Space Heat Recovery from Refrigeration

Interim Performance Report

ET Project Number: ET12PGE1391

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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
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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Pacific Gas and Electric Company®

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EXECUTIVE SUMMARY

Pacific Gas & Electric Company (PG&E) is working with a retail grocery store chain, to study heat recovery in a new supermarket in Santa Clara County, California, including energy analysis and field monitoring to increase understanding of natural gas savings and consequent electric energy penalty associated with heat recovery. As part of this project, the company (the grocery store chain) installed a heat recovery system to utilize heat from four of the six refrigeration systems to heat the sales area via a parallel-connected, direct-condensing heat recovery design.

The subject refrigeration systems and primary HVAC system were outfitted with instrumentation and data acquisition equipment to monitor their electric energy and natural gas usage. An on-site monitoring panel collects the sensor data and transmits it for processing via wireless modem. The store's Danfoss energy management system (EMS) is also used to obtain additional refrigeration system data for the compressors, condensers, and other system operating parameters.

The performance of the heat recovery system is 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 is calculated analytically.

The heat recovery system was commissioned on August 3, 2014, which marks the beginning of the data set used for this analysis. The analysis period for this report ends on September 31, although the overall data collection effort will continue through February, 2015. The analysis presented here is intended to serve only as an interim performance report.

The energy usage and savings figures presented in this report represent 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 interim results of the study are presented in Table 1 below. This store is open 24 hours per day, 7 days per week.

TABLE 1: INTERIM HEAT RECOVERY PERFORMANCE RESULTS SUMMARY

	Energy Usage				Energy Cost			
	Energy (kWh)	Natural Gas (Therms)	Total Energy (MMBtu)	Peak Demand (kW)	Energy ($)	Demand ($)	Natural Gas ($)	Total ($)
Without Heat Recovery	50,539	1,098.1	273.2	76.0	$5,992.16	$3,276.19	$849.91	$10,118.26
With Heat Recovery	52,102	0	177.8	77.8	$6,131.79	$3,347.29	$0.00	$9,479.08
Savings	(1,563)	1,098.1	95.4	(-1.8)	($139.63)	(-$71.10)	$849.91	$639.18
Savings (%)	(-3.1%)	100%	34.9%	(-2.4%)	(-2.3%)	(-2.2%)	100%	6.3%

Interval data for August 3 to September 31, 2014

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

HISTORY

Use of heat from refrigeration systems to provide space heating in supermarkets has a long history and at one time was used to extensively and provided all or most of the heat in many stores, both in California and across the US. However, largely to reduce refrigerant change and leakage, heat recovery became less common in recent decades. To reduce energy usage, operating costs and meet sustainability objectives, many supermarket chains are again considering heat recovery. In addition, the 2013 California Title 24 Building Energy Efficiency Standards include requirements to use at least 25% of the heat from refrigeration for space heating in new supermarkets.

ASSESSMENT OBJECTIVES

This project will assess 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 will compare the performance of the heat recovery system to a system with no refrigeration heat recovery and a standard natural-gas furnace for space heating.

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 will be processed and compared to a theoretical scenario consisting of refrigeration and HVAC systems with no connected heat recovery capacity.

TECHNOLOGY EVALUATION

The system being monitored consists of a four-circuit direct-condensing heat recovery coil installed inside the main air handling unit (AHU) serving the supermarket sales area. The coil is upstream of the natural gas furnace, with each coil circuit connected to the discharge of one of four refrigeration systems (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. 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 inside the recovery coil, 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 systems is shown.

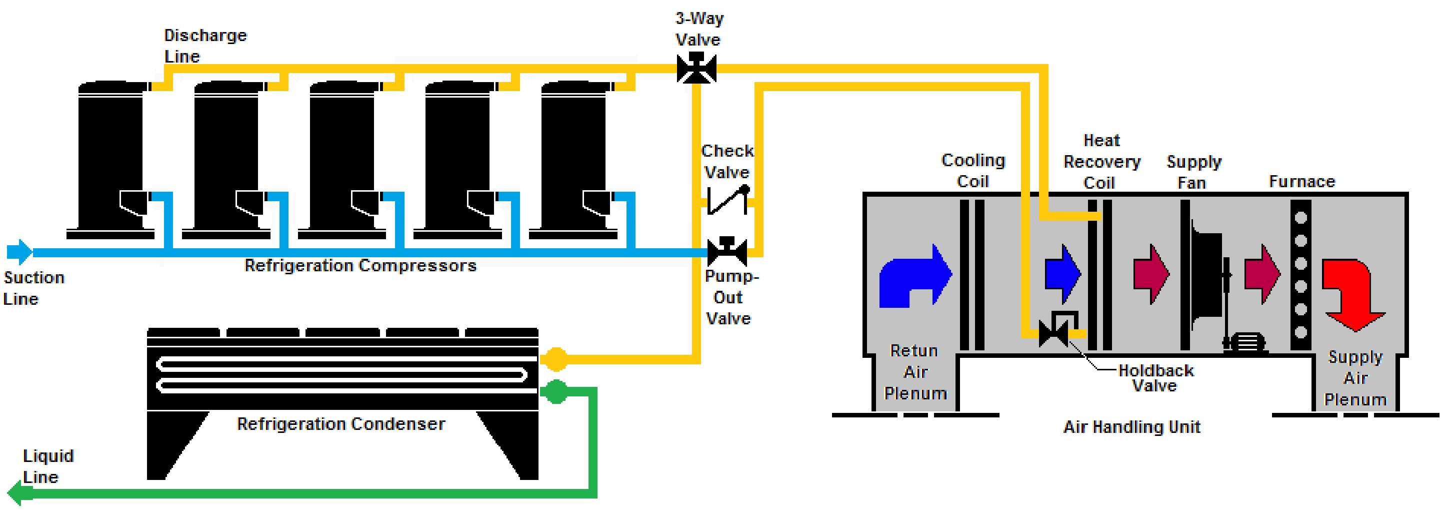


FIGURE 1: SCHEMATIC DRAWING OF HEAT RECOVERY SYSTEM

Heat recovery is the primary space heating method. 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. Space heating consists of three stages:

- Stage 1 is 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 Stage 1 and 2, and adds the modulating 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 will be added to the main air handling unit where the heat recovery condensing coil is installed. A monitoring panel located in the condenser section of the air handling unit collects the sensor data and transmits it for processing via wireless modem. The existing Danfoss energy management system will be used to obtain additional refrigeration system data concerning the compressors, condensers, and other system operating parameters.

