as the DOE-2.2R. The DBT for the simulated case is different from the actual data because the simulation uses a typical meteorological year data file which contains aggregated weather information from many years of measured data.

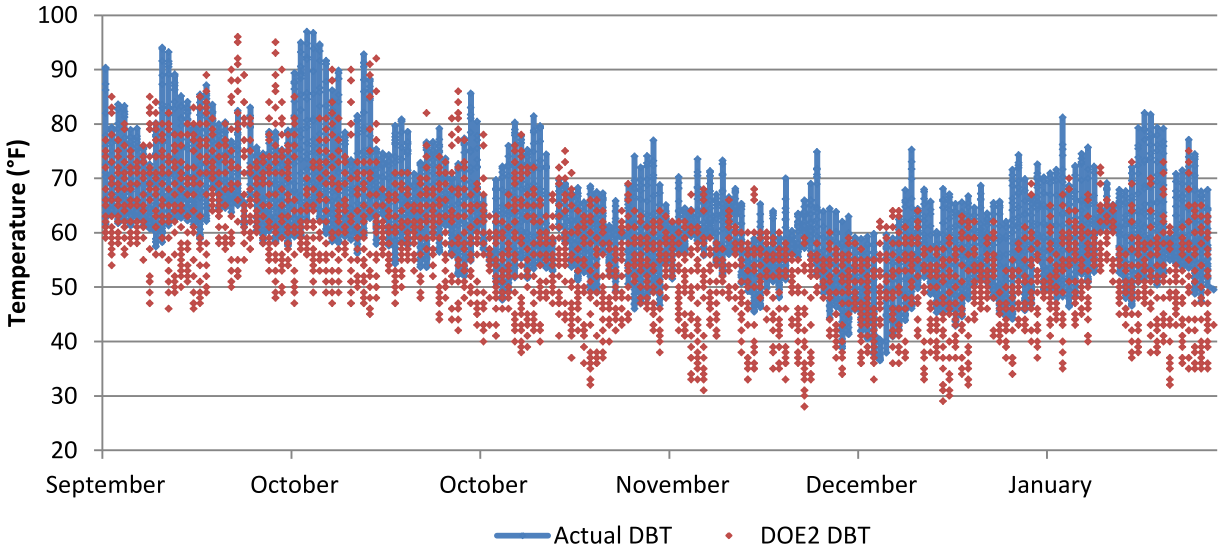
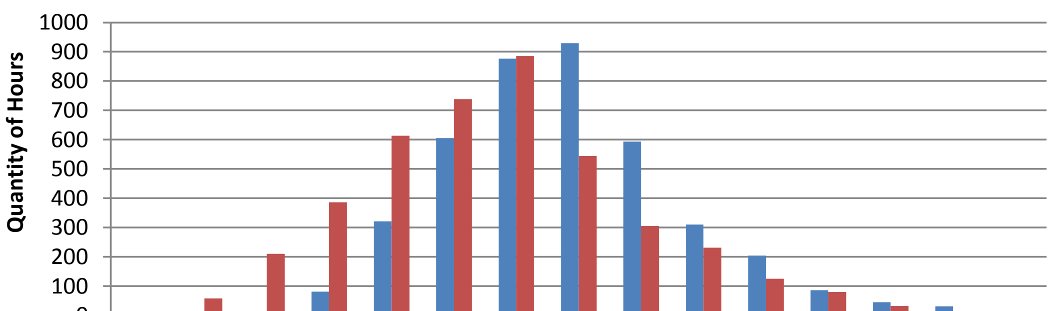


FIGURE 11: AMBIENT DRYBULB TEMPERATURE FOR SUBJECT TEST PERIOD FROM ACTUAL DATA AND DOE-2.2R ANALYSIS

Figure 11 shows that the actual DBT and the DOE-2 simulated DBT trend similarly, in general, with comparable differences in daytime and nighttime temperature swings. However, the chart shows that the DOE-2 simulated DBT is uniformly lower than the actual DBT.

Figure 12 below shows a bin-analysis of the ambient drybulb temperature for the subject test period.



	25-30	30-35	35-40	40-45	45-50	50-55	55-60	60-65	65-70	70-75	75-80	80-85	85-90	90-95	95-100
Actual DBT			14	81	321	605	876	929	593	310	204	86	45	31	2
DOE2 DBT	4	58	210	386	613	738	885	544	305	231	125	80	32	9	4

Temperature Bins (°F)

FIGURE 12: BIN ANALYSIS OF AMBIENT DRYBULB TEMPERATURE FROM TEST DATA AND DOE-2.2R SIMULATION

Figure 12 confirms that the DOE-2 simulated ambient drybulb was lower than the actual drybulb temperature. Table 6 below shows the number of hours in which the ambient drybulb temperature was below 65°F, from both the actual data as well as the DOE-2.2R simulation.

TABLE 6: HOURS WITH AMBIENT DBT BELOW 65°F: ACTUAL DATA VS. DOE-2.2R SIMULATION

Hours Below 65°F			
From Actual Data	DOE-2.2R Simulation	Difference	
2,826	3,438	612	22%

Table 6 confirms that the simulated ambient drybulb temperature in the DOE-2.2R model was cooler, on average, than the actual temperature. The actual data included 612 more hours where the ambient temperature was below 65°F, a 22% difference.

Figure 13 and Figure 14 show the air handling unit natural gas use for the 6-month subject test period for both the DOE-2.2R simulation and from the test data. Figure 13 is the Base Case usage, and Figure 14 shows the usage with heat recovery.

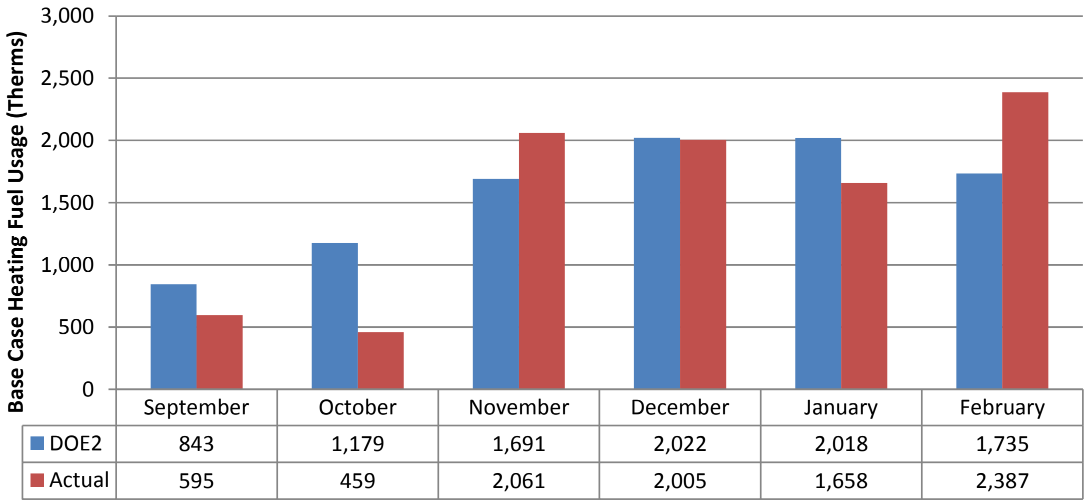
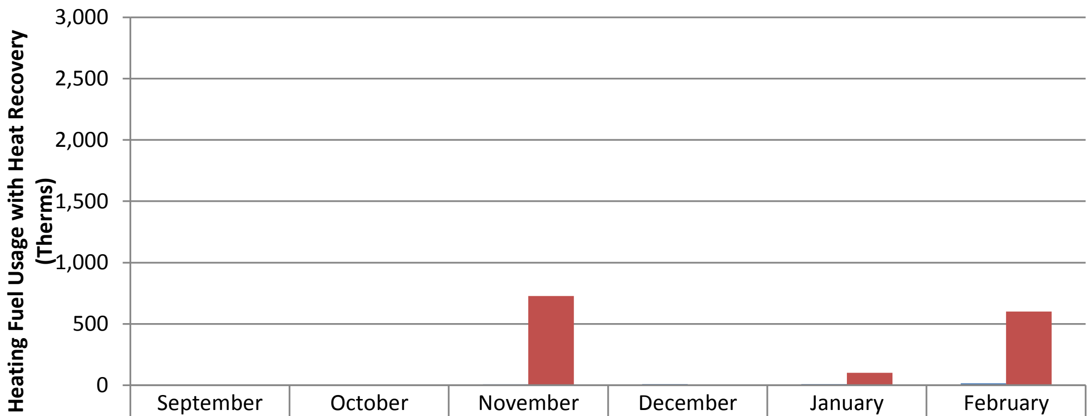


FIGURE 13: BASE CASE HEATING FUEL USAGE FOR MAIN SALES AIR HANDLING UNIT FROM DOE-2.2R SIMULATION AND OBSERVED DURING TEST PERIOD



	September	October	November	December	January	February
DOE2	0	0	7	10	8	15
Actual	0	0	728	0	101	602

Figure 14: Heating Fuel Usage with Heat Recovery for Main Sales Air Handling Unit from DOE-2.2R Simulation and Observed during Test Period

For the six month subject test period, the DOE-2.2R-simulated AHU natural gas usage was 9,488 therms without heat recovery, while the calculated usage from the test data was 9,165 therms. With heat recovery, the DOE-2.2R-simulated AHU natural gas usage was 48 therms, while the measured natural gas use from test data was 1,430 therms.

The data shows that the total calculated heating load over 6 months for the main AHU was within 4% of the DOE-2.2R energy model for the case without heat recovery. There was a larger discrepancy in the simulation comparison for the case with heat recovery, in the energy model, the AHU served nearly 100% of the heating demand with heat recovery alone. The measured performance over the 6 months illustrated that 84% of the space heating from the AHU came from heat recovery, while the remaining 16% came from the natural gas burner. This difference between model and data can be likely attributed to differences in ideal (modeled) heat recovery system performance versus actual performance.

Refrigeration System Performance

In this section, the performance of the refrigeration systems in the DOE-2.2R energy model is compared to the measured performance from test data. Topics include:

- Refrigeration Load
- Suction Group Energy and Demand
- Condenser Energy

Refrigeration Load

Table 7 and Figure 15 below show the total refrigeration load for each month of the subject test period from the DOE-2.2R energy model as well as the calculated load from the test data.

TABLE 7: REFRIGERATION LOAD FROM DOE-2.2R ENERGY MODEL AND FROM TEST DATA

	DOE2	Actual	DOE2	Actual	DOE2	Actual	DOE2	Actual	DOE2	Actual
	Total Load Served (BTU x 1 Million)									
September	94.5	104.6	35.3	61.6	25.9	35.7	25.0	23.5	94.3	92.1
October	94.1	100.1	37.6	65.1	27.4	35.6	25.4	19.7	94.8	82.5
November	87.4	88.9	37.7	64.6	27.2	33.9	24.0	14.0	89.0	73.6
December	89.4	91.4	39.3	67.1	28.3	34.9	24.7	18.1	91.2	76.7
January	89.4	92.9	39.3	65.4	28.3	35.3	24.7	18.4	91.2	71.7
February	82.2	84.4	35.3	59.2	25.5	30.7	22.4	19.2	83.4	75.2
	Peak Load (MBH)									
September	184.4	195.4	55.7	124.1	39.7	64.4	44.0	48.6	170.4	189.0
October	181.7	190.1	56.2	126.6	41.3	79.3	43.8	81.6	171.0	186.1
November	153.4	252.2	56.5	123.5	41.0	80.3	40.3	104.7	150.8	145.2
December	157.4	212.1	56.9	121.7	41.3	64.6	40.8	53.7	154.3	147.8
January	164.1	191.4	57.0	122.0	40.9	80.4	40.9	55.8	157.8	152.8
February	166.1	168.7	56.7	118.4	41.4	64.5	42.4	100.0	159.8	187.9

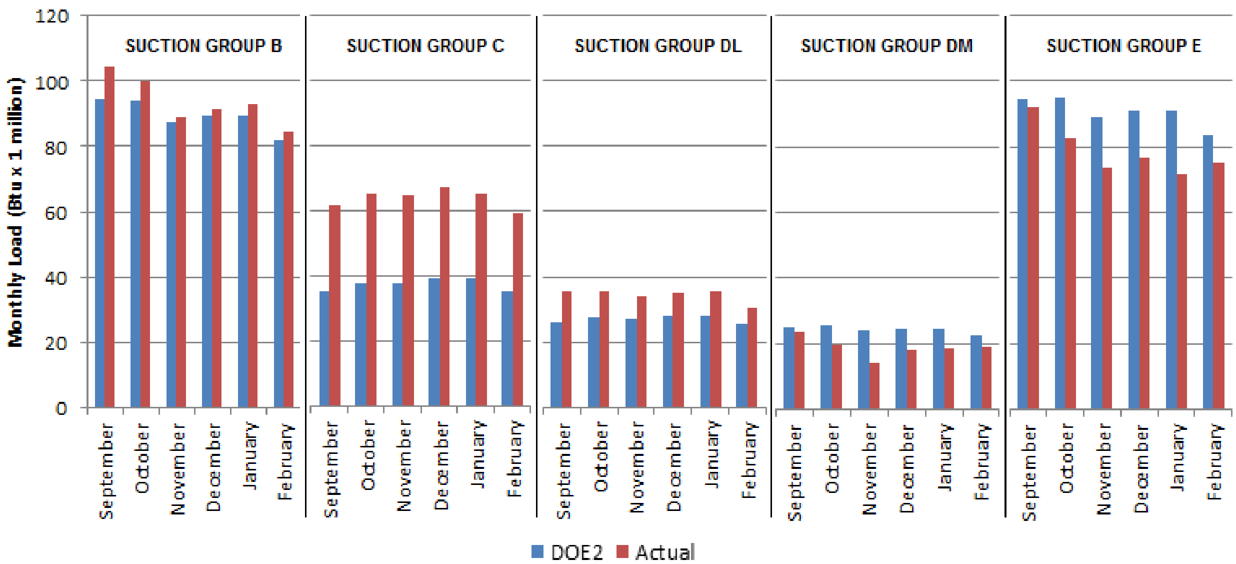
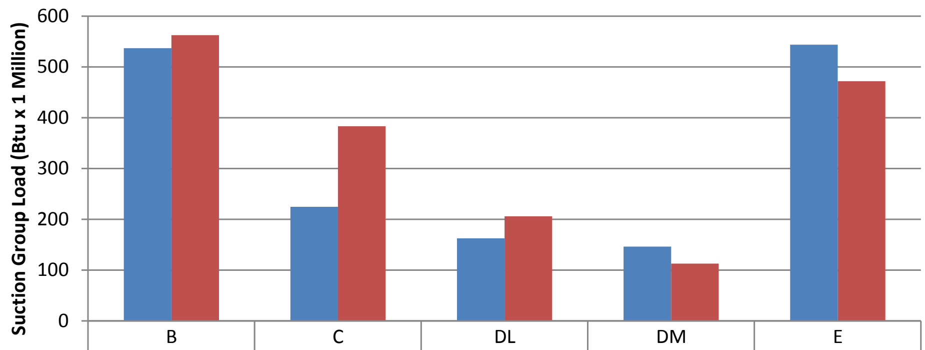


FIGURE 15: REFRIGERATION LOAD FROM DOE-2.2R ENERGY MODEL AND FROM TEST DATA

Figure 16 below aggregates the refrigeration loads in Figure 13 for the entire subject test period for each of the refrigeration system.



	B	C	DL	DM	E
DOE2	537.0	224.5	162.6	146.2	543.9
Actual	562.3	382.9	206.0	112.8	471.8

FIGURE 16: TOTAL REFRIGERATION LOAD FOR SUBJECT TEST PERIOD FROM DOE-2.2R ENERGY MODEL AND FROM TEST DATA

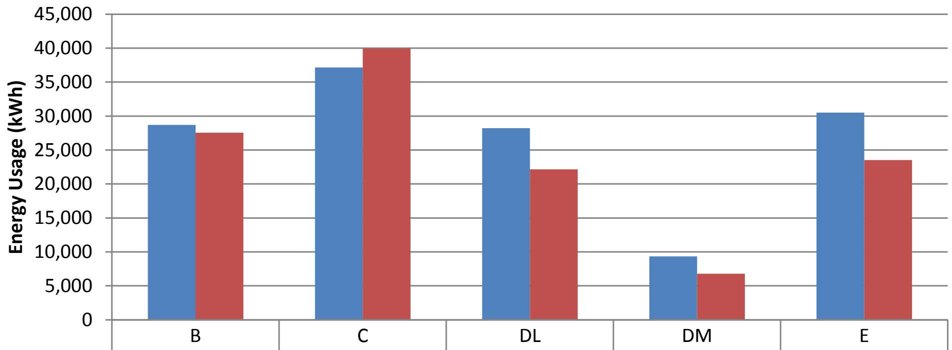
For the subject test period, the total refrigeration load served by the refrigeration systems was 1,736 million BTUs. The DOE-2.2R energy model predicted a refrigeration load of 1,614 million BTUs, within 8% of the test data.

SUCTION GROUP ELECTRIC ENERGY AND DEMAND

Table 8 and Figure 17 below show the total Base Case suction group energy usage for each month of the subject test period.

