FINAL REPORT

DEMONSTRATING A COMBINED HEAT AND POWER (CHP) SYSTEM:
AN INTEGRATED MICROTURBINE/DESICCANT SYSTEM FOR SUPERMARKET APPLICATIONS

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NOTICE

EXECUTIVE SUMMARY

This report summarizes the field-monitored performance results from a Combined Heat and Power (CHP) system installed at a Waldbaums Supermarket in Hauppauge, New York. The system used a 60 kW Capstone microturbine to generate electricity along with a Unifin Heat Exchanger (HX) to recover heat from the turbine exhaust. The Unifin HX provided heat recovery for either space heating or desiccant dehumidification, depending on the season. This supermarket was a cost-effective application since the CHP equipment could be mounted on a rooftop skid next to the store's main Air-Handling Unit (AHU). The AHU provided heating, cooling, and dehumidification for the entire facility.

A data acquisition system was installed to monitor the performance of the CHP system with funding from NYSERDA. Oak Ridge National Laboratory (ORNL) and the National Renewable Energy Laboratory (NREL) also provided funding to install additional instrumentation to evaluate the microturbine and desiccant components. Detailed data were collected at 15-minute intervals from August 19, 2002 to June 9, 2004. The system was also tested as part of the Environmental Test and Verification (ETV) Program run by the Environmental Protection Agency (EPA). NYSERDA contracted with the Southern Research Institute (SRI) to take detailed, high accuracy emission and energy readings at the site for a short period in June 2003. This high precision data was used to verify the long term field monitored data collected in this report.

The CHP system started operating in April 2003. During the first 12 months of operation the microturbine and Unifin HX had various problems that caused it not to operate for extended periods. Some faults were minor while others required major repair of the turbine or HX unit. Overall the CHP system operated 54% of the time during the 12-month period ending in March 2004. All maintenance and repair has been provided by Capstone. During last four months (June through September 2004), the system has operated continuously with only minor shutdowns due to grid disturbances and power outages.

The measured efficiency and power output of the microturbine was found to be in line with the manufacturers performance expectations at the ISO rating condition. However, the microturbine's power output and efficiency were found to drop off faster than expected at higher ambient temperatures. When a new turbine engine was installed in December 2003, the efficiency of the microturbine increased by at least one percentage point. The efficiency of the new turbine engine was also higher than rated performance at colder outdoor conditions.

The overall CHP efficiency of the system – considering parasitic power use of the compressor and pumps as well as the useful heat recovery provided by the system – ranged from more than 60% based on higher heating value (HHV) on cold winter days to over 50% HHV on humid summer days. Net electrical efficiency was as low as 21% HHV on hot summer days. Space heating heat recovery was less than expected due to the small differential between the gas furnace and heat recovery coil set points. More ideal heat recovery control settings would have resulted in daily CHP efficiencies over 70%.

The measured performance data trends for the CHP system components and building loads were combined with typical meteorological year (TMY) weather data and utility tariffs to predict

energy use, efficiency and cost savings of the system for a full year operation. The hourly model was also used to understand the impact of system hardware configurations, control scenarios and utility rate options. The base system installed at the store had a CHP efficiency of 38% across the year. Total gas savings from CHP heat recovery totaled nearly 12,000 therms per year. If the measured performance trends for new turbine engine were assumed, and the heat recovery controls for space heating are assumed to be closer to ideal, then the CHP efficiency increases to 52% for the year. Displaced gas use due to heat recovery exceeds 24,000 therms per year in this scenario. The net cost savings for this more ideal CHP system after including maintenance costs exceeds $5,300 per year. The model was also used to investigate various other scenarios:

- decreasing gas commodity costs by $0.10 per therm increases annual savings by $4,500 per year,
- Operating the turbine only during the day (7 am to midnite) from April to October increases net savings by nearly $2,500 per year,
- Using the rated performance specifications for the Capstone resulted similar net savings compared to the measured turbine performance trends.

The model was also used to estimate annual savings using utility rates and weather data for other locations around the US. Net cost savings in Consolidated Edison territory increased to $18,800 per year. Similar savings were realized in Southern California. Savings in Chicago were slightly less than $12,000 per year. In contrast, Portland, Oregon resulted in a net loss exceeding $20,000 per year.

The ETV testing and follow-up emissions testing by CDH showed that the microturbine exceeded its emissions specifications. The NO_x emissions from the microturbine were 3 to 5 ppmv (@ 15% O_2) at full load. An annual emissions evaluation using the model described above predicted that the CHP system lowered net NO_x emissions from the site by more than 1,300 lb per year (or 11%) compared to the onsite burners and local utility power plants that serve the facility. CO_2 emissions were reduced by more than 300,000 lb per year (or 2.7%).

Based on the installed cost of $147,000 (or $2,450 per kW) the system has a 30 year payback under the current LIPA rates in Hauppauge. In Consolidated Edison territory, the payback drops to 8 years.

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1 INTRODUCTION

1.1 OVERVIEW

This report summarizes the field-monitored performance results from a Combined Heat and Power (CHP) system installed at a Waldbaums Supermarket in Hauppauge, New York. The system used a 60 kW Capstone microturbine to generate electricity along with a Unifin Heat Exchanger (HX) to recover heat from the turbine exhaust. The Unifin HX transferred heat to a glycol loop that delivered heat for either space heating or desiccant dehumidification depending on the season. This supermarket was a cost-effective application since the CHP equipment could be mounted on a rooftop skid next to the store's main Air-Handling Unit (AHU). The AHU provided heating, cooling, and dehumidification for the entire facility. Heat recovery coils for both space heating and desiccant drying were added to the AHU at factory.

A data acquisition system was installed to monitor the performance of the CHP system with funding from NYSERDA. Oak Ridge National Laboratory (ORNL) and the National Renewable Energy Laboratory (NREL) also provided funding to install additional instrumentation to evaluate the microturbine and desiccant components. Detailed data were collected at 15-minute intervals from August 19, 2002 to June 9, 2004. Basic monitoring of microturbine gas input and power output are still ongoing.

This site was also selected by NYSERDA to participate in the Environmental Test and Verification (ETV) Program run by the Environmental Protection Agency (EPA). NYSERDA contracted with the Southern Research Institute (SRI) to take detailed, high accuracy emission and energy readings at the site for a short period in June 2003. This high precision data was used to verify the long term field monitored data collected in this report.

1.2 PROJECT OBJECTIVES

The goal of this project was to collect detailed monitored data on the performance on a microturbine-based CHP system in an actual supermarket application. The monitoring effort was focused on understanding the economic and environmental benefits of this technology in this application. Data were collected to quantify the performance of the CHP system and its individual components as well as to understand the electrical and thermal loads in the facility. The specific objectives were to:

- Quantify the variation of microturbine output, gas consumption, and efficiency over wide range of operating conditions; compare measured performance to manufacturer's ratings;
- Quantify heat recovery performance of other components in the system and compare to ratings;
- Measure parasitic loads (e.g., gas compressor, Unifin pump, etc.);
- Measure emissions rates from microturbine and other equipment at the site to quantify environmental benefits of CHP system;

- Quantify the thermal and electrical loads imposed on the CHP system by this application; quantify the variation of these loads with ambient conditions so that the findings from this site can be extended to other locations and utility rates around the US.