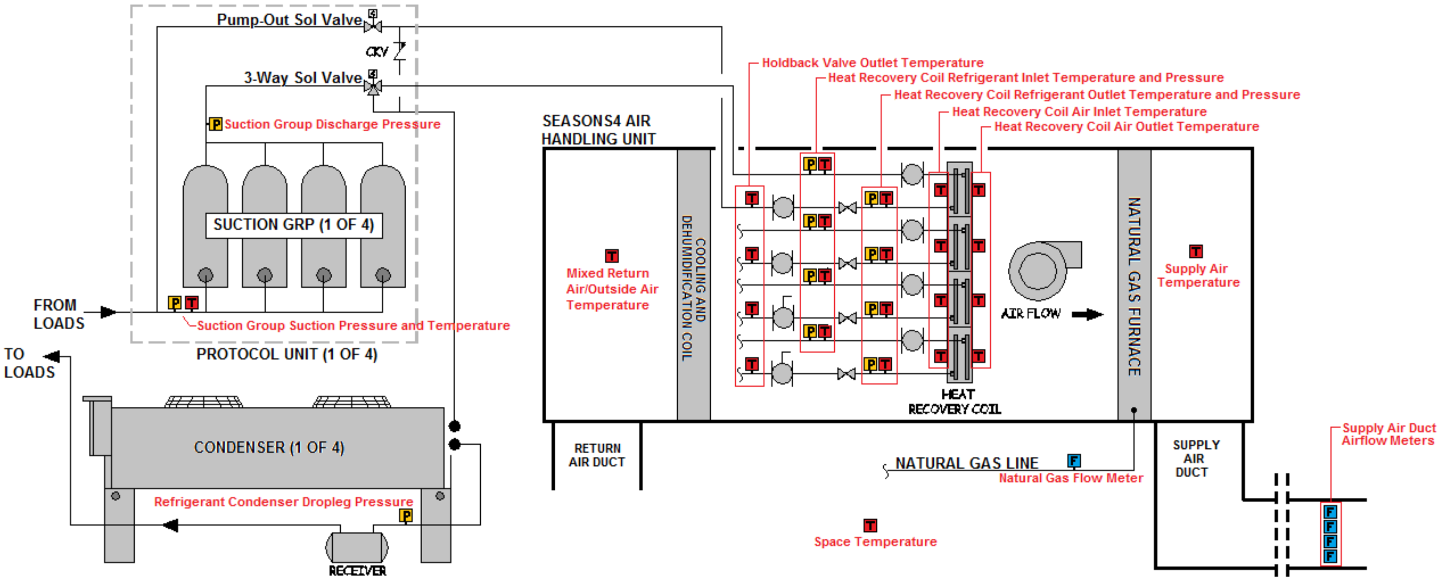


FIGURE 2: INSTRUMENTATION DIAGRAM FOR HEAT RECOVERY EVALUATION

RESULTS

Presented below are the interim results for the heat recovery ET project. The results are presented in the following sections:

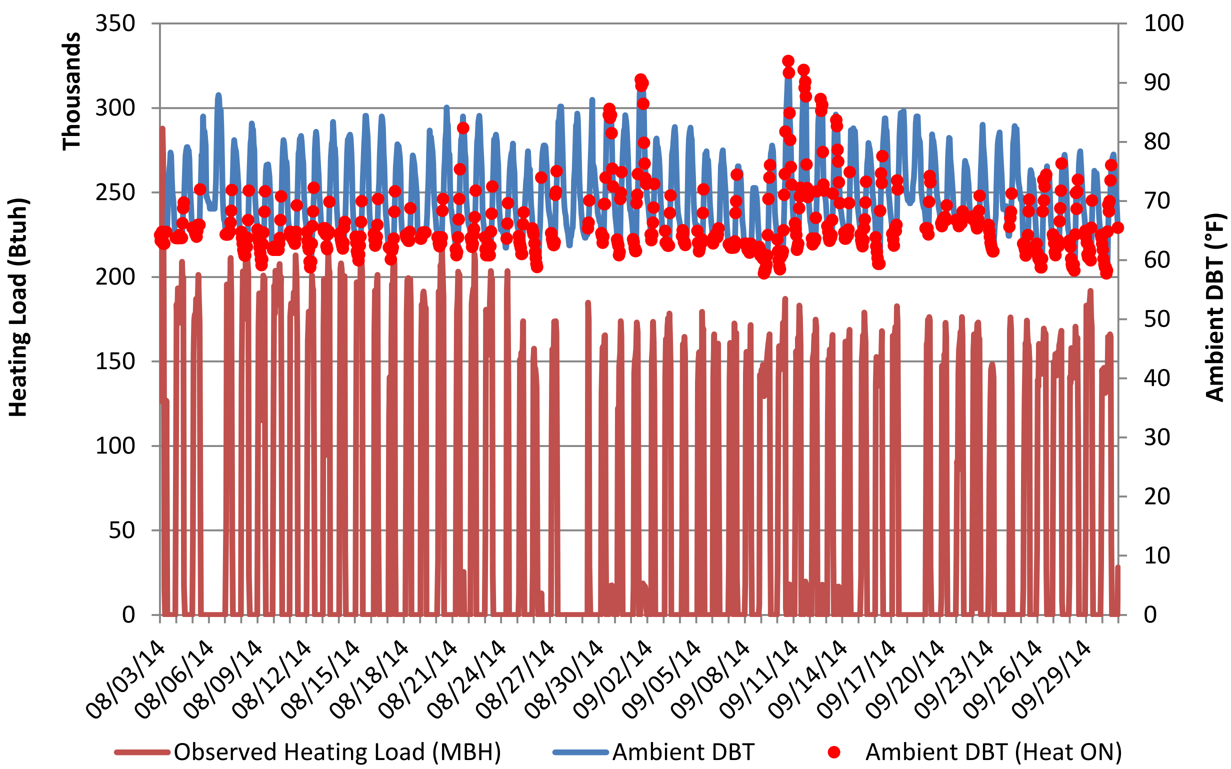
- Air Handling Unit
- Refrigeration Condensers
- Refrigeration Compressors
- Overall Energy Usage

AIR HANDLING UNIT

Analysis of the refrigeration load and energy usage for the refrigeration systems with heat recovery is presented in the sections below.

HEATING LOAD ANALYSIS

Figure 3 shows the observed heating load superimposed with ambient drybulb temperature for the subject test period versus time. Heating load is calculated based on recovered heat from the refrigeration systems. The heating load references the left scale while the ambient drybulb references the right vertical scale. Shown below the heating load and drybulb trends is a status trend of the System B and C heat recovery coil, the System D and E heat recovery coil, and the natural gas furnace.



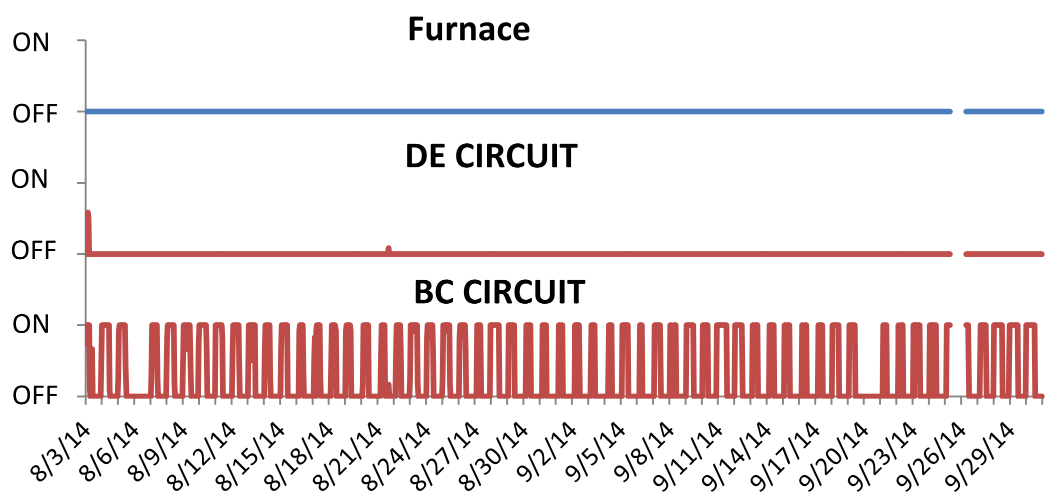


FIGURE 3: HEATING LOAD AND AMBIENT DBT VERSUS TIME

Figure 3 shows that the heating load served by the main air handling unit was satisfied almost entirely by the first stage of heat recovery for the entire subject test period. Neither the second stage of heat recovery nor the natural gas furnace was effectively ever required. Heating capacity is required almost daily for during the late night and early-morning hours.