TABLE 8: BASE CASE SUCTION GROUP ELECTRIC ENERGY AND DEMAND

	B		C		DL		DM		E	
	DOE2	Actual	DOE2	Actual	DOE2	Actual	DOE2	Actual	DOE2	Actual
	Energy (kWh)									
September	5,888	5,903	6,582	7,122	5,080	4,285	1,850	1,549	6,196	5,297
October	5,361	5,424	6,546	7,375	5,005	4,124	1,790	1,265	5,689	4,542
November	4,438	4,162	6,055	6,596	4,573	3,526	1,455	841	4,748	3,493
December	4,428	4,061	6,183	6,508	4,656	3,508	1,454	1,039	4,735	3,445
January	4,451	3,997	6,181	6,265	4,654	3,473	1,451	1,015	4,730	3,155
February	4,144	3,989	5,628	6,099	4,253	3,227	1,348	1,064	4,418	3,596
	Peak (kW)									
September	16.8	14.6	12.6	14.6	9.5	8.7	5.3	3.8	17.3	11.8
October	16.4	13.8	12.0	15.0	9.3	10.5	5.0	4.5	16.6	11.3
November	9.8	14.8	10.6	14.3	8.1	9.8	3.1	4.6	10.6	9.0
December	8.6	11.8	9.6	13.8	7.3	7.7	2.8	3.0	9.1	7.9
January	8.9	9.2	9.6	12.4	7.3	9.4	2.8	3.1	8.8	7.5
February	10.2	11.1	10.4	13.1	7.7	8.6	3.2	4.9	10.4	9.7



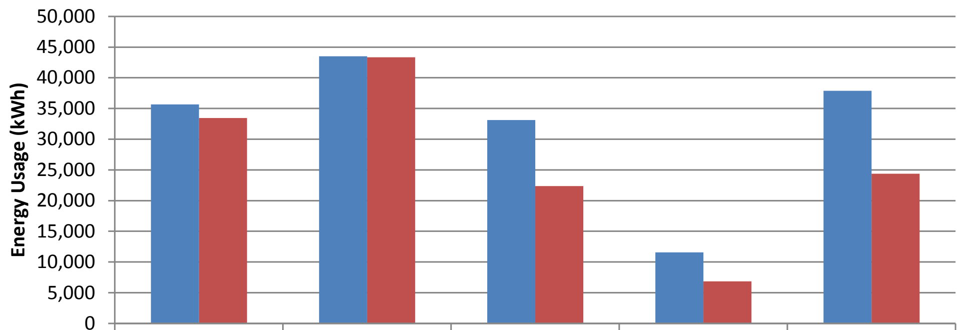
	B	C	DL	DM	E
DOE2	28,710	37,175	28,221	9,348	30,516
Actual	27,536	39,965	22,142	6,773	23,529

FIGURE 17: TOTAL BASE CASE SUCTION GROUP ENERGY FOR SUBJECT TEST PERIOD

Table 9 and Figure 18 below show the suction group energy usage with heat recovery for each month of the subject test period.

TABLE 9: SUCTION GROUP ELECTRIC ENERGY AND DEMAND WITH HEAT RECOVERY

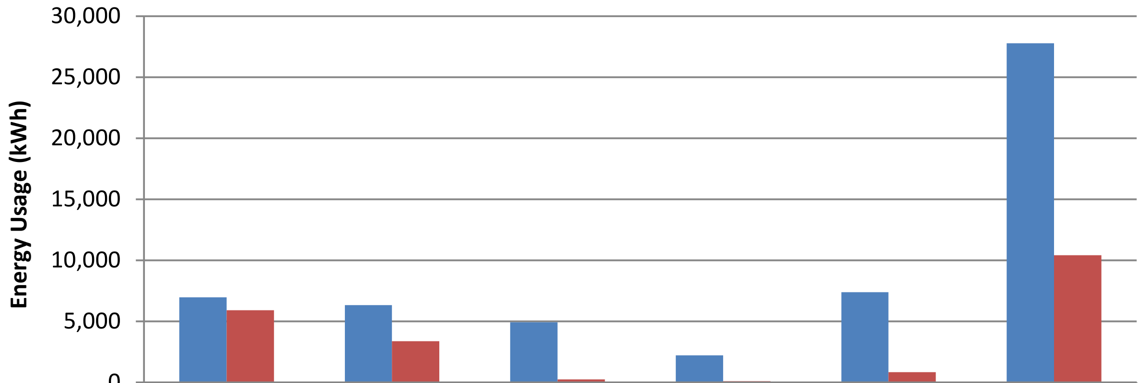
	B		C		DL		DM		E	
	DOE2	Actual	DOE2	Actual	DOE2	Actual	DOE2	Actual	DOE2	Actual
	Energy (kWh)									
September	6,439	6,329	7,068	7,328	5,472	4,295	2,027	1,554	6,768	5,312
October	6,217	5,973	7,318	7,695	5,615	4,125	1,998	1,266	6,595	4,547
November	5,719	5,364	7,223	7,400	5,477	3,562	1,874	853	6,112	3,590
December	5,909	5,603	7,549	7,438	5,703	3,582	1,940	1,065	6,313	3,755
January	5,982	4,693	7,582	6,695	5,730	3,517	1,951	1,028	6,354	3,293
February	5,397	5,482	6,768	6,772	5,134	3,298	1,756	1,095	5,753	3,856
	Peak (kW)									
September	16.7	14.6	12.6	15.7	9.6	8.7	5.3	3.8	17.3	11.8
October	16.5	13.8	12.1	15.2	9.4	10.5	5.1	4.5	16.7	11.3
November	9.9	15.9	11.1	14.6	8.5	9.8	3.1	4.7	10.7	9.0
December	10.2	13.6	11.2	14.9	8.5	7.7	3.2	3.2	10.6	9.1
January	10.5	9.4	11.2	14.1	8.5	9.4	3.1	3.1	10.8	7.8
February	10.5	11.7	11.1	14.6	8.5	8.6	3.2	5.0	10.9	10.5



	B	C	DL	DM	E
DOE2	35,663	43,508	33,131	11,546	37,895
Actual	33,443	43,328	22,379	6,861	24,353

Figure 18: Total Suction Group Energy with Heat Recovery for Subject Test Period

Figure 19 below shows the aggregated electric energy penalty from heat recovery for the DOE-2.2R energy model and from the test data.



	B	C	DL	DM	E	All Systems
DOE2	6,953	6,333	4,910	2,198	7,379	27,773
Actual	5,907	3,363	237	88	824	10,419

Figure 19: Net Electric Suction Group Energy Penalty from DOE-2.2R Energy Model and from Test Data

For the subject test period, the Base Case suction group energy usage was approximately 134,000 kWh in the energy model, and 120,000 kWh from the test data, a difference of approximately 10%. With heat recovery, the suction group energy usage was 161,700 kWh from the energy model, and 130,400 kWh from the actual test data, a difference of approximately 19%. The calculated suction group electric energy penalty with heat recovery from the energy model is 27,800 kWh, an increase of approximately 20% over the Base Case data. From the actual test data, the electric energy penalty was 10,400 kWh, an increase of 9% over the Base Case data.

Condenser Energy

Table 10 and Figure 20 below show the Base Case condenser fan energy usage with heat recovery for each month of the subject test period.

TABLE 10: BASE CASE CONDENSER FAN ENERGY FOR SUBJECT TEST PERIOD FROM DOE2 ANALYSIS AND FROM TEST DATA

	Condensers B, C		Condensers D, E	
	DOE2	Test Data	DOE2	Test Data
	Energy (kWh)			
September	1,310	1,958	1,502	1,746
October	1,231	1,967	1,410	1,648
November	919	1,649	1,052	1,253
December	817	1,641	936	1,290
January	760	1,575	872	1,205
February	818	1,558	934	1,291
	Peak (kW)			
September	2.5	4.9	2.7	4.9
October	2.5	4.9	2.8	4.9
November	2.1	3.7	2.4	2.8
December	2.1	4.9	2.4	2.6
January	2.2	3.9	2.4	2.6
February	2.2	4.6	2.5	2.8

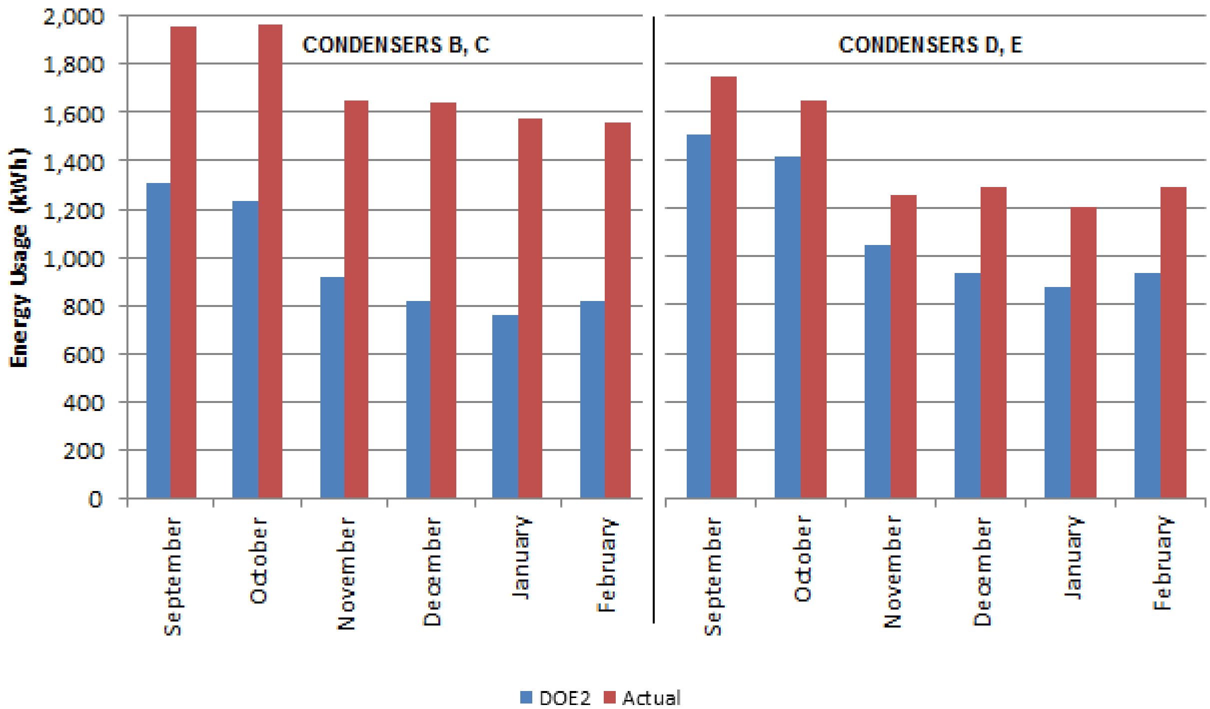


FIGURE 20: BASE CASE CONDENSER FAN ENERGY FOR SUBJECT TEST PERIOD FROM DOE2 ANALYSIS AND FROM TEST DATA

Table 11 and Figure 21 below show the Base Case condenser fan energy usage with heat recovery for each month of the subject test period.

TABLE 11: CONDENSER FAN ENERGY WITH HEAT RECOVERY FOR SUBJECT TEST PERIOD FROM DOE2 ANALYSIS AND FROM TEST DATA

	Condensers B, C		Condensers D, E	
	DOE2	Test Data	DOE2	Test Data
	Energy (kWh)			
September	954	1,567	1,363	1,733
October	790	1,733	1,201	1,648
November	432	918	749	1,160
December	308	622	595	1,173
January	283	1,280	555	1,172
February	350	639	624	1,068
	Peak (kW)			
September	2.4	4.9	2.7	4.9
October	2.5	4.9	2.8	4.9
November	1.4	3.2	2.1	2.8
December	1.2	2.4	1.8	2.6
January	1.2	3.9	1.8	2.6
February	1.4	4.6	1.9	2.8

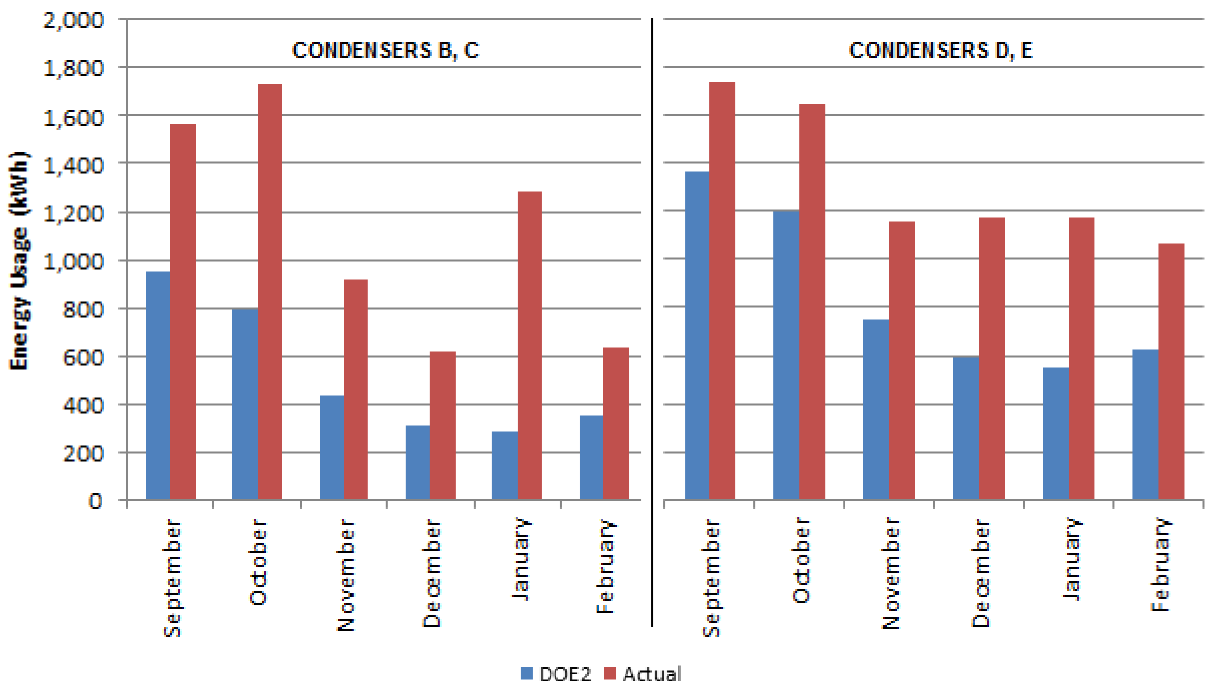


FIGURE 21: TOTAL CONDENSER FAN ENERGY WITH HEAT RECOVERY FOR SUBJECT TEST PERIOD

Figure 22 shows the difference in condenser fan energy (with heat recovery minus Base Case) for the DOE-2.2R energy model and from actual test data.

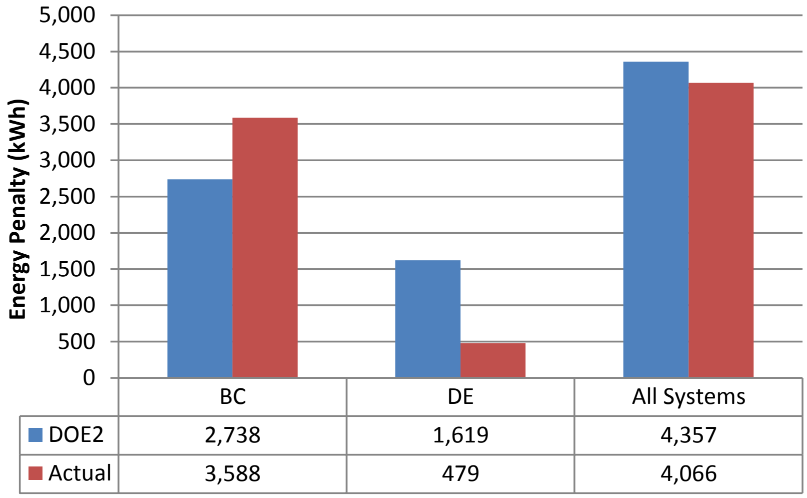


FIGURE 22: DIFFERENCE IN BASE CASE CONDENSER FAN ENERGY VERSUS WITH HEAT RECOVERY

For the subject test period, the Base Case condenser energy usage was approximately 12,600 kWh in the energy model, and 18,800 kWh from the test data. With heat recovery, the condenser energy usage was 8,200 kWh from the energy model, and 14,800 kWh from the actual test data. The calculated condenser electric energy reduction with heat recovery from the energy model is 4,400 kWh. From the actual test data, the electric energy reduction was approximately 4,100 kWh.

AHU SUPPLY FAN ENERGY

Table 12 and Figure 23 show the total air handling unit supply fan energy usage for each month of the subject test period, for both the Base Case and with heat recovery.