2 SITE AND CHP SYSTEM DESCRIPTION

The host site is a new supermarket that was originally a 35,000 sq ft retail facility. The building was gutted to the block walls, expanded, and totally rebuilt into to 57,000 sq ft supermarket. The store is located at 1235 Veterans Memorial Parkway on Long Island in Hauppauge, New York. It opened in July 2002. The store is open 24 hours per day for all days of the week except Sunday. The store uses energy-efficient T4 light fixtures, so the light load in the sales area is about 1.2 Watts per square foot. The peak demand for the total facility is in the 400-600 kW range. The demand never drops below 100 kW in this store. Figure 1 shows the front of the supermarket.

2.1 CHP SYSTEM

A Capstone 60 kW microturbine was integrated with a Munters AHU that was originally part of Waldbaums standard store design. The Munters AHU provided cooling and heating to the main sales areas of the store. The AHU also included a gas-fired desiccant wheel to provide dehumidification. A Unifin HX was installed to recover heat from the microturbine exhaust; that heat was used to provide either space heating or dehumidification. The glycol piping from the Unifin was directly connected to two hot water coils in the Munters unit that supply either space heating or air preheating for desiccant regeneration. The Munters unit was configured to use recovered heat, when available, or use the conventional natural gas-fired furnace sections/ regeneration burners when the CHP system did not operate for any reason. The main CHP components and the Munters AHU are shown in Figures 1, 2, and 3. More details about the system installation and design and are given in Appendix D.

Front of Supermarket

CHP Skid Located on Roof - Front

Figure 1. Waldbaums – Veteran's Memorial Highway, Hauppauge, NY

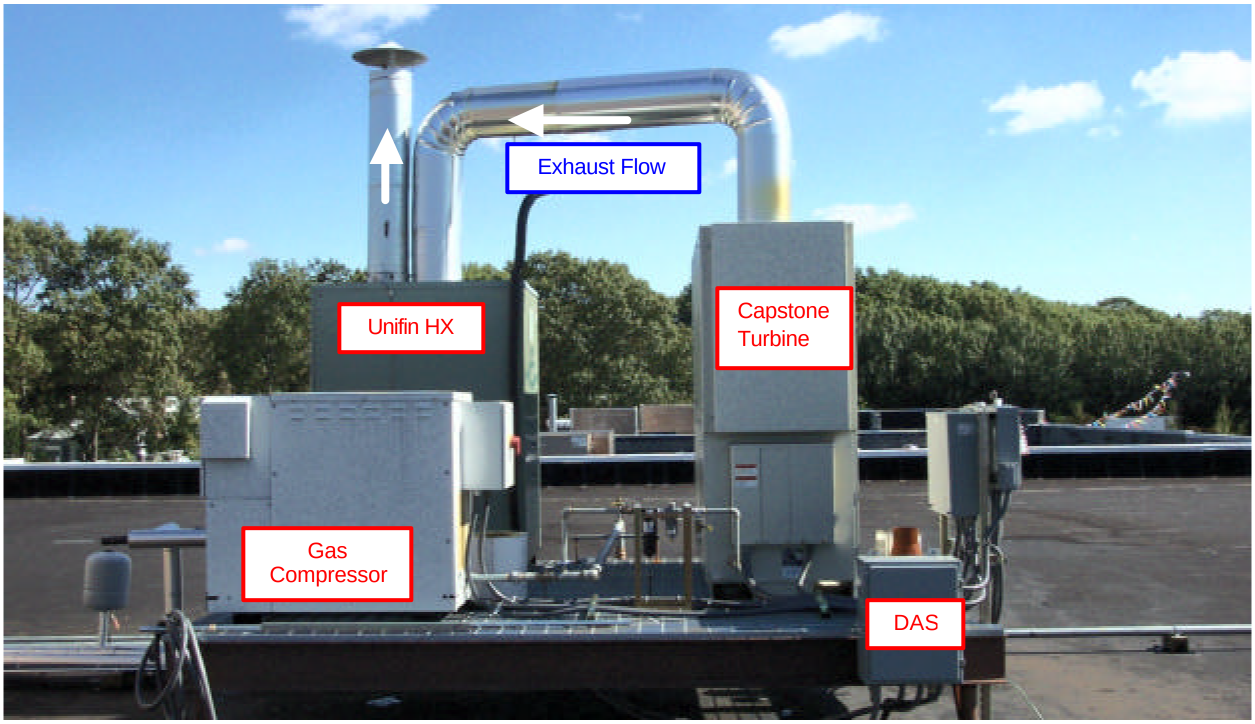


Figure 2. Completed Microturbine CHP Skid at Waldbaums

Figure 3. AHU and Microturbine Skid (During construction, before screens installed on Munters AHU)

Figure 4 schematically shows the layout of the Munters AHU. The factory-added hot water coils were added to the AHU to provide space heating and desiccant regeneration. The gas-fired burner in the desiccant section heats outside air and supplies it to one side of the desiccant wheel to regenerate the desiccant material. Then the wheel rotates into the process side of the system and removes moisture from the mixed air stream. The dried and heat air from the desiccant section then mixes back into the mixed air plenum where it goes on through the unit for further cooling and heating.

The space heating coil was installed before the supply and furnace section to heat mixed return and ventilation air. The regeneration hot water coil was added to outside of the regeneration section to preheat ambient air entering the direct fire burner. The desiccant wheel regeneration temperature was about 275°F, so the 200°F glycol loop was only capable of preheating air entering the burner to about 180°F.

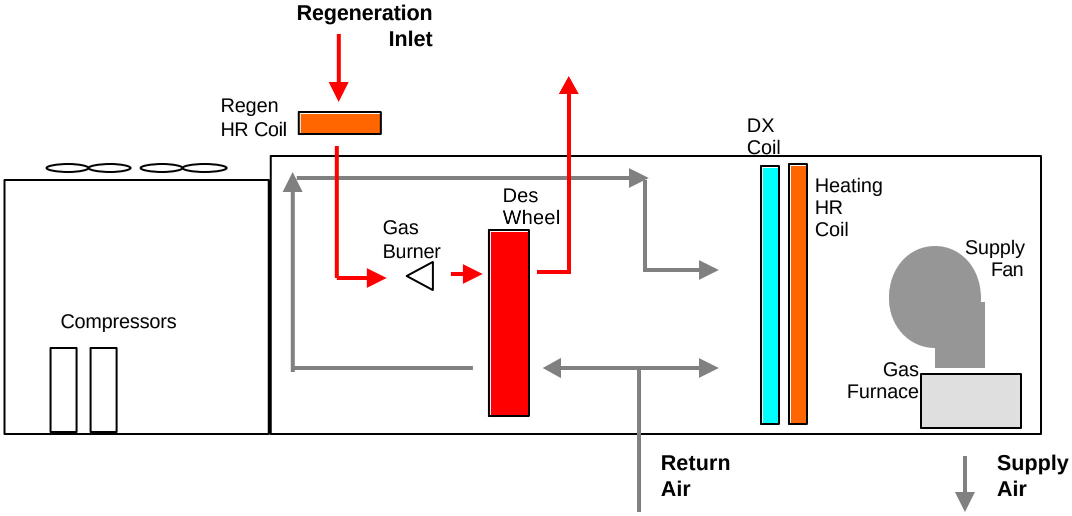


Figure 4. Air Flow Schematic of Main AHU

2.2 SYSTEM CONTROLS

The CHP system was integrated into the building controls so that the system could provide heating as required to meet the loads. The glycol loop was configured with a three-way valve[1] so that heat could be supplied to either the space heating coil or the regeneration coil as shown in Figure 5. The valve controls were set to direct glycol to the outdoor-mounted regeneration coil by default. This allowed the Unifin glycol pump, which was required to run continuously, to passively reject any heat that built up in the loop during swing seasons when no heating was required.