Figure 4 below shows a 24-hour analysis of the required heating load and the outside ambient temperature for every day of the subject test period. Heating load is shown in the top plot while ambient temperature is shown on the bottom. Each plot represents one full day during the test period. For both required heating load and ambient drybulb, the hourly bin-average of all hours across each day is highlighted as a red plot.

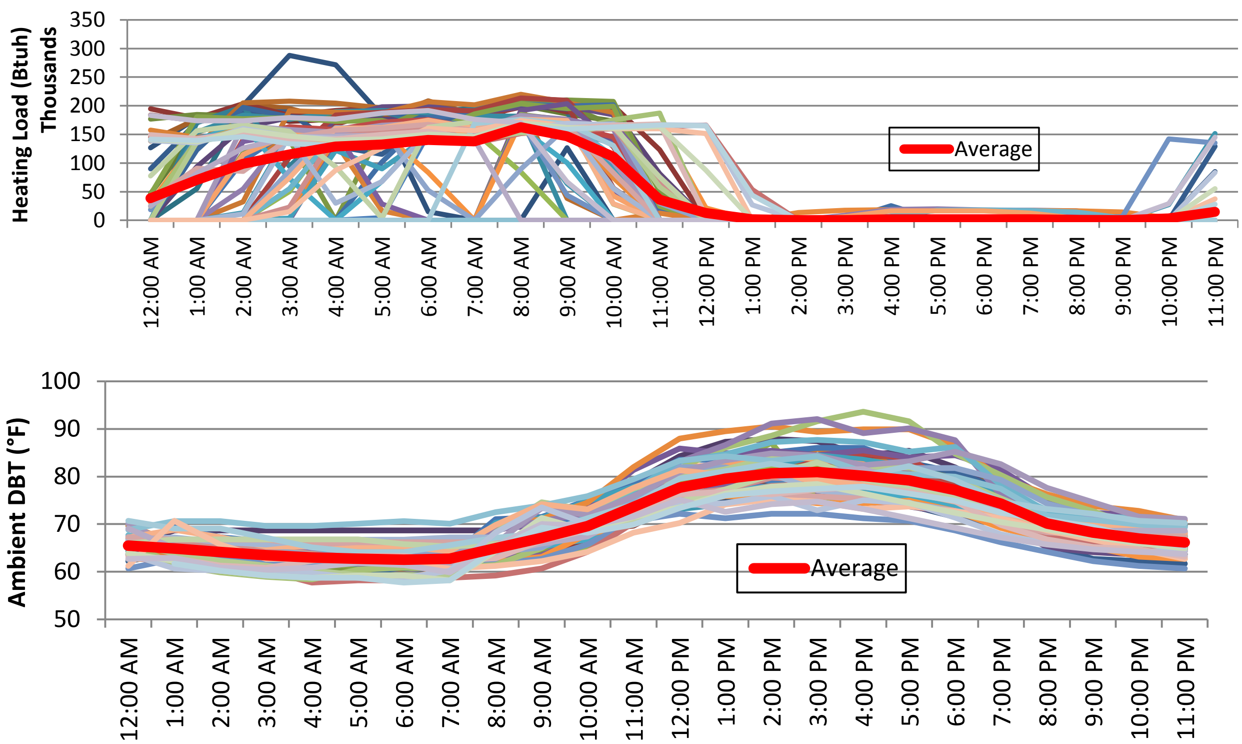


FIGURE 4: HEATING LOAD AND AMBIENT DBT OVER 24 HOURS FOR SUBJECT TEST PERIOD

Figure 5 below shows just the 24-hour bin-averaged heating load and ambient drybulb temperature from the data shown in Figure 4, with both plots superimposed.

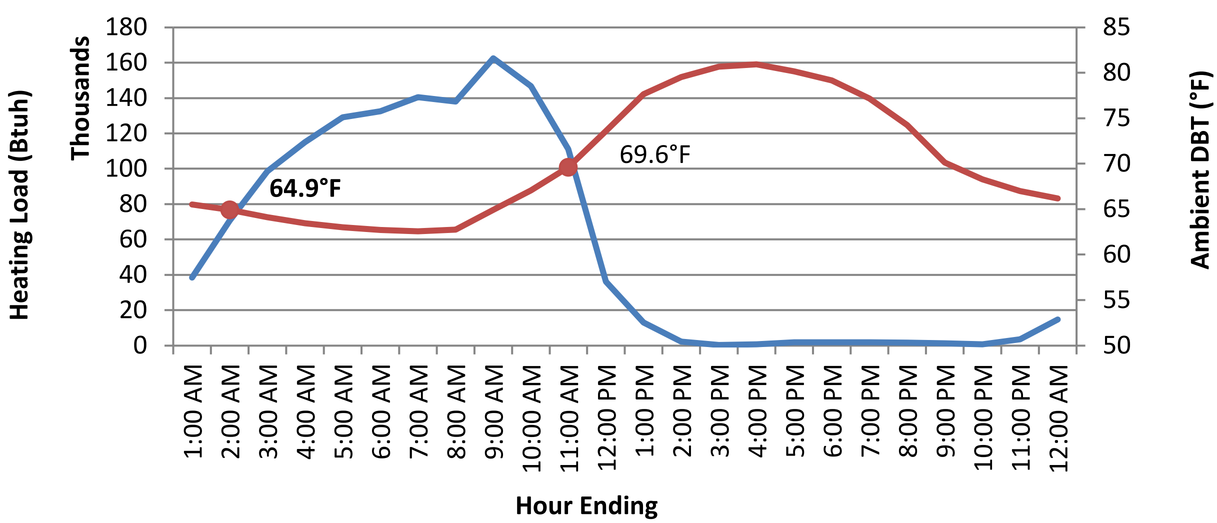


FIGURE 5: 24-HOUR BIN-AVERAGE HEATING LOAD PROFILE AND OUTSIDE AMBIENT DRYBULB TEMPERATURE

Figure 4 shows that, on average, the heating load started between the hours of 11:00 PM and midnight, and lasted until approximately noon. The peak load occurred between approximately 8:00 AM and 9:00 AM every day. Ambient drybulb temperature was generally between the low-60's and mid-80's for every day of the test period. Figure 5 shows a correlation between the heating load profile and the ambient drybulb temperature, with the first stage of heating turning on when the DBT is lower than 65°F, on average, and turning off when the DBT is higher than 70°F.

Natural Gas Usage and Savings

The furnace is the third and final stage of heating capacity in the AHU. During the subject test period, the furnace was never required, and therefore the observed natural gas usage was zero.