TABLE 12: AHU SUPPLY FAN ELECTRIC ENERGY USAGE FROM DOE2 ANALYSIS AND CALCULATED FROM ACTUAL DATA

	Base Case		With Heat Recovery	
	DOE2	Actual	DOE2	Actual
September	8,229	3,069	9,242	3,437
October	8,372	3,260	9,399	3,650
November	7,380	4,516	8,283	5,056
December	7,347	7,562	8,246	8,467
January	7,336	6,754	7,575	7,562
February	6,749	5,033	8,682	5,635
Total	45,413	30,194	51,427	33,808

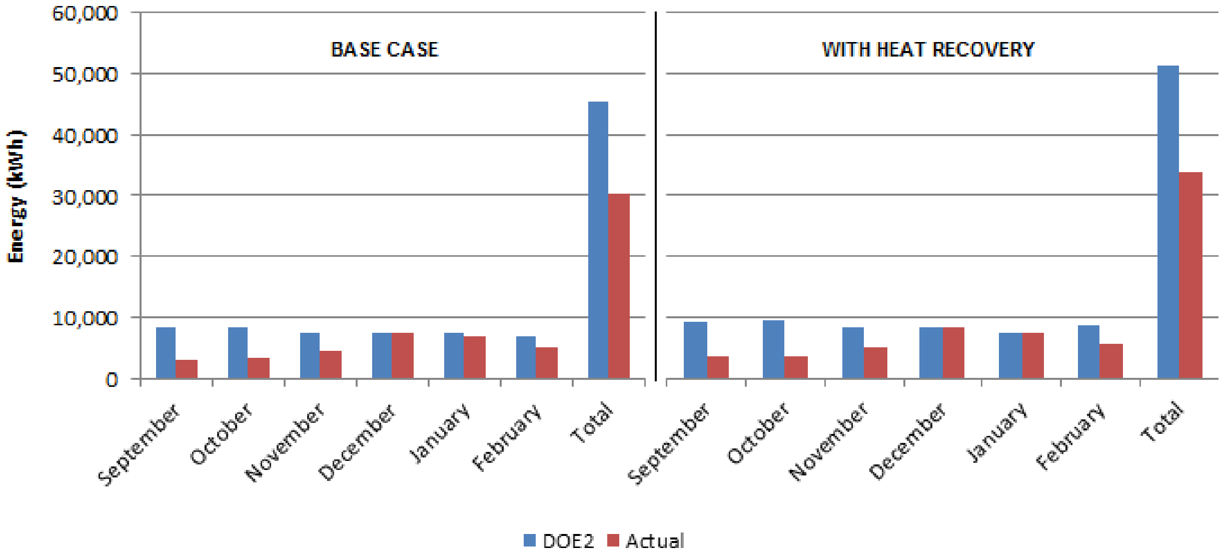


FIGURE 23: AHU SUPPLY FAN ENERGY FOR SUBJECT TEST PERIOD FROM DOE2 ANALYSIS AND CALCULATED FROM ACTUAL DATA

For the subject test period, the Base Case AHU supply fan energy usage was approximately 45,400 kWh in the energy model, and 30,200 kWh from the test data. With heat recovery, the supply fan energy usage was 51,400 kWh from the energy model, and 33,800 kWh from the actual test data. The calculated supply electric energy penalty with heat recovery from the energy model is approximately 6,000 kWh. From the actual test data, the electric energy penalty was approximately 3,600 kWh. System Performance versus Title 24 Requirements

In 2013, the California Energy Commission adopted updates to the California Building Energy Efficiency Standards which included mandatory requirements for newly-constructed supermarket (e.g. "commercial") refrigeration systems. Included in the new standards are mandatory requirements for refrigeration heat recovery systems for space heating. Presented below is Section 120.6(b)4 of California's Building Energy Efficiency Standards:

4. Refrigeration Heat Recovery.

A. HVAC systems shall utilize heat recovery from refrigeration system(s) for space heating, using no less than 25 percent of the sum of the design Total Heat of Rejection of all refrigeration systems that have individual Total Heat of Rejection values of 150,000 Btu/h or greater at design conditions.

EXCEPTION 1 to Section 120.6(b)4A: Stores located in Climate Zone 15.

EXCEPTION 2 to Section 120.6(b)4A: HVAC systems or refrigeration systems that are reused for an addition or alteration.

B. The increase in hydrofluorocarbon refrigerant charge associated with refrigeration heat recovery equipment and piping shall be no greater than 0.35 lbs per 1,000 Btu/h of heat recovery heating capacity.

The heat recovery requirement of Title 24 (Section 120.6(b)4) applies to newly-constructed supermarkets, and newly-constructed additions to existing supermarkets where additional refrigeration capacity and space heating capacity are both included in the expansion design. The requirement does not apply to retrofits.

Also included in the new standards are requirements for condenser efficiency and control. Relevant to this study is the requirement that refrigeration head pressure shall be allowed to float to 70°F SCT or less, with mandatory ambient-following (e.g. variable setpoint or drybulb reset logic) controls and variable-speed condenser fan control.

The supermarket used for this study was permitted for construction before January 1, 2014, and is therefore not required to comply with the Title-24 standards. However, the heat recovery system was scrutinized in the context of the Title 24 requirements in the following sections, which include:

- Heat Recovered versus Whole-Building THR
- Refrigerant Charge Analysis

HEAT RECOVERED VERSUS WHOLE BUILDING THR

Per Title 24 requirements, heat recovery capacity at design conditions shall be at least 25% of the total heat rejection (THR) of all the refrigeration systems in the building whose THR is higher than 150 MBH. The design documentation for the subject store states that the THR at design conditions is 1,980 MBH, and the design heat recovery coil capacity is 781 MBH—39% of the whole-store design THR. Therefore, the subject recovery system would comply with the minimum heat recovery capacity requirements in Title 24.

REFRIGERANT CHARGE ANALYSIS

Subsection B of the Title 24 heat recovery standards prohibits heat recovery designs which increase hydrofluorocarbon (HFC) refrigerant charge by more than 0.35 lbs for every MBH of heat recovery heating capacity added, at design conditions, versus a comparably-sized refrigeration system without heat recovery. The requirement is motivated by the recognition that more refrigerant charge increases refrigerant emissions to the atmosphere, with HFC refrigerants exhibiting global warming potentials that are several thousand times higher than carbon dioxide (the global warming potential of R-507A, the refrigerant in the subject test system, is almost 4,000 times higher than CO_2).

Refrigerant charge goes up due to the addition of the recovery coil itself and the additional piping between the compressors and the recovery coil. In addition, the refrigerant leaving the recovery coil and entering the refrigerant condenser is mostly condensed, which increases the charge in the outdoor condenser compared with normal operation. Table 13 below shows the calculated refrigerant charge increase for the subject heat recovery system. Note that the values shown are only for the charge *increase* related to heat recovery, not the overall system charge.

TABLE 13: CALCULATED CHARGE INCREASE FOR SUBJECT HEAT RECOVERY SYSTEM

	Condenser Charge (lbs)	Piping Charge (lbs)	Recovery Coil Charge (lbs)	Total Charge (lbs)
Base Case	332.5	NA	NA	332.5
With Heat Recovery	630.2	67.5	14.0	711.7
Increase (lbs):				379.2
Reclaim Coil Capacity (MBH):				780.9
Charge Increase/Capacity (lbs/MBH):				**0.49**

As-built, the subject heat recovery system would not comply with the Title-24 standards. Table 13 shows that the charge increase from heat recovery is approximately 380 lbs, or 0.49 lbs per MBH of heating capacity.

The table shows that the majority of the charge increase is at the refrigeration condenser, where the calculated charge increased 298 lbs—79% of the overall charge increase for the whole recovery system. The refrigeration condensers that were selected for this supermarket have low fan power, but at the cost of higher surface area (and more internal volume and more refrigerant charge). The low-power condensers appear to be a sensible choice from the perspective of energy efficiency for a refrigeration system with fan-cycling capacity control and without heat recovery (where the charge in the condenser is less of a concern, since the refrigerant in the tubes is mostly vapor without heat recovery). Condensers with lower charge and higher fan power, however, would be attractive in this application for several reasons, including:

- With heat recovery, the condensers reject less heat through the course of the year (since much of the annual heat of rejection goes to space heating), so the incremental benefit of lower-power condensers is reduced
- With variable-speed control of the condenser fans (which is also required by Title 24 and also missed on the subject condensers), the higher fan power of the lower-charge condensers is mitigated during most hours of operation
- The lower-charge condensers would likely have a lower first-cost

For reference, the refrigerant charge increase was calculated, with lower-charge condensers (26.9 lb. summer charge) used in lieu of the subject condensers (79.6 lb.) Table 14 below shows the results of this calculation.

TABLE 14: CALCULATED CHARGE INCREASE FOR SUBJECT HEAT RECOVERY SYSTEM WITH LOW-CHARGE CONDENSERS

	Condenser Charge (lbs)	Piping Charge (lbs)	Recovery Coil Charge (lbs)	Total Charge (lbs)
Base Case	112.4	NA	NA	112.4
With Heat Recovery	213.0	67.5	14.0	294.5
Increase (lbs):				182.1
Reclaim Coil Capacity (MBH):				780.9
Charge Increase/Capacity (lbs/MBH):				**0.23**

With lower-charge condensers, the subject test system could easily comply with the Title 24 requirements, with the calculated charge increasing only 0.23 lbs per MBH of heat rejection capacity—33% lower than the Title 24 requirement and 52% lower than the as-built design.

Moreover, the series-connected, direct-condensing heat recovery design analyzed here results in the largest increase in refrigerant charge relative to other recovery designs. Alternative designs include heat recovery heat exchangers that are piped in parallel with the refrigeration condensers (e.g. parallel-connected direct-condensing), and indirect heat recovery systems where heat is exchanged first with a water or water/glycol loop in a close-coupled refrigerant/water heat exchanger, and then pumped to water/air heat exchangers in one or more air handling units. Parallel-connected heat recovery systems require no additional refrigerant charge since the refrigeration condenser is not used when heat recovery is on. Indirect heat recovery designs also require little additional refrigerant, since the refrigerant-to-fluid heat exchanger can be mounted close to the compressors, reducing refrigerant charge in the piping between the compressor racks and the heat recovery heat exchanger.

For reference, the total charge for each system is shown in Table 15 below. The information was provided by the refrigeration contractor.

TABLE 15: R-507A REFRIGERANT CHARGE PER SYSTEM IN POUNDS

System	Charge (lbs)
A	350
B	575
C	600
D	575
E	550
F	450

ECONOMICS ANALYSIS

In this section, a cost estimate and economic analysis of the project is performed.

Table 16 below shows an estimation of the installation costs for the heat recovery system.

TABLE 16: HEAT RECOVERY PROJECT COST ESTIMATE

Item	Quantity	Cost per Unit ($)	Cost ($)
Materials			
4-circuit direct-condensing reclaim coil (materials and installation)	1	$4,300	$4,300
Three-Way Refrigerant Control Valves	4	$250	$1,000
Electronic Holdback Valves	4	$350	$1,400
Check Valves	4	$60	$240
Pump-Out Solenoid Valves	4	$75	$300
Additional Piping and Insulation		$3,890	$3,890
Additional Refrigerant	182 lbs.	$3.79/lb	$690
Labor			
Engineering	40 hours	$95/hr	$3,800
Piping Installation	16 hours	$60/hr	$960
Controls programming, tuning	10 hours	$115/hr	$1,150
		Total Estimated Cost:	$17,730

Table 17 below shows the calculated simple payback, net present value (NPV) and internal rate of return (IRR) for this project. The utility cost savings represent the expectation based on DOE2.2R modeling analysis, which represents the annual expectations for savings and operating cost. The NPV and IRR results are based on an assumed escalation rate of 4% and a discount rate of 8% over 15 years.

TABLE 17: SIMPLE PAYBACK, NET PRESENT VALUE, AND INTERNAL RATE OF RETURN ANALYSIS WITHOUT INCENTIVES

Annual Utility Cost Savings vs. Base Case	$9,395
Heat Recovery Additional Capital Cost	$17,730
Simple Payback	**1.9**
Net Present Value (NPV)	**$83,799**
Internal Rate of Return (IRR)	**57%**

Table 17 shows that the simple payback is less than two years without utility incentives. Table 18 below shows the calculated simple payback, NPV, and IRR with utility incentives. The incentive amount was assumed to be $1.00 per therm of natural gas savings, with a cost limit of 50% of the measure capital cost (which *is* a factor for this analysis).

TABLE 18: SIMPLE PAYBACK, NET PRESENT VALUE, AND INTERNAL RATE OF RETURN ANALYSIS WITH INCENTIVES

Annual Utility Cost Savings vs. Base Case	$9,395
Heat Recovery Additional Capital Cost	$17,730
Utility Incentive	$8,865
Additional Capital Cost with Utility Incentives	$8,865
Simple Payback	**0.9**
Net Present Value (NPV)	**$92,664**
Internal Rate of Return (IRR)	**110%**

With utility incentives, the simple payback could be less than one year.

Furthermore, the economics of heat recovery in other locations would likely be even more attractive for several reasons, including:

- The heating load in the host site's town is relatively low
  - The number of annual heating design hours at 65°F is only 1,196
  - The ambient drybulb temperature is normally mild (the average annual drybulb temperature is 60.3°F, and the 0.4% design drybulb is 88.4°F)
  - The humidity (the biggest driver of heating load) is low (0.4% design RH: 68% at 74.4°F mean coincident drybulb)
- Natural gas prices in the host site's town are relatively low
- Electric energy prices in host site's town are relatively high

CONCLUSIONS

In six months of testing, there was a measured heat recovery system offset of 7,740 therms of natural gas. During that time, the refrigeration system energy usage went up by 10,300 kWh. The Net energy savings was 738.3 Million BTUs. The net energy cost savings was $6,400. The scope of this evaluation was limited to heat recovery for just the main sales area air handling unit, although the supermarket HVAC demand is met with several other additional HVAC units, all of which are equipped with natural gas furnaces and outside air dampers. Further optimization of the HVAC design approach could yield even more savings from heat recovery, by utilizing heat recovery capacity to serve the heating load that is currently served by secondary rooftop units. In addition to this, the same system operating in different (colder) climates or with different energy rate structures will both perform differently and have different savings. The system considered in this study was operated in a relatively mild climate with relative minor heating needs. The same arrangement installed in a colder climate may demonstrate much larger savings.

Heat recovery for space heating should be adopted as an energy efficiency measure, and significant statewide energy savings are possible with widespread adoption. While California's energy efficiency standards (Title 24) require heat recovery for newly constructed commercial refrigeration (e.g. supermarket) applications, financial incentives are recommended for supermarket retrofit applications, and new-construction applications that are exempt from the Title 24 requirements but still have space heating loads and refrigeration capacity. However, substantial market support and training by the California utility companies (beyond just incentives) is needed to achieve the intended savings levels and market penetration, while balancing electric energy penalty and refrigerant charge increase. Specifically, the industry seems to have lost much of the technical understanding related to holdback valve utilization and control.

The focus of this project was a direct-condensing heat recovery system, where heat is exchanged from refrigerant directly to air. Other heat recovery configurations are viable, with comparable (or even higher) savings expectations, which would be more suitable for retrofit applications. One configuration in particular is the indirect design, where recovered heat is transferred from the refrigerant to an intermediate fluid (normally water or water-glycol) which is circulated through a fluid-to-air heat exchanger located in the air handling unit airstream.

APPENDIX A: ANALYSIS PLAN

INSTRUMENTATION

TABLE 19: SENSORS ADDED FOR THE FIELD STUDY

Sensor	Description/Location	Make/Model	Range	Output
Natural Gas Flow Meter	Flow meter on natural gas supply line to Seasons4 main AHU, downstream of the natural gas pressure regulator, on the roof next to Seasons4 unit.	Meter: American Meter Company AC630 Pulser: Elster Meter Services RVP-VI	NA	2 ft^3 (2,030 Btu) natural gas per pulse
Supply Airflow Meter	(5) Airflow probe arrays located in supply duct approximately 100 feet downstream of Seasons4 air handling unit. All probes are connected to one transmitter	Airflow probes: 4x 24" Dwyer DAFM-009 duct airflow measuring probes. Transmitter: Dwyer MS-121 Magnesense Transmitter	400 ft/min - 12,000 ft/min	4-20 mA
Recovery Coil Air Inlet and Outlet Temperature Sensors	(8) 10-foot, 100-Ohm platinum equivalent averaging temperature sensors. Two sensors per heat recovery coil circuit, one at coil air inlet and one at coil air outlet	Temperature Sensor: Johnson Controls TE-6337P-1 Transmitter: Johnson Controls TQ-6000-1 Output Transmitter for use w/100 Ohm Platinum Probes	-50°F to 220°F	4-20 mA
Recovery Coil Refrigerant Inlet Pressure Transducers	(4) Pressure transducers, one per recovery coil circuit. Pressure transducers to be installed at the access fitting upstream of the coil isolation ball valve inside the Seasons4 air handling unit	Ashcroft G27M0242M2300#G	0 psig - 300 psig	4-20 mA

The temporary sensors will be connected to a datalogger that will be temporarily installed in the condensing section of the air handling unit. The sensors will be connected to two Advantech ADAM 4017+ analog input modules, which communicate via RS-485 Modbus to an Echelon iLon 100 Internet Server. A wireless GSM modem will be used to remotely extract data from the iLon internet server. Figure 25 below shows a simplified schematic of the monitoring panel and temporary instrumentation.

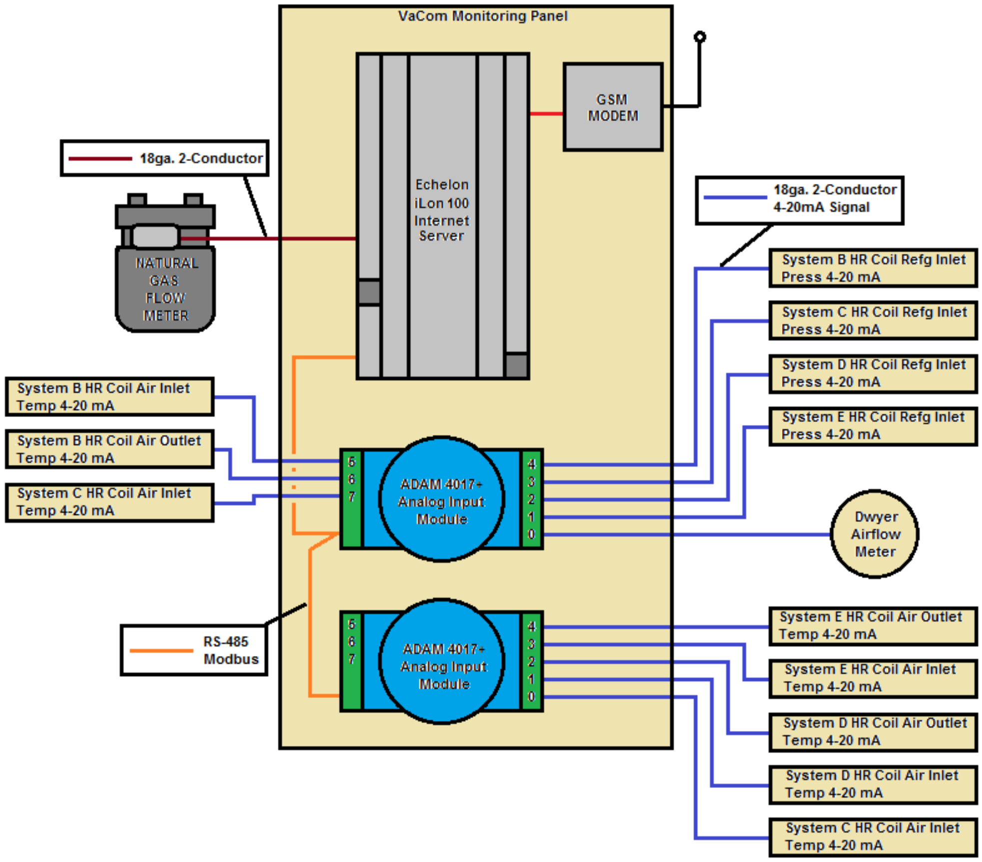


FIGURE 24: VACOM PANEL SIMPLE WIRING SCHEMATIC

The Echelon iLon internet server records the values of the connected instruments every two minutes. The ADAM input modules convert the 4-20mA signals from the sensors into unsigned 16 bit integers, ranging from 0 and 65535. The integer is a full scale percentage and calculated as follows:

$$\%mA\ Full\text{-}Scale = 100\%\ x\ \frac{Measured\ Value - Minimum\ Value}{Maximum\ Value - Minimum\ Value}$$

For the 4-20mA sensors used in this project, the full-scale percentage is:

$$\%mA\ Full\text{-}Scale = 100\%\ x\ \frac{Measured\ Value\ (mA) - 4\ mA}{20\ mA - 4\ mA}$$

The 16 bit integer is calculated as follows:

$$16\text{-}Bit\ Integer = \%mA\ Full\text{-}Scale\ x\ 65535$$

The percent of instrument full scale is calculated from the above equation as follows:

$$\%\ Instrument\ Full\text{-}Scale = \frac{16\text{-}Bit\ Integer}{65535}$$

The Engineering Value is calculated as follows:

$$Engineering\ Value\ = \%\ Instrument\ Full\text{-}Scale\ x\ Instrument\ Full\text{-}Scale\ Value$$

Example Calculation

For a reading of 6.8 mA from a 4-20mA, 0-300 psig pressure transducer, using the above equations:

ADAM 4017+ reading	$= 6.8\ mA$
%mA Full-Scale	$= 100\%\ \text{x}\ \frac{6.8\ mA - 4\ mA}{20\ mA - 4\ mA}$
	$= 17.5\%$
16-Bit Integer Recorded by iLon Server	$= 17.5\%\ x\ 65535$
	$\boxed{= 11469}$
Engineering Value	$= 17.5\%\ x\ 300psig$
	$\boxed{= 52.5\ psig}$

The natural gas flow meter will connect directly to the iLon server's meter input ports. The meter is equipped with a pulser, which will send a pulse for every 2 cubic feet of natural gas that passes through the meter.