[1] Actually two interlocked two-way valves were installed instead of a single three-way valve.

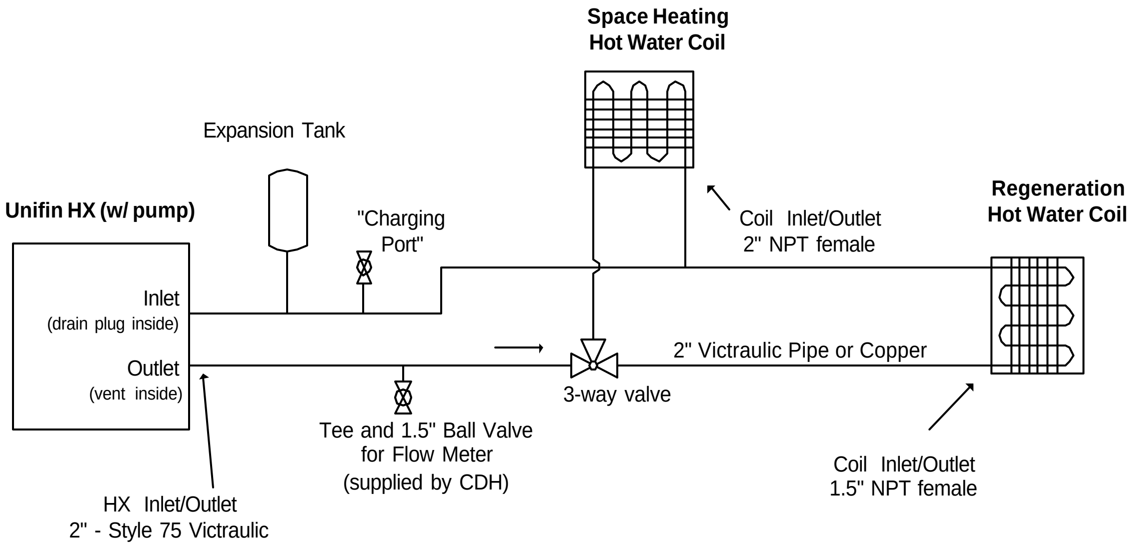


Figure 5. Schematic of Glycol Piping

Upon a call for desiccant dehumidification, the regeneration fan started and pulled air through the hot coil. The Unifin cycled its internal bypass damper to maintain the specified temperature set point for the glycol loop.

Upon a call for space heating, the valve diverted glycol flow to the space heating coil inside the AHU. The valve was controlled by the store's Danfoss control system that also controls the other functions in the Munters AHU. The valve was controlled based on the space temperature. The control set point was set about 1°F lower than the first stage heating set point of the gas furnace section. This bias allowed heat recovery to meet more of the heating load before the gas furnace section was activated.

The Unifin HX also had on-board control functions to maintain the glycol loop temperature and shut down if various faults occur. The Unifin's main control function is cycling the exhaust bypass damper to maintain the glycol loop within the specified temperature limits (in this case 180-200°F). When there was no heat recovery load in the swing season, the bypass damper would spend most of the time open and only close when the passive heat rejection from the loop would push the temperature below the set point. In the space heating mode the damper was continuously closed since the glycol loop continuously operated at about 120-160°F (well below the damper set point). The Unifin also had control features that shut the HX unit down if the Capstone shut down or if glycol or exhaust temperatures exceeded a certain level. More details on the control system are given in Appendix D.

2.3 CHP PROJECT INSTALLATION COSTS

The CHP system was installed in the Spring of 2002 before the new store was fully completed but after the contractors had been selected in a competitive bidding process. Therefore, the costs to install the CHP system were somewhat higher than expected. Table 1 itemizes the costs to install the system. Installation costs were $147,000 or $2,450 per nominal kW. The cost of the CHP hardware alone was about half of the total project costs (or $1,280 per kW). We estimate that total installation costs would have been about $30,000 to $40,000 lower if the CHP system had been on the store drawings and included as part of the original bid package. This would have dropped the total installed cost to above $1,800 to $2,000 per nominal kW.

Table 1. Summary of CHP System Installation Costs at Waldbaums Supermarket

Item	Cost
Capstone Microturbine (C60), Gas Compressor (510447-001), and Unifin HX (MG2-C2H2)	$70,000
Heat recovery coils added to Munters AHU (at Factory)	$7,000
Structural steel platform, roof patching	$15,000
Crane (to lift components in place)	$5,000
Electrical (for Turbine, Gas Compressor, & Unifin	$8,000
Plumbing (additional gas meter and piping from rear of store to roof; glycol piping and valves from Unifin to Munters AHU)	$32,000
Control Upgrade from Danfoss Control System	$5,000
Management by General Contractor	$5,000
TOTAL	**$147,000 ($2,450 per kW)**

Installation costs outside the New York Metropolitan area would have been 25-50% lower while equipment costs would have stayed about the same. We estimate that the installed total system costs out of the NY area would have been in the range of $1,400 to $1,600 per nominal kW.

2.4 ELECTRICAL INTERCONNECTION WITH LOCAL UTILITY

The CHP system was completed and fully commissioned in the Summer of 2002. However, the system was not approved for operation by the local utility until April 2003. The system met all of the technical requirements for interconnection because the Capstone C60 had been "type tested" in early 2002 and was certified by the Public Service Commission as meeting the Standard Interconnection Requirements (SIR) in New York State. The problem was with the contractual and administrative requirements of LIPA's version the SIR application.

The supermarket was not owned by Waldbaums but by a third party landlord. The mall facility also had other tenants. A single electrical feed from LIPA served the transformer for Waldbaums as well as a second transformer for the other tenants in the mall. This "non-radial" feed from LIPA did not strictly meet the requirements of the SIR. As a result, LIPA wanted the SIR agreement to be signed by the landlord instead of Waldbaums. Alternatively, LIPA would require primary-side fusing be added to the Waldbaums transformer at a cost of $40,000 to the customer. While the LIPA engineers agreed that adding primary-side fusing would not provide any additional safety protection, the requirement was driven by LIPA's administrative rules.

After considerable negotiation, the final agreed-upon solution was to have the landlord sign the SIR along with the Keyspan Energy R&D division. The agreements were signed and LIPA engineers approved the installation in April 2003.

A full report on the technical and administrative interconnection issues for the Waldbaums site is available at www.cdhenergy.com/waldbaum_haupauge/A&P interconnection report.pdf

3 DATA ACQUISITION SYSTEM

A data acquisition system (DAS) was installed at the site to measure the performance of the CHP system. Sensors were installed to quantify CHP component performance, record parasitic energy use and determine building loads. Several diagnostic points were also added to help understand the performance details of certain CHP and HVAC components. The monitored data points associated with the CHP skid are listed in Table 2 and schematically shown in Figure 6. A Campbell Scientific CR10X data logger was used to capture these data points. Appendix A provides more details on the DAS and instrumentation used at the site.