In the Base Case, the theoretical natural gas usage is calculated by dividing the calculated recovered heat by the furnace thermal efficiency, which is 80%. Table 2 below summarizes the recovered heat and calculated Base Case natural gas savings.

Table 2: Recovered Heat and Base Case Therm Savings

Total Heat Recovered from Refrigeration Systems	Base Case Therm Usage	Savings
803.9 Therms	1,004.9 Therms	1,004.9 Therms (100%)

Further analysis may reveal that the Base Case the furnace efficiency is higher than when heat recovery is on because the system exergy will be higher; the temperature of the air entering the furnace would be lower in the Base Case (since it would not be pre-heated by the heat recovery system), 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 combusted, and thus higher thermal efficiency. As the ambient temperature gets lower toward the end of the test period and heating load increases (and the stage-3 furnace is presumably required periodically), this hypothesis will be tested. Furnace heat will be calculated from the supply air flowrate, the leaving reclaim coil air temperature, and the supply (e.g. leaving AHU) 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. Base Case verification data will also be collected, in which the heat recovery coils are disabled for a week or more.

Supply Fan Operation

The supply fan of the main air handling unit is equipped with an ABB variable speed drive. The fan is controlled at one of three fixed speeds, depending on the status of the heating and cooling operation. According to Seasons4 application engineers, the intended supply fan speed control strategy is as follows:

- The fan operates at 100% speed (60 Hz.) any time the AHU is in the second stage of cooling or if the natural gas furnace is on

- The fan operates at 90% speed (54 Hz.) if one or both stages of heat recovery are activated, but the natural gas 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

AHU supply airflow rate is measured in the air supply duct by a differential pressure sensor array installed approximately 100 feet downstream of the air handling unit. Figure 6 shows the measured airflow in the main supply duct versus time for the portion of the subject test period. Also shown at the bottom of the figure is the furnace, cooling, and heat recovery status for comparison.

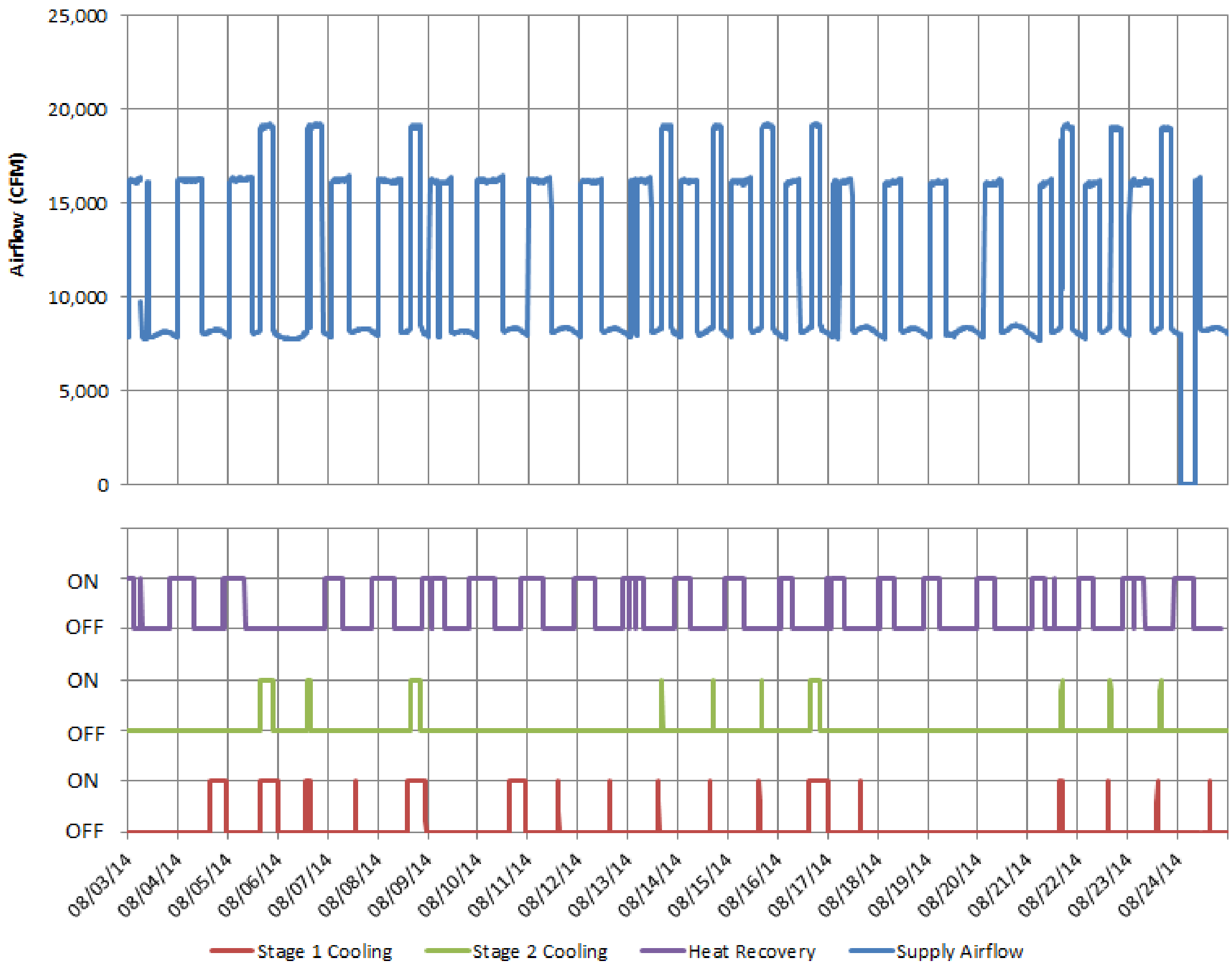


FIGURE 6: AHU SUPPLY AIRFLOW (TOP) AND COOLING/HEATING ON-TIME (BOTTOM) FOR MAIN AHU

The airflow trend at the top of Figure 6 confirms that the supply fan speed is controlled at three distinct fan speeds, which correlate with the fan speed thresholds described by Seasons4, as shown below in Table 3.

TABLE 3: OBSERVED PERCENT AIRFLOW RATE VERSUS ASSUMED FAN SPEED

Fan Speed (%, Hz)	Average Observed Airflow Rate (CFM)	Percent of Maximum Measured Airflow Rate
100%	19,100 CFM	100%
90%	16,300 CFM	85%
35%	7,900 CFM	41%

Figure 6 also shows that the airflow rate is roughly 85% of the maximum measured rate when at least one stage of the heat recovery was activated. The airflow rate was approximately 40% of the maximum measured rate when no stage of either heating or cooling was activated. The maximum measured airflow rate occurred when just the second stage of cooling was activated. Since the natural gas furnace was not used during the test period it could not be verified that the fan runs at full speed when the furnace is activated.

SUPPLY FAN ENERGY

Analysis of the supply fan energy usage includes consideration for the additional static pressure drop associated with the heat recovery coils, which is directly proportional to the energy penalty from the same. Table 4 below shows the static pressure, power, and calculated energy usage for the main AHU supply fan.