Recorded values are stored in a Comma Separated Value (.csv) file on the iLon 100 internet server. An example of the .csv file from the iLon 100 is found below in Table 11. The far left column lists the time the data was recorded, the "UCPTpointName" identifies the device, and the "UCPTvalue" column contains the 16 bit number corresponding to the 4-20 mA signal received. This 16 bit number will be converted to the appropriate engineering units using the equations described above for analysis.

TABLE 20: .CSV FILE FORMAT

UCPTlogTime	UCPTpointName	UCPTlocation	UCPTlogSourceAddress	UCPTpointStatus	UCPTvalueDef	UCPTvalue	UCPTunit	UCPTpriority
2007-12-17T12:04:01.460-08:00	MOD_RC-01_disFlow	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:01.460-08:00	MOD_RC-02_disFlow	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:01.460-08:00	MOD_RC-09_disFlow	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:01.460-08:00	MOD_RC-10_disFlow	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:02.790-08:00	MOD_RC-01_disPress	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:02.790-08:00	MOD_RC-02_disPress	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:02.790-08:00	MOD_RC-09_disPress	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:02.790-08:00	MOD_RC-10_disPress	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:02.790-08:00	MOD_RC-01_disTemp	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:02.790-08:00	MOD_RC-02_disTemp	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:02.790-08:00	MOD_RC-09_disTemp	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:02.790-08:00	MOD_RC-10_disTemp	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:06.090-08:00	MOD_RC-01_disPress	iLON100/MOD	0	AL_NO_CONDITION		65535		255
2007-12-17T12:04:06.090-08:00	MOD_RC-02_disPress	iLON100/MOD	0	AL_NO_CONDITION		65535		255

DANFOSS EMS SYSTEM INSTRUMENTATION

Table 12 below shows the instrumentation that is connected to the Danfoss energy management system.

TABLE 21: INSTRUMENTATION CONNECTED TO DANFOSS EMS SYSTEM

Pressure	Temperature	Status
Sensors at Refrigeration Compressor Racks		
Suction Pressure (1 per Suction Group, 5 Total)	Suction Temperature	Compressor Run Proofs
Discharge Pressure (1 per Protocol Unit, 4 Total)		Compressor Alarms
Sensors at Condenser		
Dropleg Pressure (1 per Condenser)	Ambient Temperature (1 per Condenser)	Fan Run Proof Fan Speed
Sensors in Air Handling Unit		
Heat Recovery Coil Holdback Pressure (1 per Circuit, 4 Total)	Sales Area Space Temperature	Air Flow Switch
	Sales Area Space Humidity	
	Supply Air Temperature	
	Return Air Temperature (sensor to be relocated to account for outside air)	
	Heat Recovery Coil Inlet Temperature (1 per Circuit, 4 Total)	
	Heat Recovery Coil Outlet Temperature (1 per Circuit, 4 Total)	
	Heat Recovery Coil Holdback Valve Outlet Temperature (1 per Circuit, 4 Total)	

The refrigeration system controllers are Danfoss AK-SC 255 units. These units have enough memory to store history data at 1-minute intervals for over one year.

ANALYSIS PLAN

PRE-INSTALLATION INSTRUMENTATION COMMISSIONING

Each instrument will be tested at a known set of conditions to verify they are functioning properly before they are installed in the field. The data acquisition system will also be tested for proper operation. An EXTECH CMM-15 signal generator will be used to generate outputs between 4 mA and 20 mA to verify that the signal is read accurately by the ADAM-4017+ analog input module and recorded correctly in the data file written by the iLon 100 e3 internet server. The signals will also be verified with a digital multimeter. A test log .csv file will be generated with the recorded values and their corresponding time stamp. This log will be correlated to the physical test logs to validate the monitoring system from source to data log.

INSTRUMENTATION FIELD COMMISSIONING

After the refrigeration and HVAC systems are installed and the system controls have been installed, and commissioned, the temporary field instruments will be commissioned to ensure accuracy and proper operation. During the instrumentation field commissioning period, several parameters will be measured and recorded, which will be necessary to calculate net Therm savings and consequent energy penalty:

STATIC PRESSURE DROP ACROSS AIR HANDLING UNIT COMPONENTS:

The air handling unit supply fan power usage would be less in a theoretical system absent of a heat recovery coil because there would be no static pressure drop associated with the coil. The difference would be proportional to the difference in total static pressure "seen" by the supply fan in each of the scenarios.

One-time measurements of pressure drop across the AHU heat recovery coil, the furnace, the air filters, and the cooling and dehumidification coils will be taken during the instrumentation field commissioning period. The measured values will be used to calculate the total AHU static pressure as well as the theoretical total static pressure without the heat recovery coil. The measurements will be taken when the supply fan is at high- and low-speed.

AIRFLOW SYMMETRY ACROSS HEAT RECOVERY COIL CIRCUITS:

The heat recovery coil assembly consists of four separate coils in parallel, assembled in a common chassis, as shown in Figure 26.

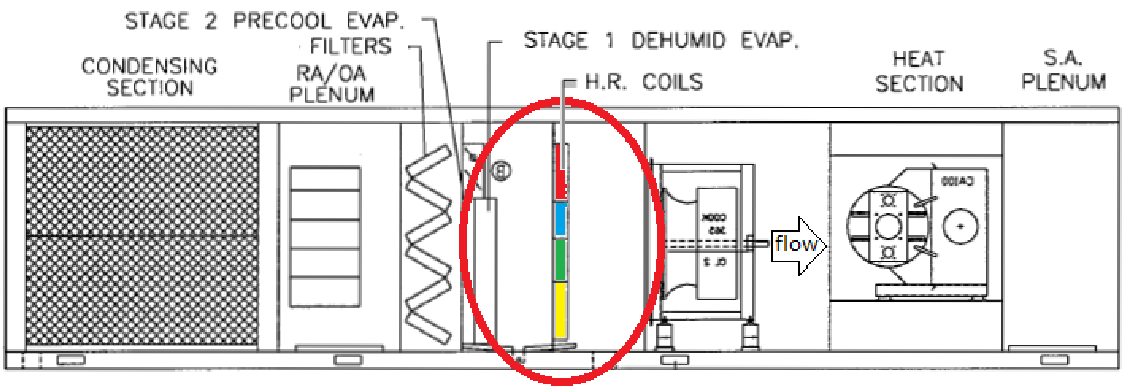


FIGURE 25: SCHEMATIC DRAWING OF AIR HANDLING UNIT, SHOWING 4-CIRCUIT HEAT RECOVERY COIL

The symmetry of airflow volume across each of the coil circuits may not be equal, for any number of reasons, including:

- Coil heights are not the same
- Upstream components such as filters, dehumidification evaporators, and bypass dampers concentrate airflow
- The location of return air ducts and outside air dampers may concentrate the direction and volume of airflow

Airflow symmetry directly upstream of the heat recovery coil will be tested using a handheld anemometer, and systemic differences in airflow across each of the heat recovery coils will be noted and accounted for in the recovered heat calculation.

AMPERAGE DRAW AND POWER FACTORS:

During the instrumentation field commissioning period, the amperage draw and power factors will be measured for the following components:

- Condenser fans
- Compressors (each unique compressor type on each suction group)

- Seasons4 air handling unit supply fan (at high- and low-speed)

DATA COLLECTION

The data measured by the added instrumentation will be transmitted via wireless modem to VaCom's EnergyDashboard (EDB) server for processing and automated analysis. In addition, operations data from the Danfoss control system will be manually downloaded weekly and sent to EDB for processing as well (Danfoss is developing a PC-based software tool called AK-EM 800 that can automatically download history files from Danfoss EMS systems. This tool may be used in lieu of manual download when it is available). The processed data and efficiency comparisons of the test and reference systems will be viewable using web access to EnergyDashboard, and will be designed to allow user selection of time intervals and levels of comparison.

MEASURING HEAT RECOVERY PERFORMANCE

The primary measurement metrics for this study are Therms of natural gas heating energy saved by heat recovery, and the additional energy consumed by the refrigeration suction groups and air handling unit supply fan as a result of heat recovery. The natural gas savings will be found by calculating the total heat recovered from the refrigeration system, and then calculating the equivalent amount of natural gas required by the furnace to add the same quantity of heat in the Base Case. The additional energy will be equal to the sum total of the required energy for the refrigeration compressors, refrigeration condensers, and air handling unit supply fan, minus the required energy for the same components in the theoretical Base Case.

ESTABLISHING BASE CASE PERFORMANCE

For this analysis, the basis of comparison for the heat recovery system is a theoretical system consisting of the same refrigeration and HVAC components, and the same ambient conditions, refrigeration loads, and heating loads as the system with heat recovery, but absent all of the components related to heat recovery. In order to calculate the performance of the Base Case system, a Base Case performance establishment period will be held where the heat recovery system is disabled for approximately 2 weeks. Data collected during this period will be used to calculate several performance metrics for the Base Case system, which are described below:

REFRIGERATION SYSTEM CONDENSER CAPACITY:

Condenser capacity is a function of the condenser fan speed as well as the temperature difference (TD) between the actual refrigerant saturated condensing temperature (SCT) and the ambient drybulb temperature. During the Base Case performance establishment period, the actual full-load condenser capacity will be determined from instrumentation data by calculating the total heat of rejection (THR)[2] of the associated suction group, and back-calculating full-speed capacity using known air-cooled condenser part-load performance curves from laboratory testing of an air-cooled condenser. The performance curves are shown in

Table 13 and Figure 27, below.

TABLE 22: CONDENSER PART-LOAD PERFORMANCE FROM LABORATORY TESTING

Speed:	100%	90%	80%	70%	60%	50%	40%	30%
Capacity:	100.0%	93.0%	86.0%	78.4%	69.9%	60.8%	50.9%	39.9%
Power:	100.0%	68.8%	48.7%	34.0%	22.7%	14.0%	7.8%	3.8%

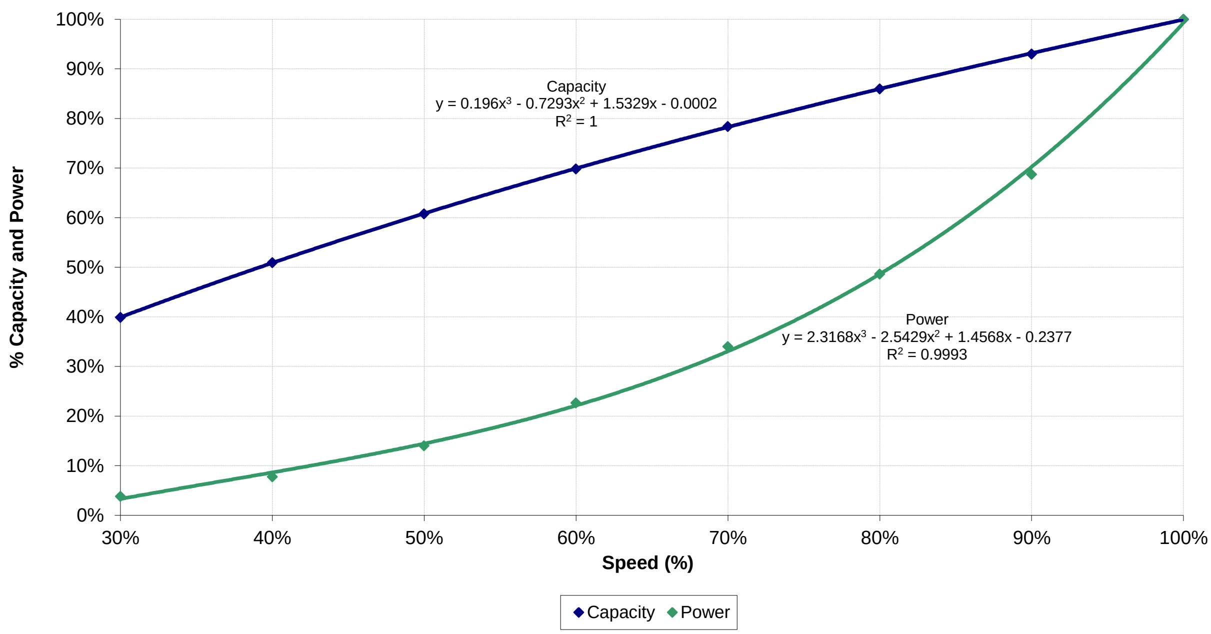


FIGURE 26: GRAPH OF CONDENSER CAPACITY AND CONDENSER POWER VERSUS FAN SPEED

The regressions will be used to predict the fan speed and fan power for the theoretical Base Case condenser. Furthermore, to the extent possible, data collected during the Base Case performance establishment period will be used to validate the laboratory results shown above as well as catalog performance data for the condensers.

[2] Total Heat of Rejection is the sum total of the refrigeration load and the heat of compression. The heat of compression is equal to the consumed compressor energy for semi-hermetic compressors.

Base Case Furnace Efficiency:

Over the course of the Base Case performance establishment period, furnace efficiency will be calculated by dividing the energy input into the airstream by the total potential heat from the natural gas stream. Energy input will be calculated from the supply air flowrate, the mixed return/outside air temperature, and the supply air temperature. Potential heat from the natural gas stream will be a known quantity from the natural gas flow meter and the thermodynamic properties of natural gas.

In the Seasons4 main air handling unit, the natural gas furnace is downstream of the heat recovery coil. In the Base Case, the furnace efficiency is expected to be higher than when heat recovery is on because the system exergy will be higher; the temperature of the air entering the furnace will be lower in the Base Case, meaning the temperature difference between the air and the furnace will be larger. The larger TD will result in more energy entering the supply airstream per unit of natural gas burned, and thus higher efficiency.

Base Case Refrigeration System Optimum Control TD

The refrigeration control temperature difference (control TD) may be optimized for each of the four subject refrigeration systems during the Base Case performance establishment period. The control TD is the temperature difference between the target refrigeration saturated condensing temperature (SCT) and the ambient drybulb temperature, and in general is optimized so that the combined total of compressor and condenser power is as low as possible. In the Base Case, the refrigeration condenser will reject all of the system total heat of rejection (THR) whereas the majority of the THR will be absent in the Heat Recovery scenario, as the heat will be used for space heating. Therefore, the optimum control TD might be different for the Base Case than when heat recovery is on.

The optimum Base Case control TD may be a moving target. There may be no opportunity to collect data when the target SCT is in the control range[3] during the first few months of operation, since the ambient drybulb temperature will likely be too low. It may be necessary to collect additional Base Case data during warm-weather conditions. Having a theoretical Base Case control TD that is different than the heat recovery scenario would only be justified if there are enough hours where the refrigeration system is in the control range and heat recovery is on at the same time to influence the economics. It is possible that the "optimum" control TD will be more influenced by peak hot-weather conditions when heat recovery is likely to be off, in which case the Base Case control TD and the heat recovery control TD will be the same.

[3] Control Range is when the target SCT for control is between the programmed maximum and minimum values, and is therefore calculated from the Control TD and ambient drybulb temperature.

CALCULATIONS SUMMARY

Figure 28 below is a flow chart of the calculation process to be taken to calculate natural gas Therm savings and associated net electric energy penalty for heat recovery.