Table 2. List of Monitored Points on the CHP Skid

Data Pt No.	Data Pt Name	Description	Eng. Units
1	TEXH1	Temperature of Turbine Exhaust	F
2	TEXH2	Temperature of Unifin Exhaust	F
3	PEXH	Static Pressure, Turbine Exhaust	in H2O
4	VEXH	Exhaust Gas Velocity	in H2O
5	TGL	Glycol Temperature Leaving Unifin	F
6	TGE	Glycol Temperature Entering Unifin	F
7	FGLY	Glycol Flowrate	gpm
8	IUP	Glycol Pump Current	amps
9	WU	Utility Meter Power (kW, Amps, Volts)	kWh
10	WT	Turbine Power Output (kW, Amps, Volts)	kWh
11	FGT	Capstone Turbine Gas Use	cf
12	WGC	Gas Compressor Power	kWh
13	SV	Status, Glycol Control Valve	minutes

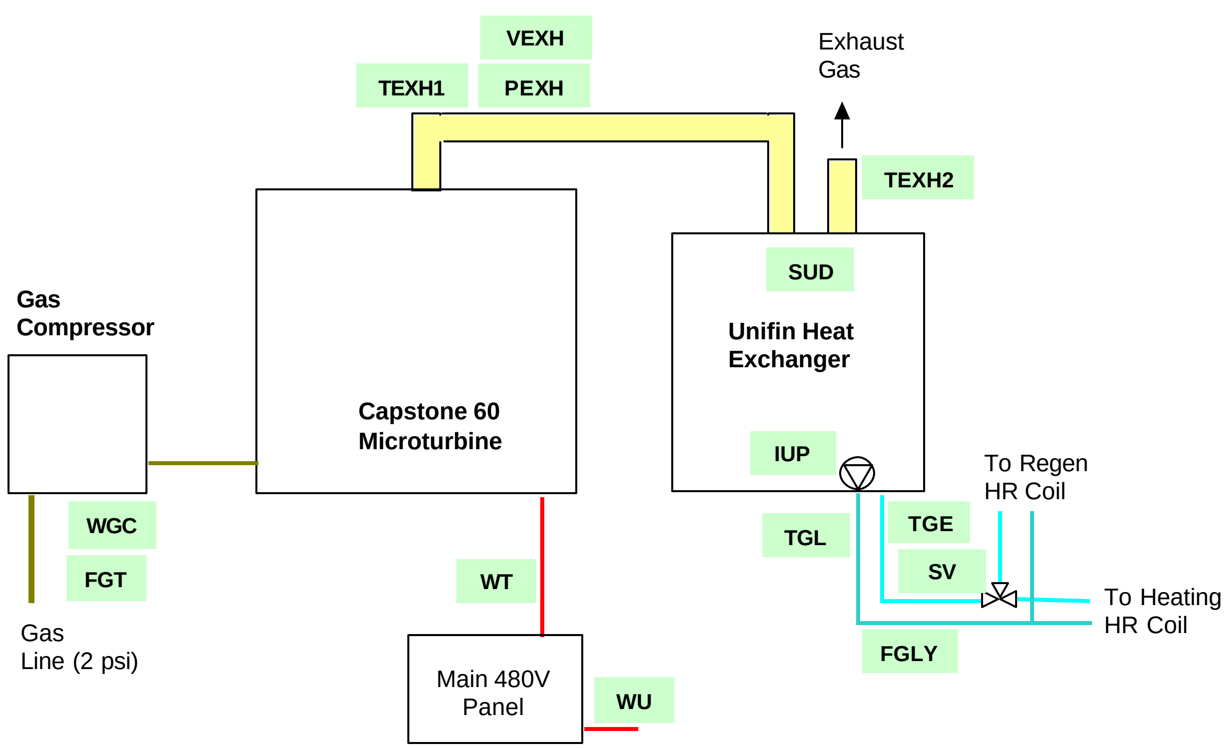


Figure 6. Schematic Location of Monitored Points on CHP Skid

A separate data logger was used to power and power quality data at the store's main distribution panel (MDP). Modbus-based Veris power transducers were used with an E-Server datalogger to record power, current and volts for the total store and the microturbine output. The monitored points are listed in Table 3. The conductors feeding the panel were arranged such that two separate power transducers (WB1 and WB2) were required to record the total store power. Data were collected at 15-minute intervals.

Table 3. List of Monitored Points Measured at Store MDP

Data Pt No.	Data Pt Name	Description	Eng. Units
1	WT	Microturbine Output (Energy)	kWh
2	WT_kW	Microturbine Demand	kW
3	WT_kVA	Microturbine Apparent Power	kVA
4	Vab	Microturbine Voltage Line A to B	V
5	Vbc	Microturbine Voltage Line B to C	V
6	Vac	Microturbine Voltage Line A to C	V
7	WT_Ia	Microturbine Current Line A	amps
8	WT_Ib	Microturbine Current Line B	amps
9	WT_Ic	Microturbine Current Line C	amps
10	WT_kWa	Microturbine Power Line A	kW
11	WT_kWb	Microturbine Power Line B	kW
12	WT_kWc	Microturbine Power Line C	kW
13	WB1	Total Store Energy - Meter 1	kWh
14	WB_kW1	Total Store Demand - Meter 1	kW
15	WB_kVA1	Total Store Apparent Power - Meter 1	KVA
16	WB_I1	Total Store Current (avg per phase) - Meter 1	Amps
17	WB2	Total Store Energy - Meter 2	KWh
18	WB_kW2	Total Store Demand - Meter 2	KW
19	WB_kVA2	Total Store Apparent Power - Meter 2	KVA
20	WB_I2	Total Store Current (avg per phase) - Meter 2	Amps

Figure 7 shows the monitored points located inside the Munters AHU. Table 4 lists these points. The Campbell Scientific CR10X data logger was also used to capture these data points. The data logger used an analog multiplexer to record all these values. The datalogger was installed on the CHP skid. It was programmed to collect at 15-minute intervals. The instrumentation used for each monitored point and details of datalogger programming and wiring are given in Appendix A.

The DAS and all instrumentation was installed and verified in August 2002. Sensors were periodically verified throughout the monitoring period by comparing them to handheld meters and other reference readings. Many one-time readings of air flow, power, and other parameters of interest were periodically collected. All these verification and calibration readings are included in Appendix A along with one time readings.