TABLE 4: MAIN AHU SUPPLY FAN STATIC PRESSURE, POWER, AND ENERGY USAGE

	Base Case	Observed
Static Pressure	4.49 In. WC	4.01 In. WC
Supply Fan Brake Horsepower	19.4 BHP	17.33 BHP
Supply Fan Electric Power Usage at Max Speed	15.8 kW	14.1 kW
Supply Fan Energy Usage for Subject Test Period	6,151 kWh	6,887 kWh

VARIABLE HOLDBACK VALVE OPERATION

The heat recovery system features four Sporlan CDS electronic pressure regulating holdback valves, one per refrigeration system, located immediately downstream of each of the heat recovery coil circuits inside the air handling unit. The valves continuously modulate in order to control the refrigerant pressure inside 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 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 20°F, subject to minimum and maximum limits. In addition, the valve control strategy includes a minimum open percentage of 10% (25% for System C), which minimizes the risk of high pressure events due to valve hunting or fast system changes. Figure 14 shows the condensing temperature, mixed return/ventilation air temperature, and holdback valve position for System B during a 12-hour period from midnight to noon on September 19. For the period shown, heat recovery is on from approximately 1:06 AM to 11:32 PM.

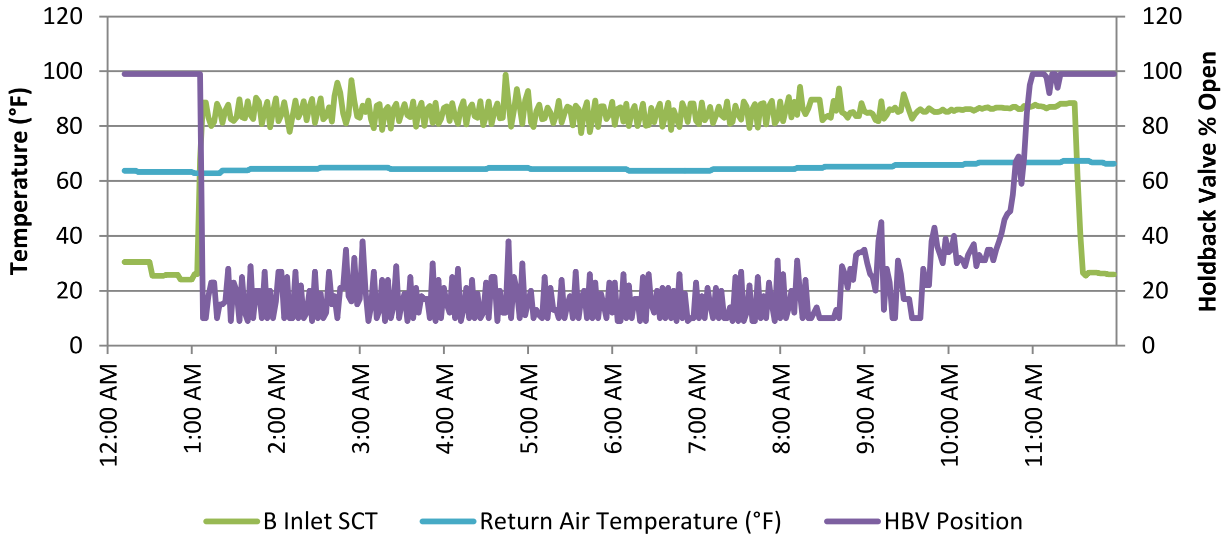


FIGURE 7: SYSTEM B RECOVERY COIL CONDENSING TEMPERATURE, MIXED AIR TEMPERATURE, AND HOLDBACK VALVE POSITION

Figure 7 shows the holdback valve position modulating to maintain a temperature difference of approximately 20°F between the holdback SCT and the mixed air temperature. The figure shows that the temperature difference is the most stable during the period from approximately 8:30 AM to 11:30 AM when the valve is mostly in the control range (e.g. between the 10% minimum open position and fully open).

REFRIGERATION CONDENSERS

The refrigeration condensers are Krack remote air-cooled units. The condensing temperature control strategy is an ambient-reset (e.g. drybulb-following) strategy with fan cycling. A target saturated condensing temperature (SCT) is established by adding a fixed control TD to the measured ambient drybulb temperature. The control TD in general is optimized so that the combined total of compressor and condenser power is as low as possible. In this case, the control TD for all four of the subject refrigeration systems is 10°F.

Analysis of the THR load and energy usage for the refrigeration condensers is presented in the sections below.

HEAT REJECTION LOAD

In the analytic Base Case scenario, all of the total heat of rejection (THR) from the associated refrigeration system compressors is rejected to the ambient by the refrigeration condensers. In the proposed case, the majority of the THR is removed by the heat recovery system whenever heat recovery is on. Figure 8 shows the sum total heat rejection load from all four remote refrigeration condensers in the Base Case and with heat recovery for a portion of the subject test period.

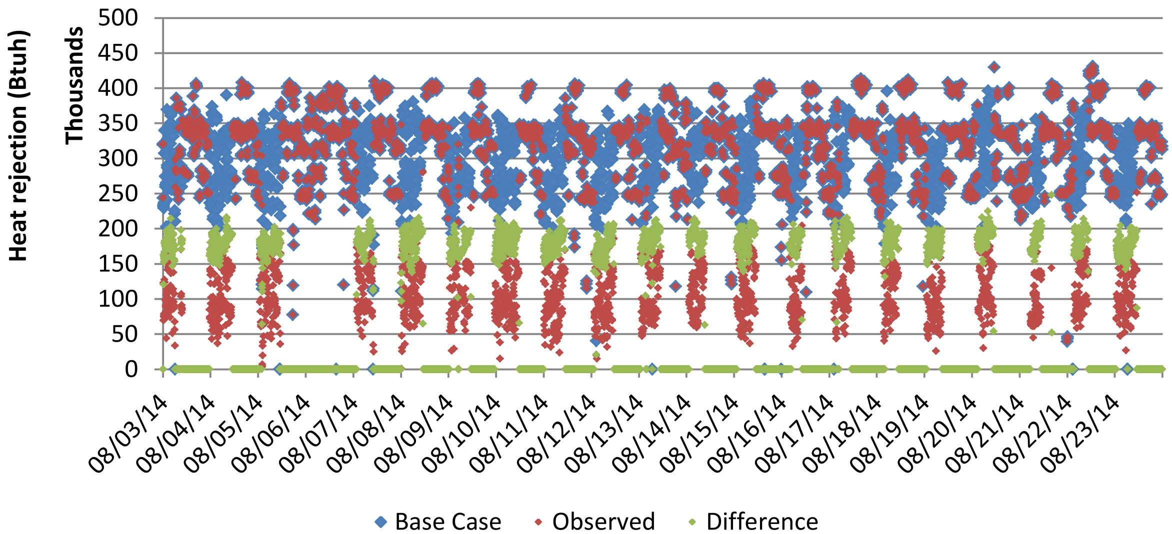


FIGURE 8: CONDENSER HEAT REJECTION FROM OBSERVATION VERSUS DOE2 SIMULATION

Table 5 below shows the average heat rejection load during the hours in which heat recovery was on for System B and C condensers for the subject test period (Systems D and E were omitted from the table since heat recovery was nearly never on for these systems).

TABLE 5: AVERAGE HEAT REJECTION LOAD DURING HEAT RECOVERY HOURS

	Base Case	Measured	Difference
System B	164.8 MBH	124.3 MBH	110.9 MBH (67%)
System C	124.3 MBH	54.7 MBH	69.6 MBH (56%)

Table 5 shows that the average THR load for the remote refrigeration condensers was reduced by over half for both systems B and C during the hours in which heat recovery was on.