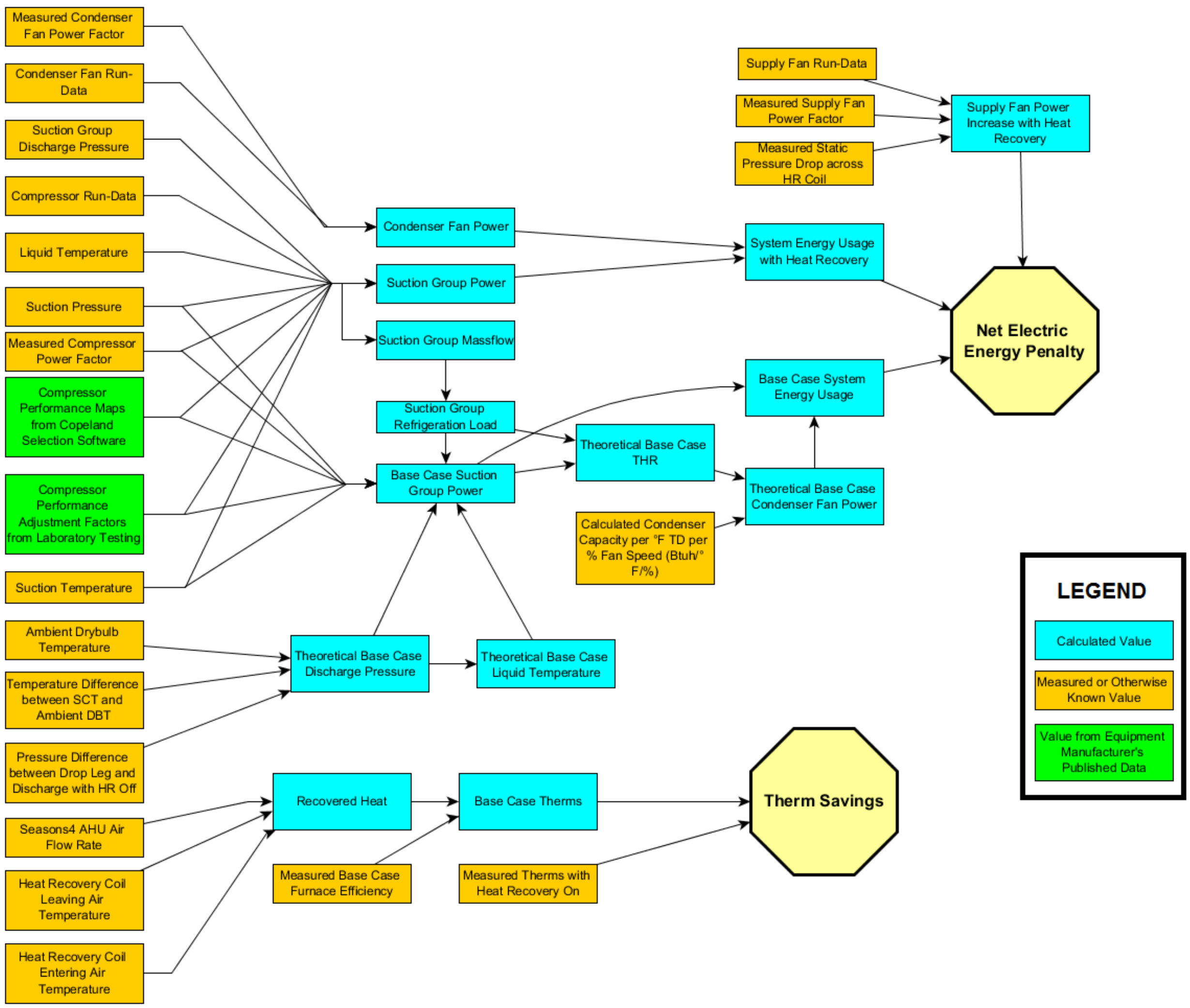


FIGURE 27: FLOWCHART OF CALCULATIONS

CALCULATIONS FOR THERM SAVINGS

RECOVERED HEAT

$$Q_{Recovered} = \sum_{All\ HR\ Circuits} 1.08\ x\ CFM\ x\ \Delta T\ x\ B.F.$$

Where:

$Q_{Recovered}$ = *Recovered heat in BTU/hour*

CFM = *Total supply airflow in ft³/min for the sample period*

ΔT = *Air temperature rise across heat recovery circuit for the sample period*

BF = *Airflow % bias factor, found by measuring airflow symmetry at the heat recovery coil during instrumentation field commissioning*

Base Case Therms

$$Base\ Case\ Therms = \frac{Q_{Recovered}}{e_{furnace}}$$

Where:

e = *Base Case furnace efficiency, calculated during Base Case performance establishment period*

Therm Savings

$$Therm\ Savings = Base\ Case\ Therms - Therms\ with\ Heat\ Recovery$$

Calculations for Net Electric Energy Penalty

Suction Group Power Calculation

$$P_{Suct\ Grp} = \sum_{All\ Comps} \left(P_{Comp\ Maps}\ x\ RF\ x\ Uld\ x\ c\right)$$

Where:

$P_{Suct\ Grp}$ = *Total suction group electric power in kW*

$P_{Comp\ Maps}$ = *Compressor power in kW from manufacturer's published performance maps*[4]

RF = *% compressor run fraction for the sample period*

Uld = *% compressor unloader fraction for sample period*[5]

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[4] Performance Map is a table of compressor massflow and power for varying saturated suction temperatures and saturated discharge temperatures. See Appendix B for sample compressor performance maps.

[5] The lead compressors of two of the suction groups have digital unloaders, which can rapidly cycle the compressor pumping capacity off and on, allowing the suction groups to more precisely match the attached load by modulating the "pulse-width" of the lead compressor pumping capacity. The compressor unloader fraction is only relevant for calculating the lead compressor power and massflow.

c = % adjustment factor, derived from comparing published compressor power data to the power measured during instrumentation field commissioning

Suction Group Massflow Calculation

$$\dot{m}_{Suct\ Grp} = \sum_{All\ Comps} \left(\dot{m}_{Comp\ Maps}\ x\ RF\ x\ Uld\ x\ c\ x\ STAF\right)$$

<u>Where:</u>

$\dot{m}_{Suct\ Grp}$ = *Total suction group refrigerant massflow in lb./Hr*

$\dot{m}_{Comp\ Maps}$ = *Compressor refrigerant massflow in lb./Hr from manufacturer's published performance maps*

RF = *% compressor run fraction for sample period*

Uld = *% compressor unloader fraction for sample period*

c = *% adjustment factor, derived from comparing published data to measured power*

$STAF$ = *Suction Temperature Adjustment Factor, a factor applied to published compressor capacity ratings, based on laboratory testing of compressor massflow at varying return gas temperatures. This parameter is variable based on the average suction temperature for the sample period*

Suction Group Load Calculation

$$Q_{Suct\ Grp} = \dot{m}_{Suct\ Grp}(h_{vapor} - h_{liquid})$$

<u>Where:</u>

$Q_{Suct\ Grp}$ = *Total suction group refrigeration load in BTU/Hr for the sample period*

$\dot{m}_{Suct\ Grp}$ = *Compressor refrigerant massflow in lb./Hr*

h_{vapor} = *Enthalpy of refrigerant vapor entering the suction group suction header, calculated from saturated suction pressure, return gas temperature, and refrigerant thermodynamic properties*

h_{liquid} = *Enthalpy of refrigerant liquid supplied to refrigeration loads, calculated from dropleg pressure, a one-time measurement of condenser liquid subcooling taken during instrumentation field commissioning, and refrigerant thermodynamic properties for Medium Temperature suction groups, and a one-time measurement of liquid subcooling and refrigerant thermodynamic properties for Low Temperature suction groups*

Theoretical Base Case Discharge Pressure

$$P_{theoretical} = \left[\text{Refrigerant saturation pressure at } [T_{ambient} + \Delta T]\right] + \Delta P_{discharge\ circuit}$$

<u>Where:</u>

$T_{ambient}$ = *Ambient drybulb temperature, measured at the condenser air inlet for each refrigeration system for the sample period*

ΔT = *Temperature difference between saturated condensing temperature (SCT, based on pressure measured at condenser dropleg) and ambient drybulb temperature for the sample period*

$\Delta P_{discharge\ circuit}$ = *Pressure drop between compressor discharge pressure and dropleg pressure, taken from data collected during the Base Case performance establishment period*

THEORETICAL BASE CASE LIQUID TEMPERATURE

For MT Systems:

$$T_{liquid,theoretical} = [\text{saturation temperature at theoretical Base Case disch. P}] - T_{amb.SC}$$

For LT Systems:

$$T_{liquid,theoretical} = Subcooler\ temperature\ setpoint$$

Where:

$T_{amb.SC}$ = *Ambient subcooling, measured during instrumentation field commissioning. Ambient subcooling is the difference between the refrigerant saturation temperature at the dropleg pressure transducer and the actual liquid temperature.*

BASE CASE SUCTION GROUP POWER CALCULATION

Base Case suction group power will be derived from the same measured suction pressure, suction temperature, and compressor maps used to calculate the suction group power in the Heat Recovery scenario, but the calculation will be based on the refrigeration load, and the theoretical Base Case discharge pressure instead of the actual discharge pressure for the sample period. The EnergyDashboard software will automatically calculate the required lead compressor unloader pulse-width percentage, and will select the mix of compressors necessary to satisfy the load based on the programmed compressor sequence for each suction group.

THEORETICAL BASE CASE TOTAL HEAT OF REJECTION (THR)

$$THR_{BaseCase} = Refg.Load + Base\ Case\ Suct.Grp\ Pwr\ x\ 3413$$

Where:

$Refg.Load$ = *Calculated refrigeration load in BTU/hour for the sample period*

$Base\ Case\ Suct.Grp\ Power$ = *Calculated theoretical Base Case suction group power for the sample period, in kW*

THEORETICAL BASE CASE CONDENSER FAN POWER

The theoretical Base Case condenser fan power will be calculated in four steps:

1. Determine the fraction of condenser capacity required to meet the theoretical Base Case Total Heat of Rejection

$$Capacity\ Fraction = \frac{THR_{BaseCase}}{Condenser\ Capacity\ at\ 100\%\ Fan\ Speed\ at\ Applied\ TD}$$

Where the Condenser Capacity at 100% Fan Speed at the Applied TD is the capacity per °F TD which was determined in the Base Case performance establishment period, multiplied by the temperature difference between the SCT and the ambient drybulb temperature

2. Determine the theoretical Base Case condenser % fan speed. The fan speed is the independent variable in the following equation, which is based on laboratory testing of an air-cooled condenser:

$$Condenser\ Capacity\ \% = 0.196x^3 - 0.7239x^2 + 1.5329x - 0.0002$$

The independent variable in a third-order polynomial can be determined with the following equation:

$$x = \sqrt[3]{\left(\frac{-b^3}{27a^3} + \frac{bc}{6a^2} - \frac{d}{2a}\right) + \sqrt{\left(\frac{-b^3}{27a^3} + \frac{bc}{6a^2} - \frac{d}{2a}\right)^2 + \left(\frac{c}{3a} - \frac{b^2}{9a^2}\right)^3}}$$

$$+ \sqrt[3]{\left(\frac{-b^3}{27a^3} + \frac{bc}{6a^2} - \frac{d}{2a}\right) - \sqrt{\left(\frac{-b^3}{27a^3} + \frac{bc}{6a^2} - \frac{d}{2a}\right)^2 + \left(\frac{c}{3a} - \frac{b^2}{9a^2}\right)^3}} - \frac{b}{3a}$$

Where, for this analysis:

a = 0.196

b = -0.7239

c = 1.5329

d = -0.0002

3. Calculate % fan power from % fan speed. % fan power is the dependent variable in the following equation, which is based on laboratory testing of an air-cooled condenser:

$$\%\ Fan\ Power = 2.317x^3 - 2.543x^2 + 1.4568x - 0.238$$

(Coefficients are irrational numbers—only four significant digits are shown, but the actual calculation will include as many significant figures as the EnergyDashboard calculation tool will allow)

4. Calculate fan power

$$P_{cond\ fan} = \%\ Fan\ Power\ x\ Measured\ Fan\ Power\ at\ 100\%\ Speed\ from\ Commissioning$$

AHU Supply Fan Power Increase due to Heat Recovery Coil

$$P_{Fan,BaseCase} = P_{Fan,HeatRecovery}\ x\ \frac{ESP_{BaseCase}}{ESP_{HeatRecovery}}$$

Where:

$P_{Fan,HeatRecovery}$ =*Supply fan power with heat recovery coil present, found by multiplying the supply fan run-fraction for the sample period by the measured supply fan power during the instrumentation field commissioning period*

$ESP_{BaseCase}$ = *The theoretical Base Case total external static pressure seen by the air handling unit supply fan, found by subtracting the measured static pressure drop across the heat recovery coil (measured during the instrumentation field commissioning period) from the total static pressure (which may be either measured during instrumentation field commissioning period, or from taken Seasons4 documentation)*

$ESP_{BaseCase}$ = *The total external static pressure seen by the air handling unit supply fan. This value will be measured during instrumentation field commissioning, or from Seasons4 documentation.*

Net Electric Energy Penalty

$$N.E.E.P = \left(P_{Supply\ Fan} + P_{Compressor} + P_{Condenser\ Fan}\right)_{Heat\ Recovery} - \left(P_{Supply\ Fan} + P_{Compressor} + P_{Condenser\ Fan}\right)_{Base\ Case}$$

APPENDIX B: DOE-2.2R SIMULATION

The supermarket used for this analysis was evaluated using DOE-2.2R energy simulation software. DOE-2.2 is a sophisticated component-based energy simulation program that can accurately model building envelope, lighting systems, HVAC systems, and refrigeration systems. The 2.2R version is specifically designed to include refrigeration systems, using refrigerant properties, mass flow and component models to model refrigeration system operation and controls system effects.

DOE2 has the capability to explicitly model direct-condensing heat recovery systems for space heating, including the heat recovery holdback valve, heat recovery supply line pressure losses, holdback valve pressure losses, and refrigeration compressor energy penalty. This analysis also considers the air handling unit (AHU) airside pressure penalty associated with the heat recovery coil.

HEAT RECOVERY SYSTEM DESCRIPTION

The heat recovery system is a direct heat recovery configuration. The heat recovery coil will be placed directly within the Seasons4 air handling unit airstream, and the discharge refrigerant vapor from the compressors will be routed through the recovery coil and then to the outdoor refrigerant condensers when in heating mode. Heat recovery condensing temperature (HRCT) will be controlled by electronic pressure regulating valves immediately downstream of the recovery coil. Three-way valves located near the Hussmann Protocol units on the roof will divert compressor discharge gas directly to the refrigeration condensers when not in heating mode.

AIR HANDLING UNIT AND HEAT RECOVERY CONTROL

Table 14 below describes the design parameters of the Seasons4 main air handling unit, while Table 15 describes the heat recovery coil sized by Seasons4 for this application.

TABLE 23: AIR HANDLING UNIT PROPERTIES

		Cooling			Furnace Heating					
Minimum Outside Air (CFM)	Total CFM	Sensible Capacity (Btuh)	Total Cooling Capacity (Btuh)	EER	Heating Input (Btuh)	Output Capacity (Btuh)	Total Static Pressure (in. WC)		Supply Fan Power (BHP)	Supply Fan Efficiency (%)
9,420	20,000	439,313	524,884	11.4	1,200,000	960,000	No HR Coil	3.67	15.9	91.7%
							With 0.49" WC HR Coil	4.16	18.0	

		Cooling			Furnace Heating					
Minimum Outside Air (CFM)	Total CFM	Sensible Capacity (Btuh)	Total Cooling Capacity (Btuh)	EER	Heating Input (Btuh)	Output Capacity (Btuh)	Total Static Pressure (in. WC)		Supply Fan Power (BHP)	Supply Fan Efficiency (%)
9,420	20,000	439,313	524,884	11.4	1,200,000	960,000	No HR Coil	3.67	15.9	91.7%
							With 0.49" WC HR Coil	4.16	18.0	

TABLE 24: HEAT RECOVERY COIL PARAMETERS

Rows	4

Fins per Inch	10
Face Area	34.2 Sq. Ft.
Airflow	20,000 CFM
Airside Pressure Drop	0.49" WC
Refrigerant Pressure Drop in Recovery Coil	2.14 psi
Design Entering Air Temperature	57.3°F
Design Heat Recovery Condensing Temperature (HRCT)	100°F

The capacity of each circuit of the heat recovery coil is described in Table 3 below.

TABLE 25: HEAT RECOVERY COIL CAPACITY

Circuit	**Capacity (MBH)**
B	230.4
C	147.4
D	172.8
E	230.4
Total Capacity:	**781.0**

SUPPLY FAN CONTROL

The Seasons4 air handling unit is equipped with an ABB variable speed drive, capable of running the supply fan at full speed, and at 65% speed. Based on the standard Seasons4 control strategy, the fan runs at reduced speed any time the air handling unit is not in heating, cooling, or dehumidification mode. This control strategy was emulated in the DOE2 simulation by analyzing heating, cooling, and dehumidification load for every hour of the year, and generating a minimum flow schedule which approximates the actual control strategy. Fan power varies with approximately the cube of the reduction in fan speed, based on affinity laws. For modeling purposes a 2.7 exponent (slightly less than the theoretical cube relationship) was used to determine power at reduced speed. The power at 65% is 31% of full speed power. Figure 29 below shows a typical fan power profile for the summer and winter.