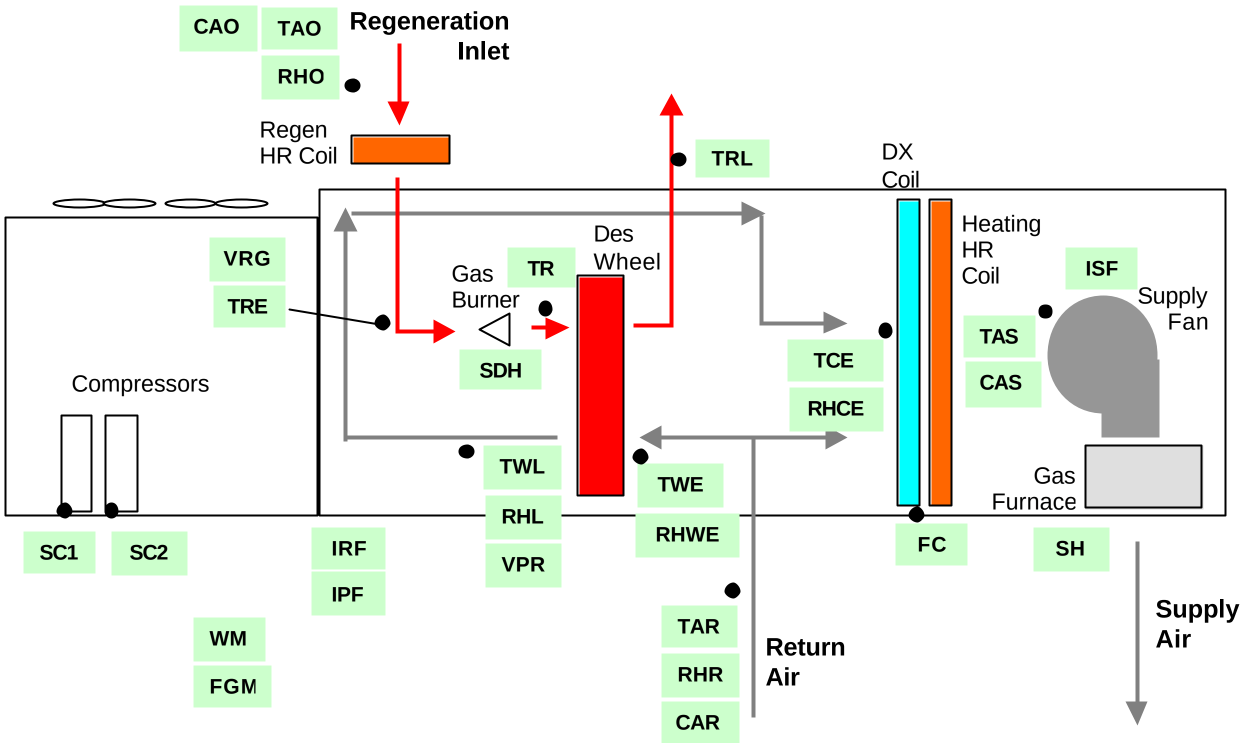


Figure 7. Schematic Location of Monitored Points in Main AHU

Table 4. List of Monitored Points Inside the Munters AHU

Data Pt No.	Data Pt Name	Description	Eng. Units
1	TAO	Outdoor Air Temperature	F
2	RHO	Outdoor Humidity	%RH
3	TAR	Temperature Return Air to Munters	F
4	RHR	RH Return Air to Munters	%RH
5	TCE	Temperature Entering DX/HR Coil	F
6	TAS	Temperature Supply Air from Munters	F
7	TRE	Temperature Entering Regen Burner	F
8	TR	Regen Temperature Entering Des Wheel	F
9	TRL	Regen Temperature *Leaving* Des Wheel	F
10	TWE	Temperature Entering Des Wheel	F
11	RHWE	RH Supply Air from Munters	%RH
12	TWL	Temperature Leaving Des. Wheel	F
13	RHWL	Absolute Humidity Leaving Des Wheel	%RH
14	VPR	Process Air Velocity	in H2O
15	VRG	Regeneration Air Velocity	in H2O
16	ISF	Supply Fan Current	amps
17	IPF	Process Fan Current	amps
18	IRF	Regen Fan Current	amps
19	CAR	CO_2 Concentration in Return Duct	ppm
20	CAS	CO_2 Concentration in Supply Duct	ppm
21	CAO	CO_2 Concentration of Outdoor Air	ppm
22	FGM	Munters Unit Gas Use	cf
23	WM	Munter Unit Power Use	kWh
24	FC	DX Coil Condensate Drain	lb
25	SC1	Status, Munters Compressor, Stage #1	minutes
26	SC2	Status, Munters Compressor, Stage #2	minutes
27	SDH	Status, Munters Burner/Process Fan	minutes
28	SH1	Status, Munters Heat Section, Stage 1	minutes
29	SH2	Status, Munters Heat Section, Stage 2	minutes

4 MEASURED PERFORMANCE

Detailed data collection began in August 2002 and concluded in June 2004. The Veris data logger remains at the site collecting power data. The microturbine did not start to operate until April 2003 when the interconnection issues with the local utility were resolved..

4.1 FACILITY AND CHP SYSTEM ENERGY USE

Table 1 summarizes the store energy use and turbine power output over the 24-month period through July 2004. Monthly energy and demand are given for the total facility as well as for the energy purchased from the utility. Micoturbine operation was intermittent for the period (operational issues are discussed in the next section). Appendix B includes tables of daily electric output for the turbine.

Figure 8 shows the impact of the microturbine on store power consumption for July 5, 2003, a hot summer day with constant turbine activity. On this day, the ambient temperature rose to a high of 95°F in the mid-afternoon with a low of 75°F in the early morning. The turbine operated continuously throughout the day with power output dropping at higher ambient temperatures. Both the turbine power output (WT) and the purchased (or imported) power from utility (WB1 & WB2) are measured by the data logging system. When the turbine operates, the total store energy use (the dashed line on the plot) is determined by summing the turbine output and the utility import.

Table 5. Summary of Facility Electricity Use and Turbine Output

Month	Utility Import Energy (kWh)	Turbine Generated Energy (kWh)	Total Facility Energy (kWh)	Utility Imported Demand (kW)	Net Facility Demand (including turbine) (kW)
Aug-02	240,281	250	240,531	482	482
Sep-02	225,624	10	225,633	444	444
Oct-02	192,960	320	193,280	421	421
Nov-02	166,070	0	166,070	311	311
Dec-02	164,049	0	164,049	301	301
Jan-03	167,012	9	167,021	277	277
Feb-03	152,590	0	152,590	287	287
Mar-03	172,511	101	172,612	311	311
Apr-03	149,853	14,477	164,330	346	346
May-03	157,098	30,127	187,225	344	344
Jun-03	167,028	38,028	205,056	442	445
Jul-03	209,359	39,185	248,544	418	464
Aug-03	221,782	11,100	232,883	438	438
Sep-03	175,895	21,032	196,926	375	407
Oct-03	143,002	32,316	175,318	320	336
Nov-03	160,465	9,768	170,234	319	369
Dec-03	159,621	5,352	164,973	299	299
Jan-04	127,775	34,703	162,478	242	274
Feb-04	125,101	27,701	152,802	253	261
Mar-04	146,947	35,160	182,107	329	329
Apr-04	178,739	3,471	182,211	323	323
May-04	204,923	4,488	209,411	421	421
Jun-04	172,275	40,892	213,167	420	471
Jul-04	182,485	40,682	223,166	370	425
Aug-02 to Jul-03	***2,164,433***	***122,507***	***2,286,940***	***482***	***482***
	95%	***5%***	***100%***		
Aug-03 to Jul-04	***2,057,672***	***266,666***	***2,324,338***	***438***	***471***
	89%	***11%***	***100%***		