ENERGY ANALYSIS

For condensers with fan cycling control, the condenser 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 THR capacity. Consequently, the condenser electric fan power is reduced when heat recovery is on because the THR load on the condenser is also reduced. Figure 9 shows the combined total condenser fan power versus time for both the Base Case and with heat recovery for a portion of the subject test period.

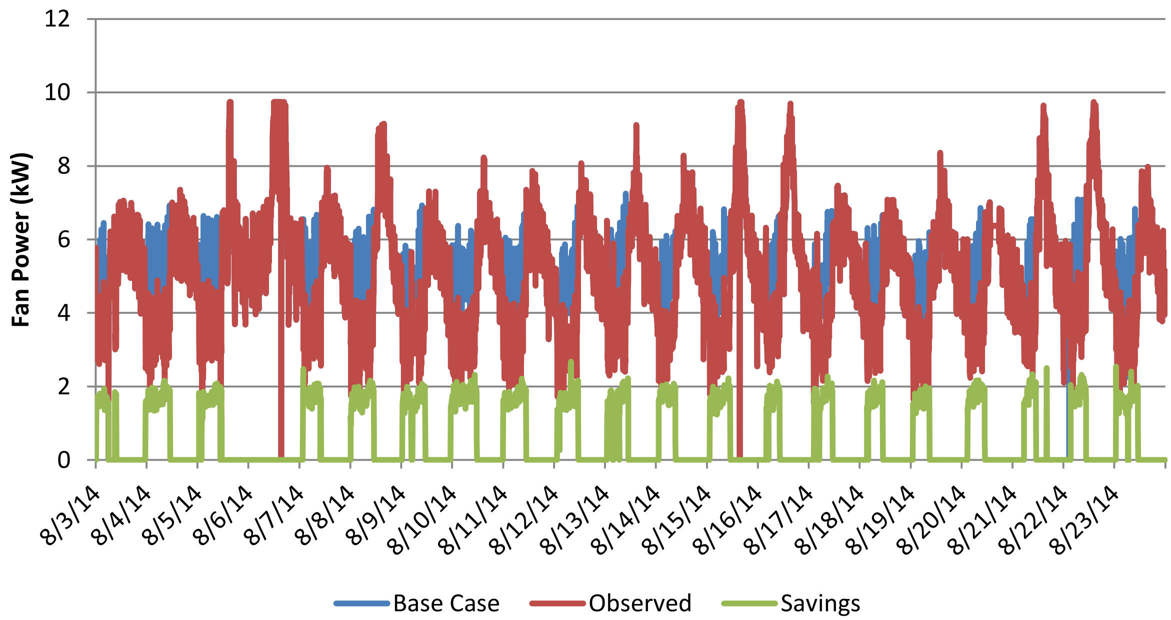


FIGURE 9: CONDENSER FAN POWER WITH HEAT RECOVERY VERSUS CALCULATED BASE CASE

Table 6 below shows the total energy usage for condenser BC during the subject test period.

TABLE 6: OBSERVED CONDENSER ENERGY USAGE DURING SUBJECT TEST PERIOD

	Base Case	Measured	Difference
Condenser BC	4,094 kWh	3,474 kWh	619 kWh (15%)

SUCTION GROUP

Analysis of the refrigeration load and energy usage for the refrigeration systems with heat recovery is presented in the sections below.

REFRIGERATION LOAD

The refrigeration load for each suction group is the basis for calculating the Base Case compressor energy usage, and is assumed to be equal in both the Base Case and with heat recovery. The refrigeration load is derived from the observed compressor capacity, using the known relationship between compressor performance and saturated suction temperature (SST), saturated condensing temperature (SCT), and return gas temperature (RGT). Table 7 below shows the total and peak refrigeration loads during the test period for the refrigeration systems with heat recovery, and Figure 10 shows the refrigeration load versus time for a portion of the subject test period.

TABLE 7: TOTAL REFRIGERATION LOAD FROM OBSERVATION VERSUS DOE2 ANALYSIS

	Total Load during Test Period (MMBtu)	Peak Load (MBH)
Suction Group B	204.9	229.6
Suction Group C	58.13	143.9
Suction Group DL	67.25	93.57
Suction Group DM	44.72	118.0
Suction Group E	177.9	210.6

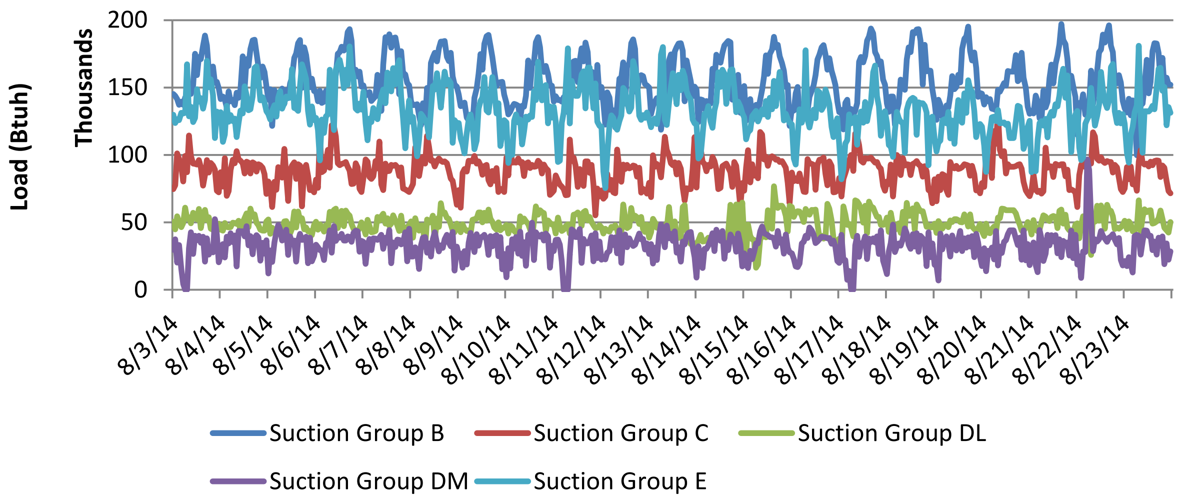


FIGURE 10: REFRIGERATION LOAD VERSUS TIME FOR SUCTION GROUPS WITH HEAT RECOVERY

ENERGY ANALYSIS

The figures below show both the actual observed suction group power usage and the calculated Base Case usage over the subject test period for Systems B and C (Systems D and E are not shown since heat recovery from those systems were not on during the test period). Figure 11 shows the power usage for Suction Group B while Figure 12 shows Suction Group C.