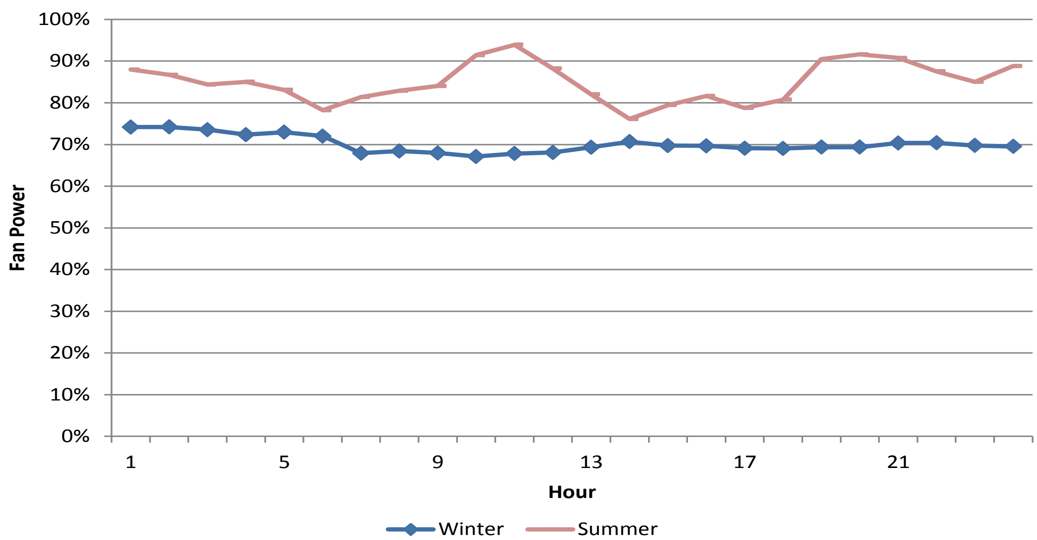


FIGURE 28: TYPICAL DAILY FAN POWER PROFILES FOR THE SEASONS4 AIR HANDLING UNIT SUPPLY FAN

The analysis model incorporated the current Seasons4 fan control logic, which only reduces fan speed in the "deadband" when there is no heating or cooling. This strategy could be improved, and in fact, may not be compliant with the intent of the new Title 24 fan speed requirements for packaged HVAC units. A more advanced strategy with speed control based on the demand for heating and cooling may provide considerable additional fan savings and reduced pressure drop penalty at the heat recovery coil (likely enough to completely offset the heat recovery electric energy penalty) as well as providing improved latent cooling performance. This opportunity should be considered for additional savings and as a possibly component of future heat recovery integration, but in order to reduce the number of study variables, only implemented after the benefit of heat recovery by itself has been determined.

HOLDBACK VALVE CONTROL

Sporlan model CDS electronic pressure regulating valves will be used for the heat recovery holdback valves, and will be controlled to a recovery condensing pressure setpoint. The control parameter will be the heat recovery condensing temperature (HRCT) setpoint. The target HRCT in the heat recovery coil circuit will be set at a value determined by adding the actual mixed return/outside air temperature in the Seasons4 unit and a preset temperature difference value (assumed to be 20°F for this analysis).

This variable HRCT setpoint will also be bounded by a maximum HRCT setpoint of 95°F and a minimum HRCT setpoint, which will be determined from the operating saturated condensing temperature (SCT) at the refrigeration condenser plus a preset temperature difference, e.g. 10°F. The latter value will insure there will always be pressure drop across the heat recovery holdback valve and resulting flash gas to maintain a reasonable velocity in the heat recovery piping and condenser; which serves the purpose of avoiding excessive charge in the condenser and piping during heat recovery.

The DOE2 TEMP-RESET scheduling capability was utilized to simulate the proposed electronic holdback valve control. The TEMP-RESET schedule correlates a temperature control parameter (HRCT setpoint in this case) to entering air temperature using a linear relationship, with user-selectable maximums and minimums. Figure 30 below shows the average HRCT setpoint for each month.

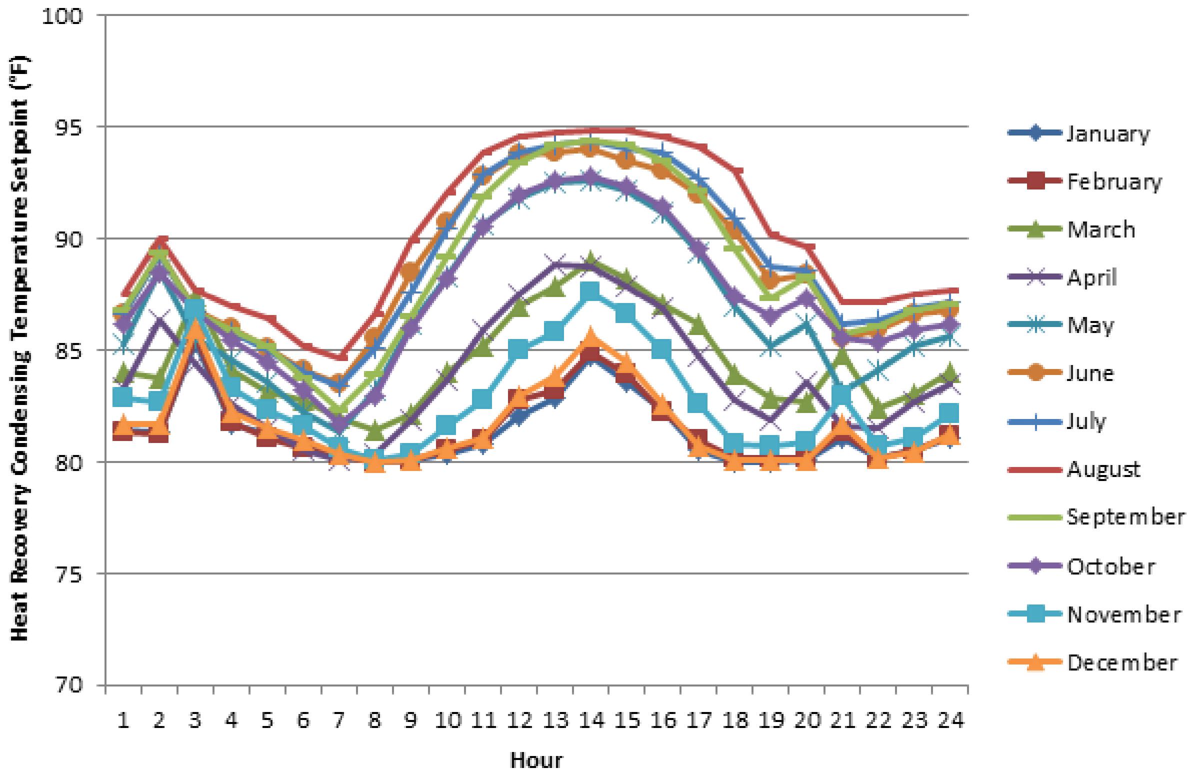


FIGURE 29: AVERAGE HOURLY HEAT RECOVERY CONDENSING TEMPERATURE (HRCT) FOR EACH MONTH

OUTSIDE AIR

The analysis assumes that the Seasons4 air handling unit modulates the outside air damper based on CO2 concentration in the return air duct, with the minimum ventilation set to handle all the outside makeup air required to replace the ventilation air from all the exhaust hoods and vents in the building.

ENERGY COSTS

PG&E's E-19V Medium General Demand-Meter Time-Of-Use energy tariff was used to calculate electric energy costs for this analysis. The tariff is summarized in Table 9, below.

TABLE 26: PG&E E-19V ENERGY TARIFF

	Energy	Demand	Period of Applicability	
			Date	Time
Summer Peak	$0.15746/kWh	$16.13/kW	May 1 - October 31	12:00PM - 6:00PM
Summer Mid-Peak	$0.0961/kWh	$3.75/kW		8:30AM - 12:00PM, 6:00PM - 9:30PM
Summer Off-Peak	$0.08223/kWh	$0.00/kW		9:30PM - 8:30AM
Winter Mid-	$0.10181/kWh	$0.21/kW	November 1 -	8:30AM - 9:30PM

Peak			April 30	
Winter Off-Peak	$0.08203/kWh	$0.00/kW		9:30PM - 8:30AM
	Energy	Demand	Period of Applicability	
			Date	Time
Summer Peak	$0.15746/kWh	$16.13/kW	May 1 - October 31	12:00PM - 6:00PM
Summer Mid-Peak	$0.0961/kWh	$3.75/kW		8:30AM - 12:00PM, 6:00PM - 9:30PM
Summer Off-Peak	$0.08223/kWh	$0.00/kW		9:30PM - 8:30AM
Winter Mid-Peak	$0.10181/kWh	$0.21/kW	November 1 - April 30	8:30AM - 9:30PM
Winter Off-Peak	$0.08203/kWh	$0.00/kW		9:30PM - 8:30AM

Natural gas cost was assumed to be $0.60/Therm.

RESULTS

The DOE2.2R simulation results are shown in Table 18 below:

TABLE 27: DOE2 SIMULATION RESULTS

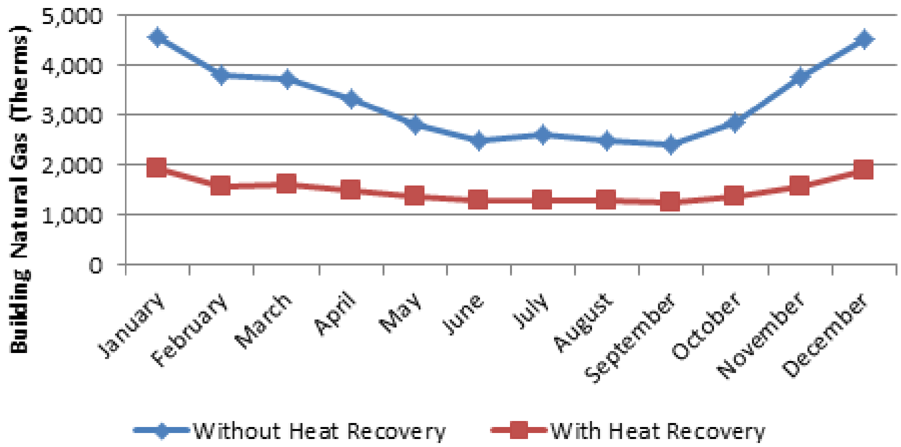
	Energy Usage				Energy Cost			
	Energy (kWh)	Natural Gas (Therms)	Total Energy (Mbtu)	Peak Demand (kW)	Electric ($)	Demand ($)	Natural Gas ($)	Total ($)
Without Heat Recovery	1,547,243	39,394	9,220	289.5	$148,888	$42,061	$23,636	$214,585
With Heat Recovery	1,588,718	18,020	7,224	286.6	$152,465	$41,912	$10,812	$205,190
Savings	**(-41,476)**	**21,374**	**1,996**	**2.8**	**$ (-3,577)**	**$149**	**$12,824**	**$9,395**
Savings (%)	**(-2.7%)**	**54.3%**	**21.6%**	**1.0%**	**(-2.4%)**	**0.4%**	**54.3%**	**4.4%**

The analysis shows that the heat recovery system saves approximately 21,400 Therms of natural gas per year, with an associated electric energy increase, related to refrigeration system and HVAC fan penalties, of approximately 41,500 kWh. The net annual total energy savings with heat recovery is 2.0 million BTU. The electric energy cost increase with heat recovery is $3,400, while the natural gas cost savings is $12,800 resulting in a net energy cost savings of approximately $9,400.

Since the DOE2.2R program is an hourly model, there are limitations in how well the program can address certain aspects of heat recovery, such as the holdback valve operation when heating is not required during the entire hour, and the operation of the variable setpoint heat recovery holdback valve. Monitoring of the actual system operation will help improve modeling assumptions and may lead to improvements in simulation tool features.

NATURAL GAS USAGE

Figure 31 below shows the monthly natural gas usage with and without heat recovery. The figures show whole building natural gas usage, including cooking equipment, space heating (from the Seasons4 air handling unit as well as other units), and domestic hot water heating.

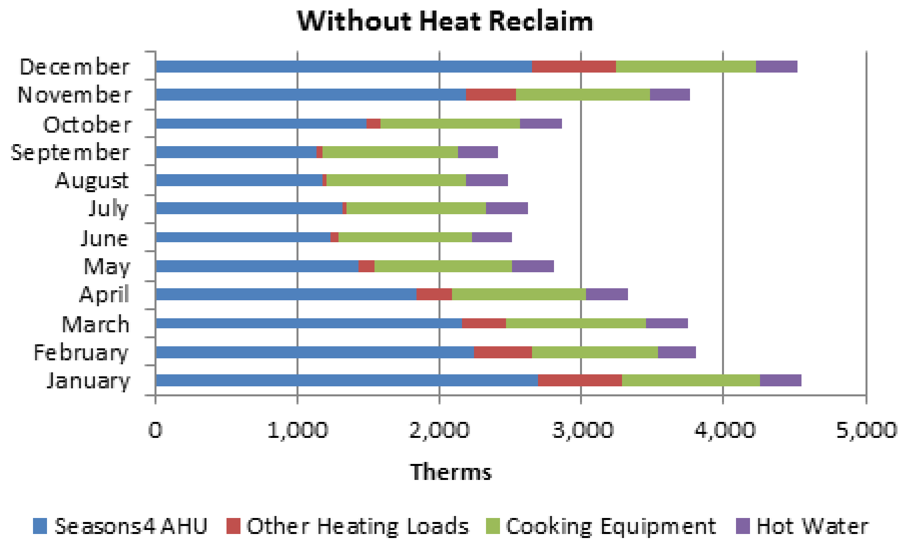


	Without Heat Recovery (Therms)	With Heat Recovery (Therms)	Difference (Therms)
January	4,554	1,917	2,637
February	3,802	1,585	2,217
March	3,743	1,599	2,144
April	3,322	1,499	1,832
May	2,806	1,374	1,432
June	2,514	1,287	1,227
July	2,621	1,311	1,310
August	2,842	1,304	1,178
September	2,413	1,275	1,138
October	2,858	1,374	1,484
November	3,763	1,590	2,173
December	4,518	1,906	2,612
TOTAL	39,394	18,020	21,374

FIGURE 30: NATURAL GAS USAGE HISTORY BY MONTH, WITH AND WITHOUT HEAT RECOVERY

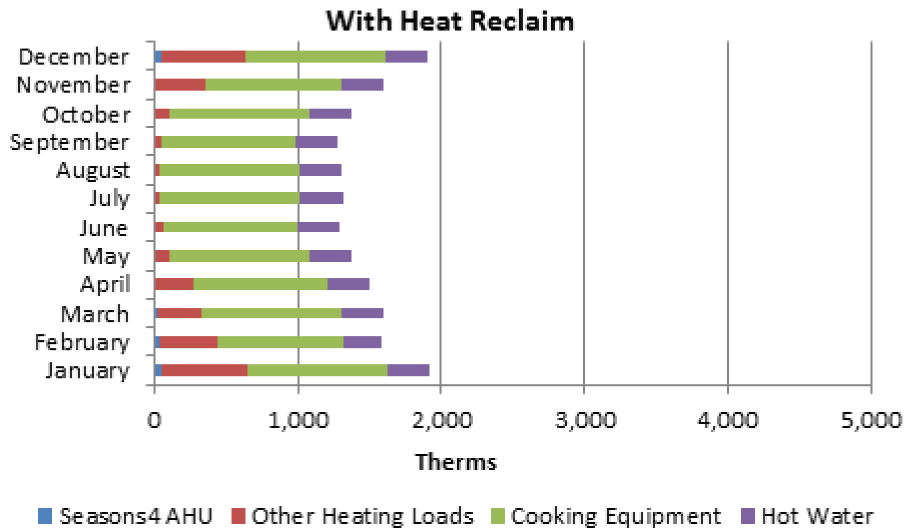
The figure shows that natural gas usage is substantially reduced, particularly in winter months. Gas usage with heat recovery is generally uniform throughout the year, with only slightly higher usage in colder winter months.

Figure 32 and Figure 33 show the natural gas consumption by each of the constituent natural gas end-uses that were considered for this analysis. Usage without heat recovery is shown in Figure 32 while usage with heat recovery is shown in Figure 33.



	Seasons4 AHU (Therms)	Other Heating Loads (Therms)	Cooking Equipment (Therms)	Hot Water (Therms)	Total (Therms)
January	2,691	593	975	296	4,555
February	2,248	406	880	267	3,801
March	2,159	314	975	296	3,744
April	1,833	259	943	286	3,321
May	1,432	104	975	296	2,807
June	1,231	54	943	286	2,514
July	1,315	36	975	296	2,622
August	1,180	31	975	296	2,482
September	1,138	46	943	286	2,413
October	1,485	103	975	296	2,859
November	2,186	347	943	286	3,762
December	2,658	589	975	296	4,518
TOTAL	**21,557**	**2,881**	**11,477**	**3,483**	**39,394**

FIGURE 31: BREAKDOWN OF NATURAL GAS END-USE BY MONTH WITHOUT HEAT RECOVERY



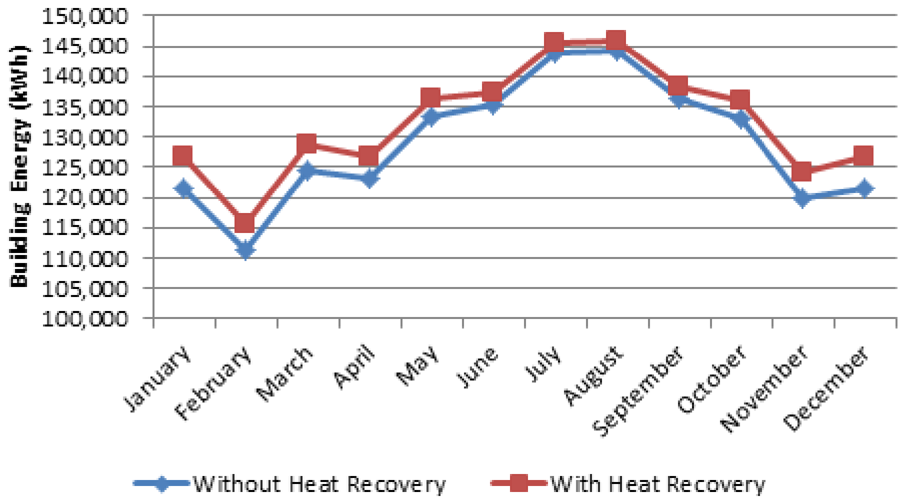
	Seasons4 AHU (Therms)	Other Heating Loads (Therms)	Cooking Equipment (Therms)	Hot Water (Therms)	Total (Therms)
January	54	593	975	296	1,918
February	31	407	880	267	1,585
March	15	314	975	296	1,600
April	10	259	943	286	1,498
May	0	104	975	296	1,375
June	4	54	943	286	1,287
July	4	36	975	296	1,311
August	3	31	975	296	1,305
September	0	45	943	286	1,274
October	1	102	975	296	1,374
November	13	348	943	286	1,590
December	47	590	975	296	1,907
TOTAL	183	2,881	11,477	3,.483	18,020

FIGURE 32: BREAKDOWN OF NATURAL GAS END-USE BY MONTH WITH HEAT RECOVERY

The figures show that natural gas usage by the Seasons4 air handling unit is nearly completely offset by heat recovery. Other natural gas end-uses are unaffected

ELECTRICAL ENERGY USAGE

Figure 34 shows the monthly whole-building electric energy usage, with and without heat recovery.