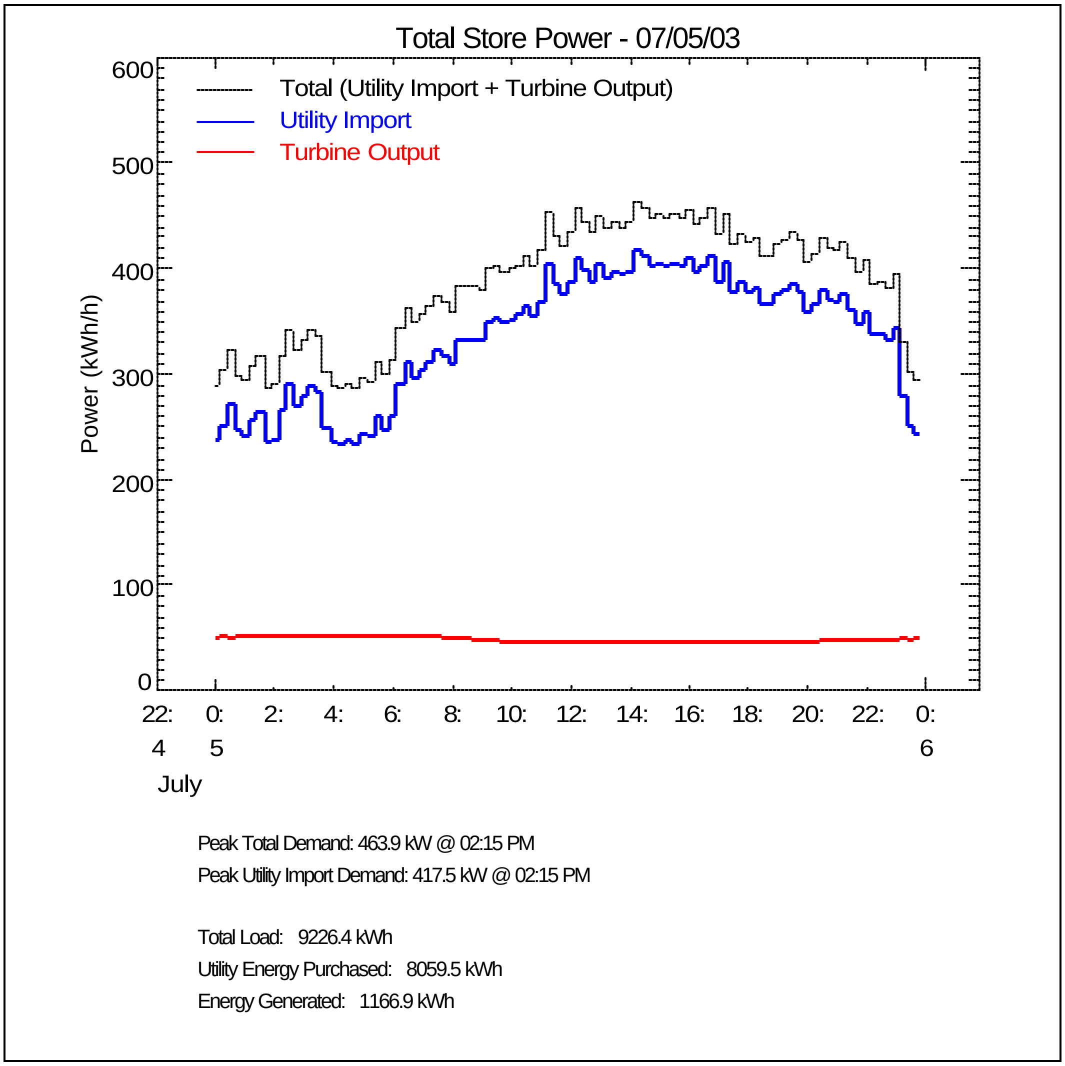
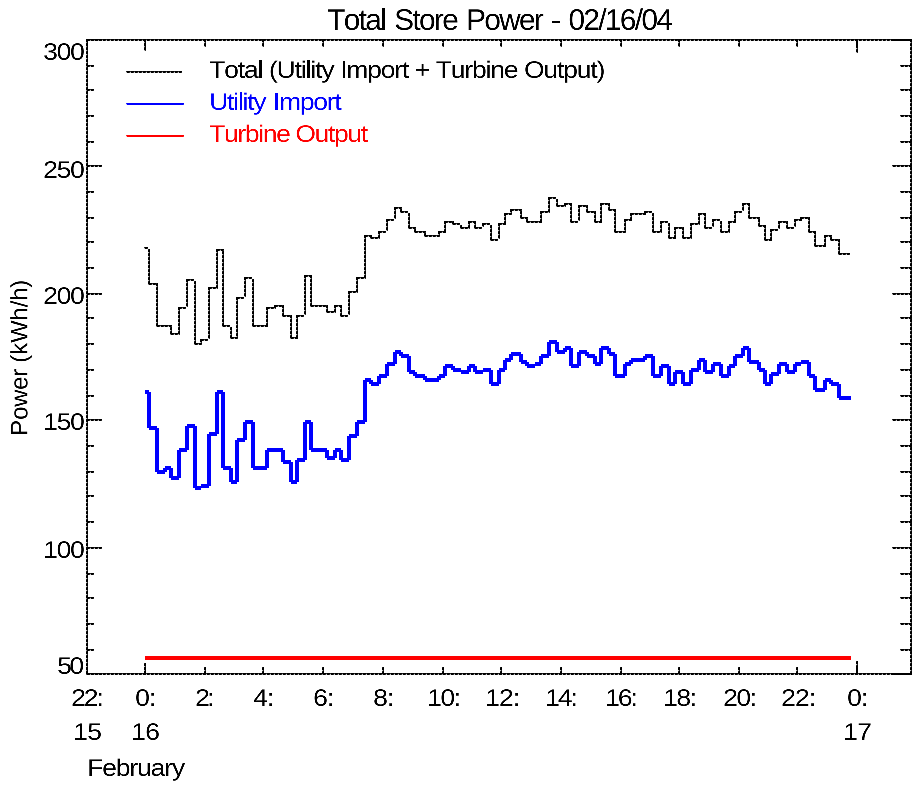


Figure 8. Impact of Turbine Operation on Purchased Utility Power for Hot Summer Day – July 5, 2003

Figure 9 shows the daily power profile for February 16, 2004, the cold winter day with constant turbine activity during the monitoring period. On this day the temperature reached a low of 15°F in the early morning and a high of 35°F in the middle of the day. Power production remained constant at 57 kW throughout the day.



Peak Total Demand: 238.1 kW @ 01:45 PM
Peak Utility Import Demand: 181.4 kW @ 01:45 PM

Total Load: 5204.5 kWh
Utility Energy Purchased: 3853.9 kWh
Energy Generated: 1350.5 kWh

Figure 9. Impact of Turbine Operation on Purchased Utility Power for Cold Winter Day – Feb. 16, 2004

Figure 10 shows the impact of the microturbine on the store's electric load line. The load line shows the expected variation of daily store energy with ambient conditions. The change in slope is due to cooling and refrigeration load variation with ambient temperature. Without the microturbine operating, the store uses between 5,000 and 5,650 kWh/day when the ambient temperature is below 51.6°F. At higher temperatures, the store energy increases up to 8,500 kWh/day at an average ambient temperature of 75°F. When the turbine operates continuously, the store's daily energy use decreases by 1,200-1,400 kWh/day, the amount of electricity generated by the turbine.