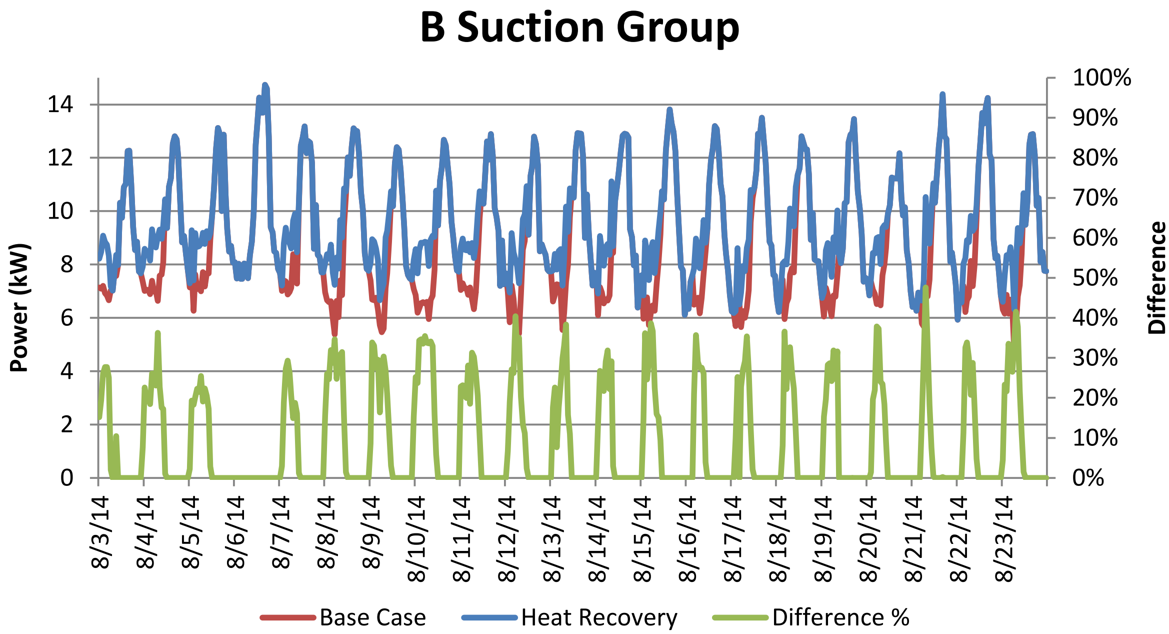


FIGURE 11: COMPRESSOR POWER FROM OBSERVATION VERSUS CALCULATED BASE CASE FOR SUCTION GROUP B

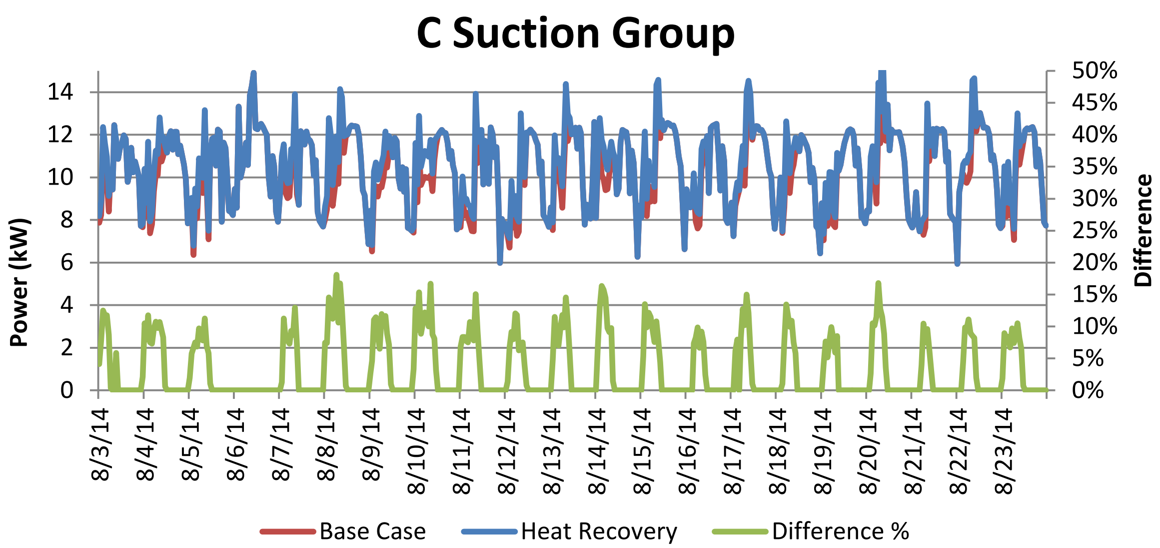


FIGURE 12: COMPRESSOR POWER FROM OBSERVATION VERSUS CALCULATED BASE CASE FOR SUCTION GROUP C

Figure 11 shows that the suction group power consumption is approximately 21% higher on average when heat recovery is on for Suction Group B, and 8% higher for Suction Group C. Table 8 below shows the overall suction group energy usage for both Suction Groups B and C during the test period.

TABLE 8: SUCTION GROUP ENERGY USAGE FOR BASE CASE AND WITH HEAT RECOVERY

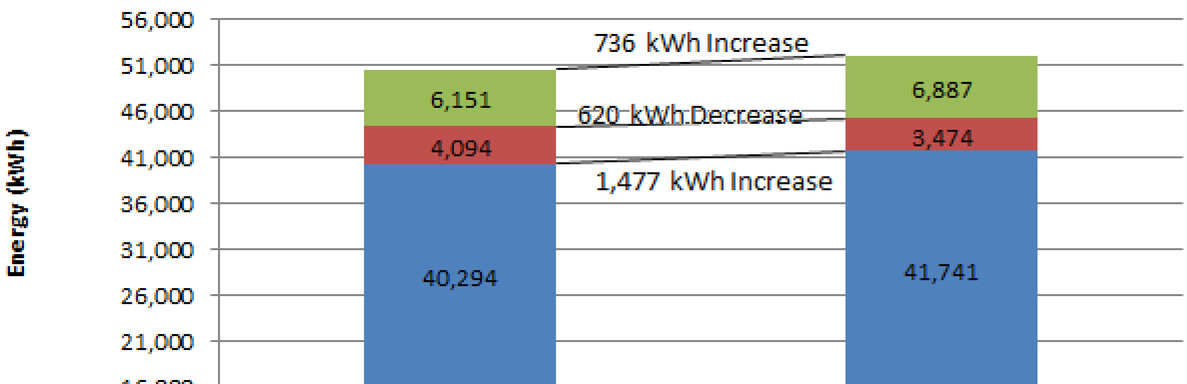
	Base Case	Measured	Difference
Suction Group B	11,719 kWh	12,514 kWh	795 kWh (6.8%)
Suction Group C	7,251 kWh	7,420 kWh	169 kWh (2.3%)
Suction Group DL	8,083 kWh	8,094 kWh	11 kWh (0.1%)
Suction Group DM	2,948 kWh	2,954 kWh	6 kWh (0.2%)
Suction Group E	10,293 kWh	10,759 kWh	466 kWh (4.5%)
Total	40,294 kWh	41,741 kWh	1,477 kWh (3.6%)

OVERALL RESULTS

The overall energy usage and cost results for the subject test period are presented in the following sections. This store is open 24 hours per day, 7 days per week.

ENERGY USE

Figure 13 below shows the electric energy usage for subject compressors, condenser fans, and the main AHU supply fan for the subject test period.