	Without Heat Recovery (kWh)	With Heat Recovery (kWh)	Difference (kWh)
January	121,434	126,892	5,458
February	111,286	115,634	4,348
March	124,380	128,774	4,394
April	123,059	126,809	3,750
May	133,272	136,387	3,115
June	135,165	137,329	2,164
July	143,753	145,644	1,891
August	144,155	145,868	1,713
September	136,384	138,285	1,901
October	133,061	136,099	3,038
November	119,755	124,244	4,489
December	121,539	126,751	5,212
TOTAL	1,547,243	1,588,716	41,473

FIGURE 33: ELECTRIC ENERGY USAGE BY MONTH, WITH AND WITHOUT HEAT RECOVERY

The figure shows that electric energy consumption is higher with heat recovery in the colder winter months, when heating demand is higher. Electric energy increase is primarily from increased compressor energy (resulting from higher average compressor discharge pressures), but is also from higher supply fan power in the Seasons4 air handling unit due to increased airside pressure penalty experienced by the Seasons4 supply fan.

Table 19 below shows the whole-building peak demand by month, with and without heat recovery.

TABLE 28: WHOLE-BUILDING PEAK DEMAND BY MONTH, WITH AND WITHOUT HEAT RECOVERY

	Without Heat Reclaim	With Heat Reclaim	Difference
January	205.8	208.3	-2.5
February	227.7	225.0	2.7
March	224.6	224.3	0.2
April	238.1	237.8	0.3
May	278.0	276.6	1.4
June	274.7	272.6	2.1
July	289.5	286.6	2.8
August	276.6	273.9	2.7
September	262.5	263.3	-0.7
October	286.5	284.2	2.4
November	217.7	217.5	0.2
December	207.5	209.9	-2.5

The table shows that heat recovery actually slightly reduces the building peak demand, despite the overall increase in energy, for all but the coldest months of the year. The compressor and AHU supply fan demand increase is offset by a decrease in refrigeration condenser fan power.

CONDENSER EFFICIENCY CONSIDERATIONS

The air cooled condensers are very efficient; with large coil areas and low fan power, on the order of half the fan power or double the efficiency of condensers used by many supermarket chains. As a result, the reduced condenser operation, due to condenser heat being instead rejected by the heat recovery coil, provides less savings than it would with more common condensers.

These very efficiency condensers are made less cost-effective if much of the annual heat rejection goes to heat recovery. There is intriguing potential to optimize the air-cooled condenser selections, using condensers with less face area and somewhat higher power motors, thereby reducing capital cost as well as reducing refrigerant charge, to also help offset the cost of heat recovery. This was discussed with the supermarket operator but it was determined that the value of this study would be highest with fewer variables, so the standard condenser selections were used. Moreover, not enough is yet known to reach a conclusion on the optimum selection criteria.

MODEL CALIBRATION AND VALIDATION

FIELD TESTING OF TECHNOLOGY

The simulated refrigeration load factor profile (hourly load shape as a fraction of design load) for each Protocol unit was generated using history of hourly refrigeration loads at two other supermarkets, one in San Diego, the other in Encinitas. Both stores have a comparable diversity of walk-in boxes and display case lineups as the store in Santa Clara County, and both stores are relatively new. Both stores also feature many of the refrigeration energy-saving components and strategies utilized in the Santa Clara store, such as "Zero-Watt" reach-in display case anti-sweat heaters, LED lighting for low-temperature reach-in display cases, and automatic display case lighting control. Refrigeration loads were determined by observing compressor operating parameters (saturated suction temperature, saturated discharge temperature, number of compressors running, etc.), and deriving the refrigeration load using the compressor manufacturer's published performance data. The load profiles were generated by averaging the refrigeration load for each hour of the day across the entire monitoring period. Refrigeration loads were monitored for approximately two weeks.

Table 29 below shows the suction groups from the San Diego and Encinitas stores that were used to generate the load profiles for each of the location's suction groups. Only suction groups that are used for heat recovery are shown. Note that the design refrigeration loads were considered for this analysis, not the design capacity.

TABLE 29: REFRIGERATION LOADS FROM STORES THAT WERE MONITORED

Sys. Designation:	B	C	DL	DM	E
Design Load:	260.57 MBH	119.36 MBH	84.41 MBH	76.91 MBH	237.74 MBH
Santa Clara County	BAKERY RETARDER	BAKERY FREEZER	DUAL TEMP	MEAT COOLER	LUNCH MEAT
	BEVERAGE	FROZEN FOODS	DUAL TEMP	MELON TABLE	M/D PRODUCE
	BEVERAGE	FROZEN FOODS	DUAL TEMP	MEAT	MEAT
	DAIRY	FROZEN FOODS	DUAL TEMP	SS FISH	MEAT PREP
	DAIRY	FROZEN FOODS	DUAL TEMP	SERVICE FISH	MEAT PREP
	DAIRY	FROZEN FOODS	DUAL TEMP	SERVICE MEAT	PROD. CLR
	DAIRY COOLER 14DRS	ICE CREAM CAKE	FROZEN FOODS	HOLDING BOX	
	M/D DELI	ICE CREAM	FZN SEAFOOD		
		ICE CREAM	GROCERY FZR		
		ICE CREAM			
Sys. Designation:	BR	C			E
Design Load:	236.18 MBH	152.10 MBH			304.36 MBH
Encinitas	BAKERY RETARDER	GROCERY FREEZER			JUICE
	BEVERAGE	ICE CREAM			LUNCH MEAT
	DAIRY	ICE CREAM			MEAT
	DAIRY	ICE CREAM			MEAT
	DAIRY	ICE CREAM			ORGANIC
	DAIRY COOLER 14DRS	ICE CREAM			PRODUCE
	EGGS	ICE CREAM			PRODUCE
	REF. CASH STAND	ICE CREAM			PRODUCE
	SERVICE & BAKERY	ICE CREAM BAKERY			PRODUCE CLR
					PRODUCE/JUICE
Sys. Designation:	B	C	DL	E	
Design Load:	301.92 MBH	160.42	74.78 MBH	197.71	
San Diego	BAKERY RETARDER	BAKERY FREEZER	DUAL TEMP	BERRIES	
	BEER	ICE CREAM	DUAL TEMP	FLORAL	
	BEER / BEVERAGE	ICE CREAM	DUAL TEMP	LUNCH MEAT	
	DAIRY	ICE CREAM	DUAL TEMP	MEAT	
	DAIRY	ICE CREAM	DUAL TEMP	MEAT	
	DAIRY	ICE CREAM	DUAL TEMP	MEAT COOLER	
	DAIRY / DELI	ICE CREAM	FROZEN MEAT	PROD. PROMO	
	DAIRY COOLER 11DRS	ICE CREAM BAKERY	FRZN SEAFOOD	SEAFOOD	
	EGGS	ICE FLAKER	GROCERY FZR	SERVICE MEAT	
		SUSHI		SVC SEAFOOD	

Figure 36 through Figure 40 show the load profiles for the two stores as well as the simulated load profile for the Santa Clara store. On average, all suction groups were observed to be approximately 40%-65% loaded (vs. the design load) during the two-week monitoring period.

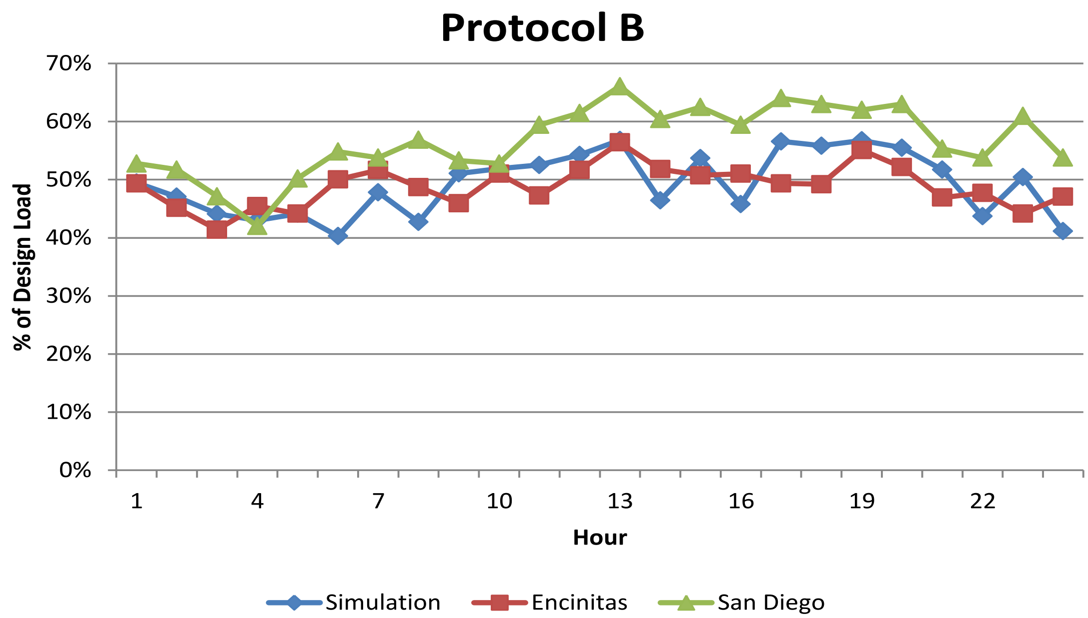


FIGURE 34: LOAD PROFILES FOR SANTA CLARA COUNTY PROTOCOL B, SHOWN WITH COMPARABLE LOADS FROM THE SAN DIEGO AND ENCINITAS STORES.

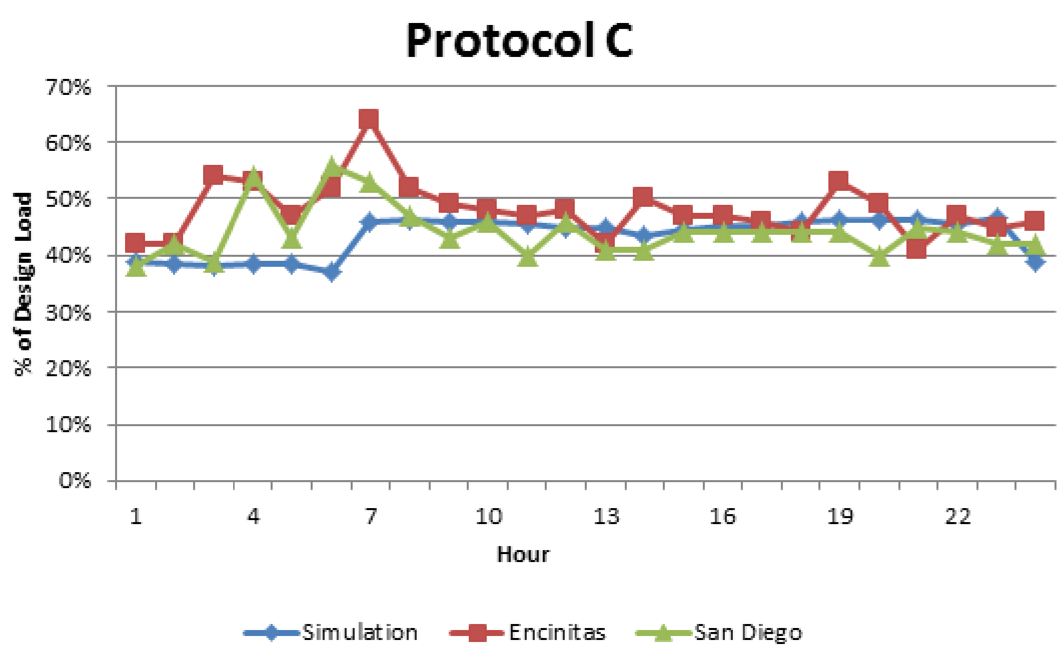


FIGURE 35: LOAD PROFILES FOR SANTA CLARA STORE PROTOCOL C, SHOWN WITH COMPARABLE LOADS FROM THE SAN DIEGO AND ENCINITAS STORES

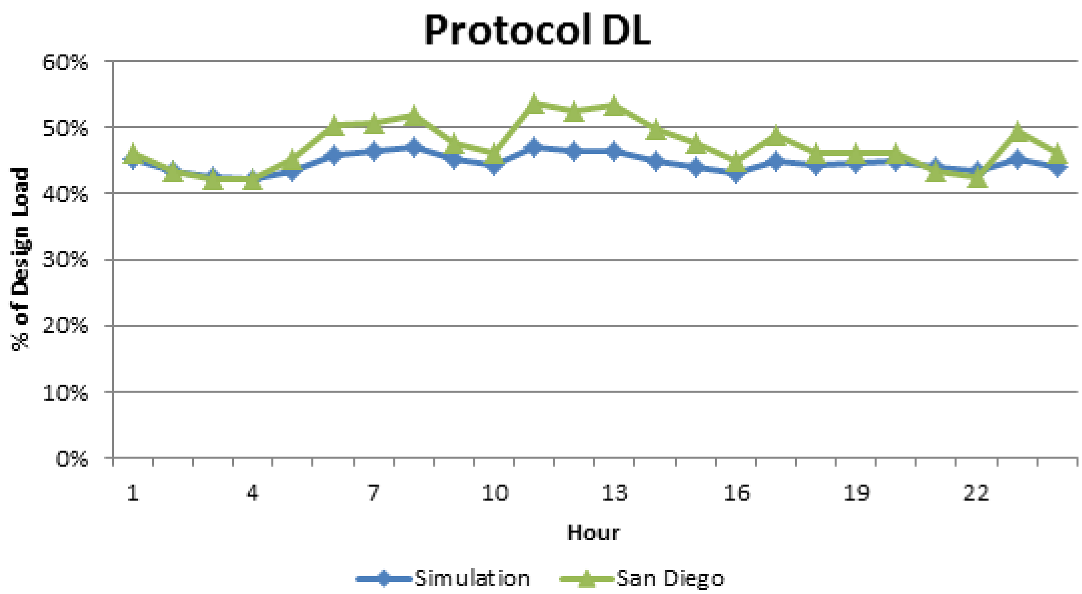


FIGURE 36: LOAD PROFILES FOR SANTA CLARA COUNTY PROTOCOL DL, SHOWN WITH COMPARABLE LOADS FROM THE SAN DIEGO AND ENCINITAS STORES

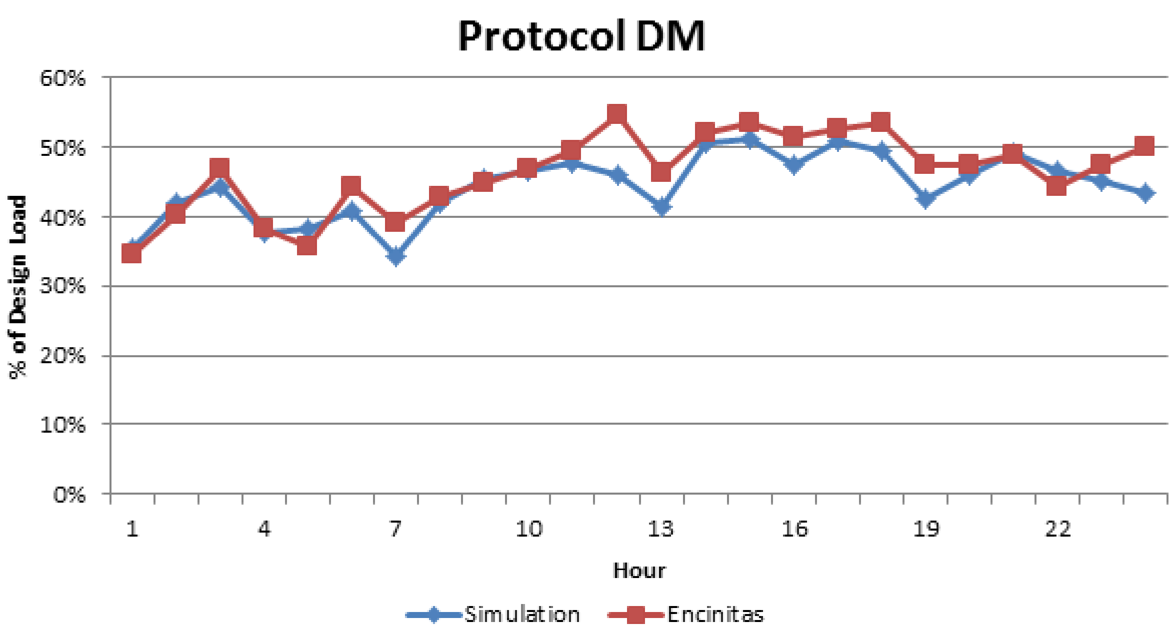


FIGURE 37: LOAD PROFILES FOR SANTA CLARA COUNTY PROTOCOL DM, SHOWN WITH COMPARABLE LOADS FROM THE SAN DIEGO AND ENCINITAS STORES