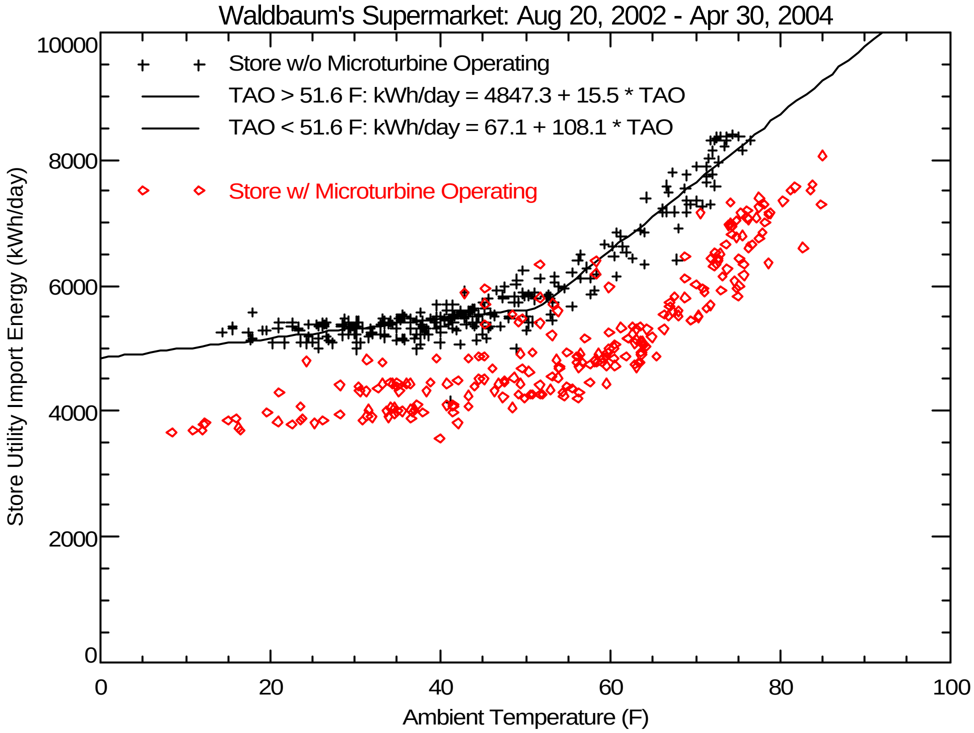


Figure 10. Daily Store Electricity Use Variation with Ambient Temperature

4.2 TURBINE OPERATING DETAILS

The microturbine began consistent operation on April 18, 2003 once the utility interconnection issues were resolved. The shade plots in Figure 11 qualitatively show the trends of power output and gas use for microturbine. Each day is shown as a vertical stripe on the plot. Darker areas indicate periods of higher turbine output. Light gray areas indicate when the turbine is off. Areas of bright white indicate missing data. The microturbine was off for significant amounts of time across the period. Table 6 lists the operating hours and % available for each month. The turbine was "up" an average of 54% across the 14-month period given in the table. For the 12-month period from May-03 through April-04, the up time was slightly better at 62%. Table 7 gives a detailed operating history of the turbine and test site since installation. Turbine shutdowns were sometimes brief events caused by utility grid disturbances. In these cases the turbine usually restarted itself (before the restart feature was disabled). In other cases the turbine or Unifin heat exchanger was down due to a fault. Appendix C includes a detailed listing of the Capstone fault code associated with each event.

<u>Postscript</u>: Since the unit was fixed and restarted on May 28, 2004, it has been running consistently (with only brief interruptions) through the middle of September 2004.

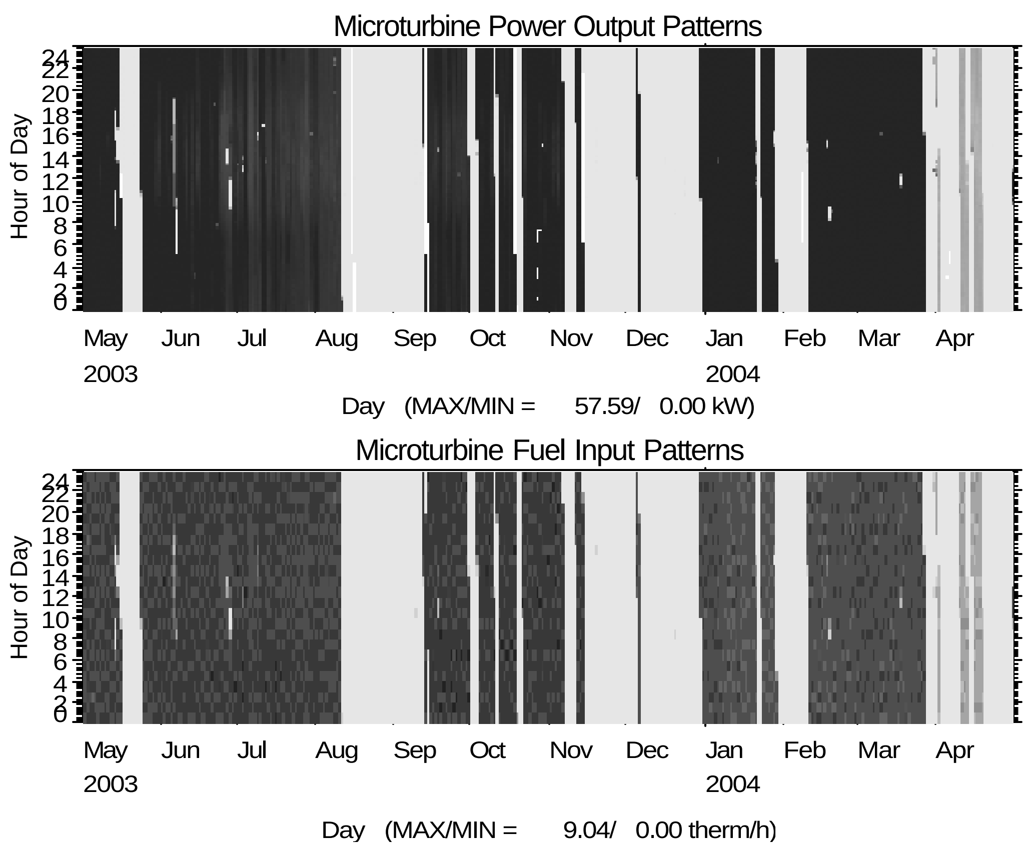


Figure 11. Shade Plot of Microturbine Power Output and Fuel Consumption

Table 6. Summary of Operating Hours for Each Month

	Turbine Operation (hrs)	(%)
April-03	283.2	39%
May-03	545.3	73%
June-03	714.1	99%
July-03	742.8	100%
August-03	216.5	29%
September-03	419.9	58%
October-03	596.7	80%
November-03	179.0	25%
December-03	95.7	13%
January-04	617.9	83%
February-04	491.5	73%
March-04	626.0	84%
April-04	186.3	26%
May-04	78.5	11%
Total	***5,793.5***	***54%***

Table 7. Summary of Major Events at the Test Site

Date(s)	Event
August 19, 2002	First data acquisition system (DAS) equipment installed
August 26, 2002	DAS completed, Most sensors verified.
October 17, 2002	Some DAS sensors repaired and replaced. Various one-time readings taken.
February 24, 2003	Some power data lost due to Veris datalogger fault
April 18, 2003	Turbine operation begins
April 22, 2003	Sensors verified. Additional one-time readings taken.
April 23-24, 2003	Turbine shut down due to a "loss of phase" grid fault. Had to be manually restarted (auto restart feature initiated after this point)
May 13-14, 2003	CDH and SRI on site to install ETV test instruments. Turbine shut down to install gas meter and other transducers.
May 15-23, 2003	Store experienced a ground fault. The microturbine suspected as the cause. Microturbine remained off until it was inspected on May 23. Unit was restarted and several protective relay settings adjusted to be more conservative.
June 5-7, 2003	SRI on site to complete ETV emissions testing
June 26-27, 2003	Turbine temporarily off on grid fault.
July 8, 2003	Open House and Media Event
July 14-18, 2003	Danfoss controller inadvertently called for heat recovery to space heating coil, causing simultaneous heating and cooling.
August 10 – September 11, 2003	Turbine shut down due to a bad igniter. Unit fixed and restarted on September 10-11.