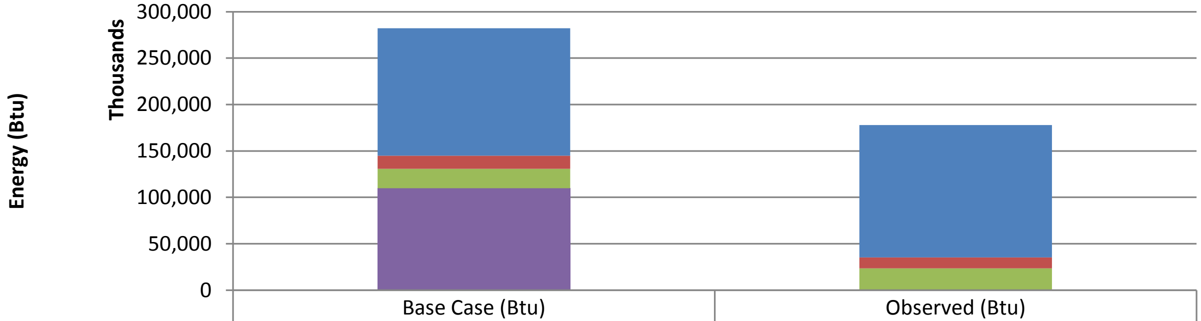


	Base Case (kWh)	Observed (kWh)
AHU Supply Fan	6,151	6,887
Condensers	4,094	3,474
Compressors	40,294	41,741

FIGURE 13: OVERALL ELECTRIC ENERGY USAGE FROM OBSERVATION VERSUS ANALYTIC BASE CASE

Figure 13 shows that, while the AHU supply fan and compressors used more electric energy with heat recovery (approximately 2,183 kWh combined) during the test period, the condenser fans used less energy (approximately 620 kWh). The overall electric energy penalty with heat recovery for the subject test period was 1,563 kWh.

Figure 14 below compares the overall energy usage, including electric energy penalty (converted to BTUs for comparison) and natural gas savings.



	Base Case (Btu)	Observed (Btu)
Compressors	137,523,422	142,462,033
Condensers	13,972,822	11,856,762
AHU Supply Fan	20,992,814	23,505,670
Natural Gas Furnace	109,829,582	0

	Savings		
	MMBtu	**% vs. Base Case**	**% of Total**
Compressors	-4.94	-3.6%	-1.75%
Condensers	2.12	15.1%	0.75%
AHU Supply Fan	-2.51	-12.0%	-0.89%
Natural Gas Furnace	110	100%	38.9%
Total	105	37.0%	

FIGURE 14: OVERALL ENERGY USAGE (ELECTRICITY AND NATURAL GAS) FOR BASE CASE AND WITH HEAT RECOVERY

ENERGY COST

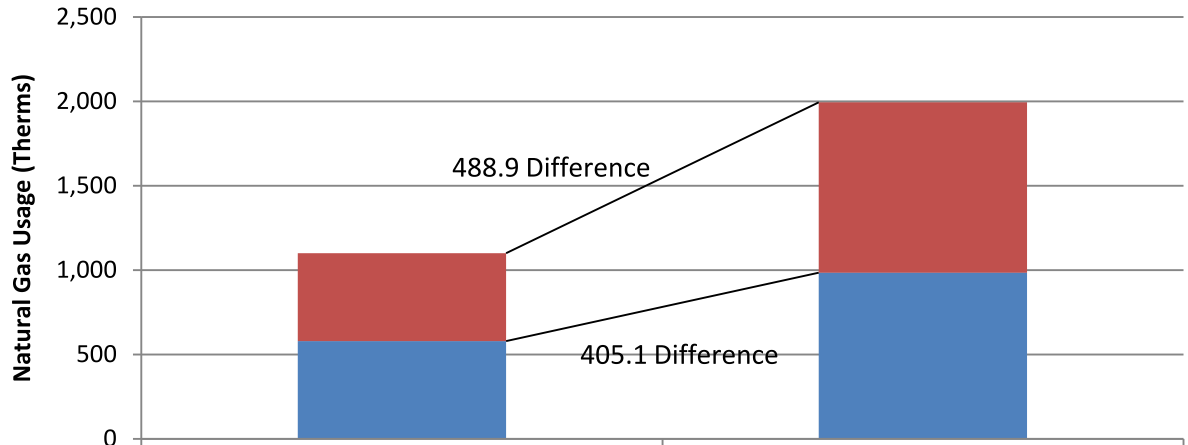
Energy cost results are shown below in Table 9.

TABLE 9: ELECTRIC ENERGY COST FROM OBSERVATION VERSUS ANALYTIC BASE CASE

	Electric Energy Cost			Electric Demand Cost			Natural Gas Cost	Total
	Peak	**Part-Peak**	**Off-Peak**	**Peak**	**Part-Peak**	**Max**		
Base Case	$1,862.89	$1,477.20	$2,652.07	$1,843.61	$384.45	$1,048.13	$779.37	$10,047.72
Heat Recovery	$1,876.69	$1,512.02	$2,743.08	$1,885.12	$390.44	$1,071.73	$0	$9,479.08
Difference	$(13.81)	$(34.82)	$(91.01)	$(41.51)	$(5.99)	$(23.60)	$779.37	$568.64

COMPARISON TO DOE2 ANALYSIS

A detailed analysis of the heat recovery system was performed during the project design phase using a calibrated DOE-2.2R whole-building yearly energy model. The natural gas savings for the main air handling unit from the energy model were compared to the observed savings during the subject test period, and are shown in Figure 15 below.



	Observed (Therms)	DOE2 Analysis (Therms)
September	520.2	1,009.1
August	579.9	985.0

FIGURE 15: NATURAL GAS SAVINGS FOR SUBJECTTEST PERIOD COMPARED TO DOE-2.2R ANALYSIS MODEL

Figure 15 shows that the observed natural gas savings for the subject test period are less than the DOE-2.2R-modeled natural gas savings. For the months of August and September, the simulated natural gas savings were 70% and 94% higher than the observed values, respectively.

The difference in savings may be attributed to a number of different factors. The DOE-2.2R analysis was performed before the new store was constructed, and was calibrated using utility data from three other stores owned by the same grocery store chain in the San Francisco Bay Area and California central coast regions. Moving forward, the DOE2 model calibration will be checked as more data is collected, particularly as ambient conditions start to get cooler.

CONCLUSIONS AND NEXT STEPS

For the first 61 days of testing, the heat recovery system at the new store saved 1,098 Therms of natural gas, and resulted in a refrigeration system electrical energy increase of 1,563 kWh. The resultant energy cost savings was $639.18.

The performance of the heat recovery system will continue to be monitored through February, 2015. Heating demand and expected energy savings is expected to steadily increase as the season transitions from hot summer weather to the cooler winter months. Topics that will be investigated in the coming months will include:

- Evaluation of the furnace efficiency with and without heat recovery to determine if the heat recovery system negatively effects the thermal efficiency of the furnace
- Performance of week on/week off tests of the heat recovery system to validate Base Case assumptions
- Continue to compare actual system performance to expectations from DOE2 analysis

The final Emerging Technologies evaluation report will be issued in March, 2015. The report will include a comprehensive analysis of system performance, including performance during cold-weather months, a comparison of actual performance to expectations from energy modeling, analysis of refrigerant charge impacts from heat recovery systems, and a parametric analysis of heat recovery systems in other climate zones using a calibrated DOE2 energy model will be conducted at the conclusion of the overall data collection period.