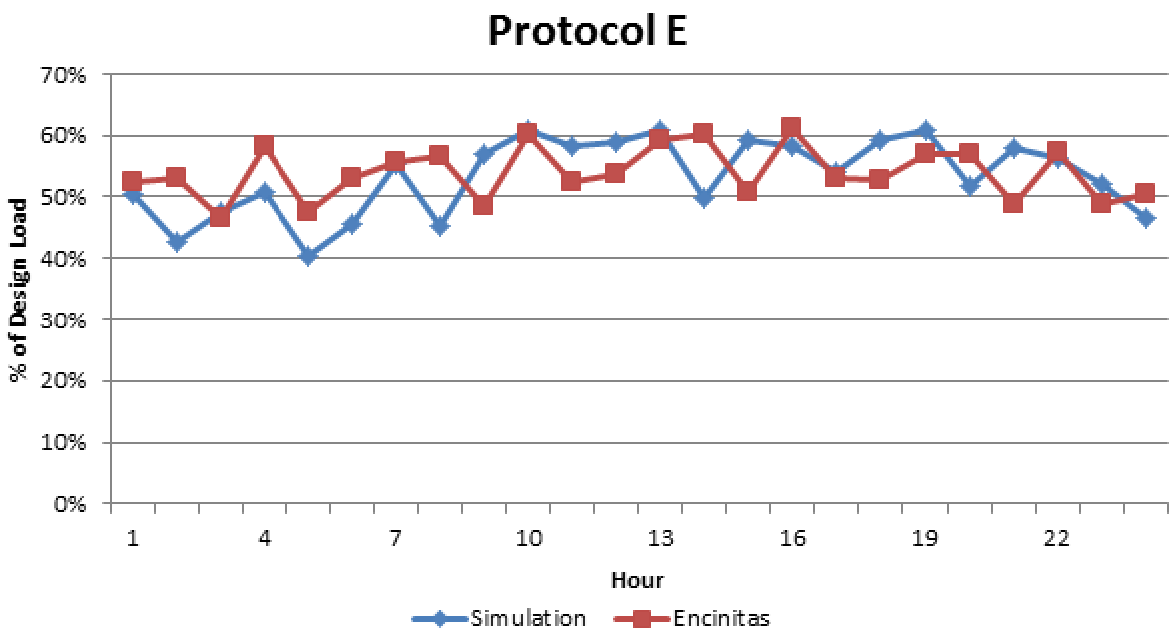
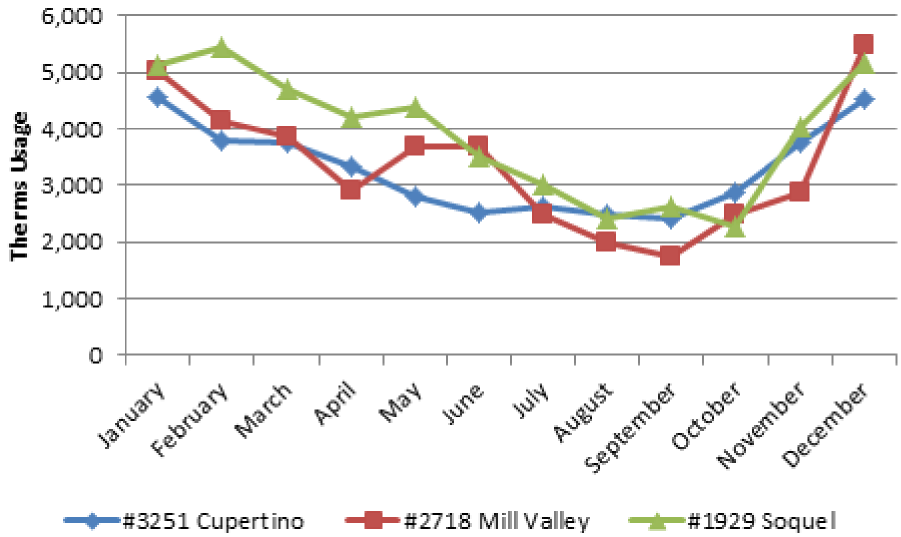


FIGURE 38: LOAD PROFILES FOR SANTA CLARA COUNTY PROTOCOL E, SHOWN WITH COMPARABLE LOADS FROM SAN DIEGO AND ENCINITAS STORES

NATURAL GAS USAGE ANALYSIS

Simulated natural gas usage for the Santa Clara County location energy model (without heat recovery) was compared to real usage history from supermarkets from the same chain located in the same climate zone as the subject store. The usage-histories being compared to 2007-year histories are for a store in Mill Valley, California, and a store in Soquel, California. The Santa Clara County store energy model is in-between Mill Valley and Soquel in terms of floor area (53,500 SF versus 43,900 SF for Mill Valley and 67,100 SF for Soquel). Lighting power density for the Mill Valley (2.24 Watts/SF) and Soquel (1.74 Watts/SF) are higher than Santa Clara County location (1.04 Watts/SF). Figure 13 below shows the monthly natural gas usage for the two stores, along with the simulated natural gas usage for the Santa Clara County store.



Month	Santa Clara County	Mill Valley	Soquel
January	4,555	5,010	5,139
February	3,801	4,151	5,460
March	3,744	3,862	4,704
April	3,321	2,908	4,229
May	2,807	3,674	4,373
June	2,514	3,683	3,524
July	2,622	2,492	3,000
August	2,482	1,998	2,405
September	2,413	1,755	2,612
October	2,859	2,486	2,287
November	3,762	2,880	4,020
December	4,518	5,471	5,169

FIGURE 39: NATURAL GAS USAGE FOR THE SANTA CLARA COUNTY STORE, COMPARED TO TWO OTHER SUPERMARKETS

The figure shows that the simulated natural gas usage is generally comparable to the two other stores. The lighting power density at the Santa Clara County location is lower than Mill Valley and Soquel, which would expectedly increase natural gas usage. The host site, however, also utilizes heat pumps instead of natural gas heaters for several secondary heating loads, and also has less display case lighting and reach-in anti-sweat heater energy. Identify all the variables that will provide the output performance of each technology/product and power input to the technology/product so that the efficiency of the technology/product can be determined.

APPENDIX C: BUILDING SUMMARY

The supermarket used for this analysis is approximately 54,000 S.F., out of which 53,500 S.F. is refrigerated or conditioned. The main sales area is conditioned by the Season4 air handling unit. Other areas are conditioned using Trane packaged rooftop units and Fujitsu heat pumps.

Refrigeration is accomplished by one low-temperature (LT) system, two dual-temperature (DT) systems and three medium-temperature (MT) systems, which use R-507 refrigerant. The refrigeration systems consist of six distributed Hussmann Protocol parallel rack systems, utilizing Copeland scroll compressors. Krack air cooled units work as the refrigeration condensers. The following tables outline the primary building systems and components that were considered in the analysis.

For this analysis the store was assumed to be open 24 hours per day, 7 days per week.

ENVELOPE AND LIGHTING POWER

Building construction details and lighting power are described in Figure 35 and Figure 36 below.

TABLE 30: ENVELOPE DESCRIPTION

Wall Construction	Mass Heavy Wall. U-factor: 0.65
Roof Insulation	Wood Framed and Other Roof. U-factor: 0.039
Roof Absorptivity	Absorptivity = 0.45

TABLE 31: LIGHTING POWER

Space	Area (SF)	Lighting Power Density (Watts/SF)	Total Power (Watts)
Sales	38,309	1.04	39,972
Bakery	1,635	1.74	2,840
Pharmacy	288	2.17	625
Deli	1,247	1.84	2,296
Receiving	6,462	0.40	2,578
Produce Prep	288	1.71	492
Offices/Restrooms	2,315	1.26	2,909
Meat Prep	880	2.09	1,836
Total	**51,424**	**1.04**	**53,547**

HUSSMANN PROTOCOL REFRIGERATION SYSTEMS

The Santa Clara County store uses Hussmann Protocol parallel rack distributed refrigeration systems. Table 22 below summarizes the design parameters for the systems.

TABLE 32: HUSSMANN PROTOCOL REFRIGERATION SYSTEM SUMMARY

System Configuration	Hussmann Protocol single-stage parallel systems with scroll compressors Design saturated suction temperatures and design loads as follows: Protocol A Suction Group AL: -15°F, 11.1 MBH Suction Group AM: +18°F, 198.36 MBH Protocol B: +24°F, 260.57 MBH Protocol C: -20°F, 119.36 MBH Protocol D Suction Group DL: -23°F, 84.41 MBH Suction Group DM: +18°F,76.91 MBH Protocol E: +24°F, 237.74 MBH Protocol F: +18°F, 39.8 MBH
Compressor Quantity and Model	Copeland scroll compressors Protocol A Suction Group AL: (1) ZF11K4E economized Suction Group AM: (5) ZB45KCE Protocol B: (6) ZB45KCE Protocol C: (6) ZF18KVE economized Protocol D Suction Group DL: (1) ZFD18KVE economized digital, (3) ZF18KVE economized Suction Group DM: (1) ZBD45KCE digital, (1) ZB45KCE digital Protocol E: (6) ZB45KCE Protocol F: (6) ZB45KCE
Refrigerant	R-507

The compressor control strategy includes electronic sequencing and floating suction pressure logic, allowing the refrigeration system suction pressure to float rather than operating at a fixed suction setpoint. During low-load periods, the compressors operate at a higher suction temperature, improving compressor pumping efficiency, while still maintaining temperature setpoint in the display cases and walk-ins. All suction groups except suction group AL were assumed to have floating suction pressure control.

Suction groups DL and DM each have one compressor which features Copeland's digital unloaders. The "digital" feature of these compressors allows the suction group capacity to more closely match the required refrigeration load, compared to cycling the compressors off and on, resulting in more stable and higher average suction pressures on all systems.

KRACK AIR-COOLED CONDENSERS

The Protocol refrigeration systems are served by Krack LEVE air-cooled refrigeration condensers.

TABLE 33: KRACK LAVE CONDENSER SUMMARY

Protocol	Condenser Make/Model	Capacity (MBH at 10°F TD)	Number of Fans	kW/Fan	Specific Efficiency at 10°F TD (Btuh/Watt)
A	Krack LAVE 15410	282	5	0.41	139
B,C	Krack LEVE 26410	338	6	0.41	139
D,E	Krack LEVE 26410	338	6	0.41	139
F	Krack LEVE 16410	338	6	0.41	139

The refrigeration condensing control strategy includes floating head pressure to 70°F saturated condensing temperature (SCT) with ambient following control logic and condenser fan cycling control. The ambient following control logic sets the target SCT by adding a fixed control temperature difference (TD) to the ambient temperature. For this analysis, the ambient-following control TDs were initially simulated at the design TDs and then checked to determine if a lower TD would increase savings. The optimum TD was determined iteratively and then increased by two degrees to avoid over-optimization of simulation results.

As noted previously in the report, these condensers are very efficient (low fan power).

REFRIGERATED DISPLAY CASES AND WALK-INS

The refrigerated display cases are summarized in the following tables. Table 24 summarizes the open cases.

TABLE 34: OPEN CASE SUMMARY

Suction Group	Description	Case Model	Case Length (ft)	Number of Canopy Lights	Number of Shelf Lights	Design Load (MBH)
AM	Sandwich Prep	Hussmann CSP	10	Two rows	Zero rows	12.50
AM	Ref. Front Case	Hussmann FMSS	10	Zero rows	Zero rows	4.50
AM	Pick-up	Hussmann FC-2D	2	Two rows	Zero rows	0.80
AM	Grab and Go	Structural Concepts FSE65R and FDSC45R	20	Two rows	Zero rows	16.55
AM	Grab and Go	Hussmann E3	16	Two rows	Three Lit Shelves	19.50
AM	Sushi	Hussmann Q3-SP	6	Two rows	Zero rows	6.96
AM	Service Deli	Hussmann ESBDVS	24	Two rows	Zero rows	7.56
AM	Cheese Table	Hussmann RI-4	12	One row	Three Lit Shelves	24.60
AM	Pizza	Hussmann D5XLEP	6	Two rows	Five Lit Shelves	7.95
AM	Cheese	Hussmann Q1-SS	24	Zero rows	Zero rows	15.48
AM	Service and Bakery	Hussmann C3VB96	12	Two rows	Zero rows	13.20
AM	Bakery	Hussmann RI-3	10	One row	Two Lit Shelves	17.33
AM	Island Dairy	Columbus IMBRSS8216	18	Two rows	Zero rows	43.08
B	Dairy	Hussmann D5XLEP	76	Two rows	Zero rows	106.68
B	Beverage	Hussmann D5XLEP	64	Two rows	Zero rows	80.96
B	Deli	Hussmann D5XLEP	12	Two rows	Zero rows	22.92
DL	Dual Temp	Hussmann FWG/FWEG	36	Zero rows	Zero rows	37.86
DM	Seafood	Hussmann Q3-SS	8	One row	Two Lit Shelves	10.00
DM	Service Seafood	Hussmann Q1-FC	12	Two rows	Zero rows	2.70
DM	Service Meat	Hussmann Q3-MC	12	Two rows	Zero rows	4.32
DM	Melon	Hussmann RI-2	8	One row	Two Lit Shelves	11.11
E	Lunch Meat	Hussmann D5XLEP	28	Two rows	Zero rows	39.76
E	Meat	Hussmann M5XEP	52	Two rows	Three Lit Shelves	49.86
E	Dairy	Hussmann P4XEP	28	Two rows	Zero rows	36.12
F	Floral	Borgen	13	Two rows	Zero rows	11.43
F	Berry	Hussmann E3	24	Two rows	Three Lit Shelves	28.32
F	Produce	Hussmann P2XEP	56	Two rows	Zero rows	49.84
F	Produce	Hussmann D5XLEP	54	Two rows	Zero rows	68.31
F	Organic	Hussmann C2XXLEP	16	Two rows	Zero rows	14.72
F	Beverage End	Hussmann D5XLEP	6	Two rows	Zero rows	7.59

The medium-temperature display cases include Hussmann EcoShine II Plus LED lights. The following table outlines the assumed power consumption for the LED lights.

TABLE 35: OPEN CASE LED LIGHT SUMMARY

Suction Group	Description	Model	Line Up Length (ft)	Canopy LED Watts/12 ft	Shelf LED Watts/12 ft
AM	Service Deli	ESBDVS	24	93	0
AM	Pizza	D5XLEP	6	93	158
B	Dairy	D5XLEP	76	93	0
B	Beverage	D5XLEP	64	93	0
B	Deli	D5XLEP	16	93	0
DM	Seafood	Q3-SS	8	63	63
DM	Service Seafood	Q1-FC	12	63	0
DM	Service Meat	Q3-MC	12	63	0
E	Lunch Meat	D5XLEP	28	93	0
E	Meat	M5XEP	52	93	95
E	Dairy	D5XLEP	24	93	0
F	Produce	P2XEP	56	93	0
F	Produce	D5XLEP	54	93	0
F	Organic	C2XXLEP	16	93	0
F	Beverage End	D5XLEP	6	93	0
F	Produce	P4XEP	28	93	95

Display case lights are automatically controlled by the store's EMS system. For this analysis the display case lights are assumed to be automatically turned off between 11pm and 6 am.

The low-temperature reach-in display cases include Hussmann's "Zero-Watt" Hussmann RL Innovator II doors. The frame anti-sweat heaters in the Innovator II doors draw 54 watts, while the door anti-sweat heaters draw no power. Table 26 below summarizes the assumed power consumption of the anti-sweat heaters.

TABLE 36: DOOR CASE SUMMARY

Suction Group	Case Line Up	Case Model	No. of Door Lights	Case Length (Door)	Lighting kW/door
C	Ice Cream	Hussmann RL	Two vertical LED lights at each mullion and one vertical LED light at each end.	51	0.0270
C	Frozen Food	Hussmann RL	Two vertical LED lights at each mullion and one vertical LED light at each end.	72	0.0270
DL	Frozen Food	Hussmann RL	Two vertical LED lights at each mullion and one vertical LED light at each end.	8	0.0270

REFRIGERATED DISPLAY CASES AND WALK-INS

Included in this project are assumptions for process electric and natural gas consumption. The following are descriptions of both background electric and natural gas loads included in this analysis.

ELECTRIC ENERGY

Table 27 describes the background electric power loads per space that were included in the simulation:

TABLE 37: SUMMARY OF ASSUMED BACKGROUND LOADS PER SPACE

Space	Power (kW)
Bakery	20.3
Customer Service	0.11
Deli Department	8.86
Stockrooms	7.86
Employee Room	0.14
Meat Department	3.40
Sales Area	5.11
All remaining spaces	0.25 Watts/SF

Loads considered as part of this analysis include kitchen equipment, exhaust fans, stand-alone refrigerators, cash registers, vending machines, computers, and task lighting, among other loads. The background energy loads are subject to assumed 24-hour load profiles that were developed for each space. The estimated usage patterns of the individual equipment loads were aggregated into overall space load profiles for each of the mentioned spaces and departments. The following DOE2 code snippet is an example 24-hour load profile developed for the Bakery Space:

```
Bakery_Pwr_Sched              = SCHEDULE
  TYPE                        = FRACTION
  THRU  DEC 31  (ALL)
        (1,2)   (0)
        (3,4)   (0)
        (5,6)   (0)
        (7,8)   (0)
        (9,10)  (0.575)
        (11,12) (1)
        (13,14) (0.913)
        (15,16) (0.284)
        (17,18) (0.077)
        (19,20) (0.077)
        (21,22) (0.077)
        (23,24) (0.077)
        ..
```

The following is an abbreviated example of the loads that were accounted for in the simulation. The example below shows the estimated power and duty cycle of a shortened list of equipment in the Bakery Space:

TABLE 38: ABBREVIATED LIST OF LOADS FOR THE BAKERY SPACE

Department	Load	HP	Amps	Phase	Volts	Est. Power (Watts)	Est. Duty Cycle
Bakery	Cash Register		3	1	120	72	100%
	Debit Machine		1.6	1	120	38	100%
	80 Quart Mixer	3		3	208	2,798	80%
	Spiral Mixer	15		3	208	12,503	50%
	Icing Warmer		12.5	3	208	900	75%
	Donut Fryer		1.7	3	120	50	100%
	Proofer		42.5	1	120	4,800	25%
	Proofer Control Power		3.5	1	120	84	25%

NATURAL GAS

The following table describes the kitchen equipment that was simulated as a background natural gas load in the simulation:

TABLE 39: ASSUMED BACKGROUND NATURAL GAS LOADS

Load	Rated MBH
Pan Washer	55
Donut Fryer	60
Rack/Revolving Oven	275
Dishwasher	25
Pizza Oven	190
Chicken Fryer	44
Gas ReThermalizer	55
Combo Oven	82

It was assumed that all natural gas equipment operates approximately three hours per day.