Date(s)	Event
September 30 – October 2, 2003	An igniter problem with the turbine causes natural gas to ignite and damage the Unifin heat exchanger exhaust stack. The stack was repaired on October 2 and the turbine restarted. The microturbine auto-restart feature was disabled.
October 17, 2003	Some power data lost due to Veris datalogger fault (October 17).
October 18-20, 2003	Turbine shut down due to grid fault. Turbine was manually restarted via remote connection on October 20.
November 5-10, 2003	Turbine shut down by an erroneous fault signal from the Unifin. Turbine was manually restarted via remote connection on November 10.
November 13 – December 4, 2003	The turbine engine failed to light after incurring a routine protective relay fault. Capstone service personnel determined the inlet cowling on engine was a problem and entire engine was replaced. The turbine was restarted on December 4.
December 5-29, 2003	Snow was drawn into the power electronics module and caused component failure. The power electronics module was replaced and a "snow shroud" was added to the front of the Capstone unit to prevent snow from entering the unit.
January 20-22, 2004	Turbine shut down due to several over-voltage faults on one phase. The turbine shut itself down and was manually restarted via remote connection on January 22.
January 27, 2004	Service personnel replaced a temperature sensor on the Unifin HX. The bad sensor had prevented normal heat recovery operation since December 29.
January 28, 2004	Water leaked into the top cover of the microturbine and shorted out the power electronics. The leak was fixed and the power electronics were repaired on February 9.
February 17, 2004	A second bad temperature sensor on the Unifin HX was replaced. This sensor had not been affecting operation of the Unifin controls.
March 26, 2004	On March 26, the turbine shutdown due to a calibration problem with the SPV (main fuel valve). The valve erroneously reports higher fuel flow, causing a control fault with the system. To get around this problem, the turbine was set to run at at a lower output when it was restarted in April.
April 9-18, 2004	The turbine ran with the output setting at 18 kW to test the controller. The turbine shutdown due to an erroneously high fuel flow fault.
April 30, 2004	Gas valve replaced. Unit ran for 3 hours, then ECM cooling fan seized.
May 28, 2004	ECM fan motor replaced unit starts to operate.
June-September 2004	The unit operated consistently throughout the summer of 2004.

4.3 TYPICAL MICROTURBINE OPERATING PATTERNS AND TRENDS

Figure 12 displays operation of the microturbine for July 5, 2003. On this day, the turbine and Unifin HX ran continuously. The data in the figure shows that turbine output dropped as the ambient temperature increased from 75 and 95°F. Total turbine output on this hot day was 1179.8 kWh.

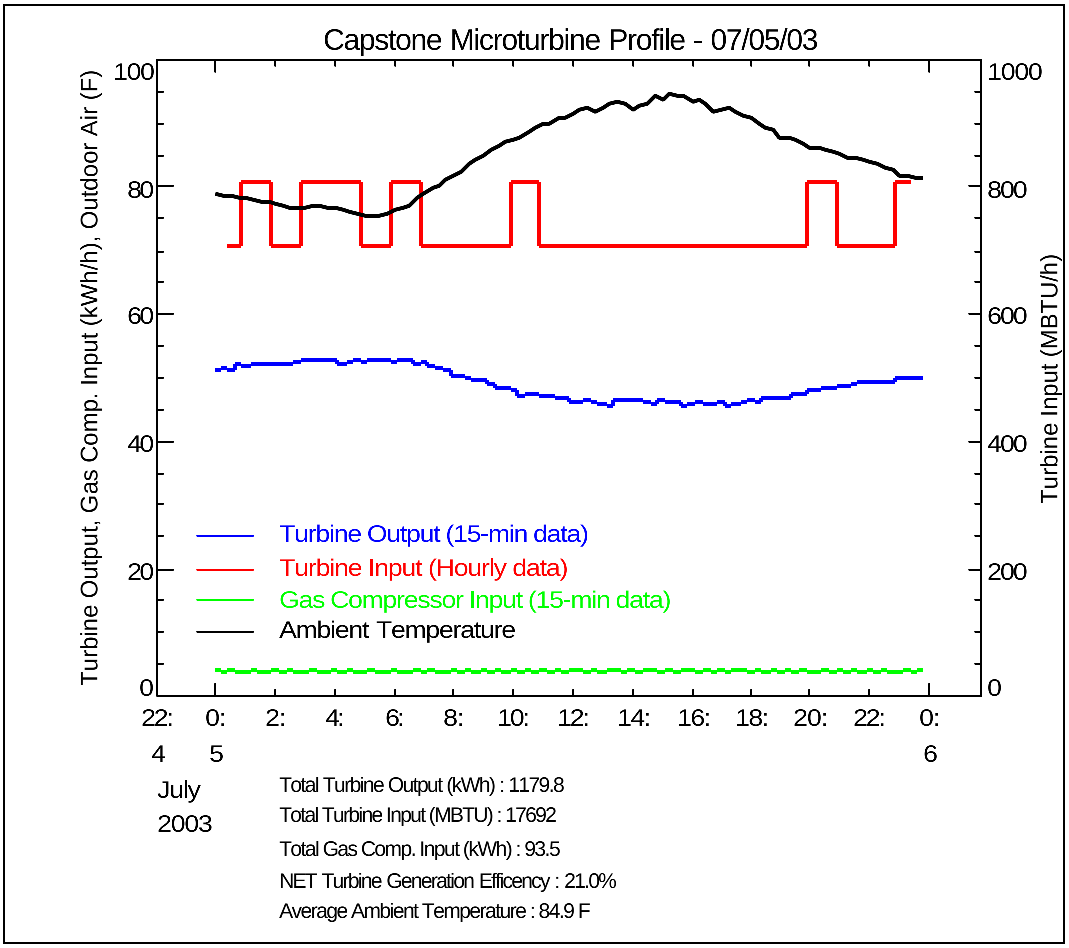


Figure 12. Capstone Microturbine Peak Cooling Day Operation – July 5, 2003

Figure 13 displays operation of the turbine for February 16, 2004, a winter day when the turbine and heat exchanger ran continuously. The turbine output for this day remained at 57 kW while the ambient temperature varied from 15 to 35°F. The total turbine output for this cold day was 1364.8 kWh, or 16% more than on a hot summer day.

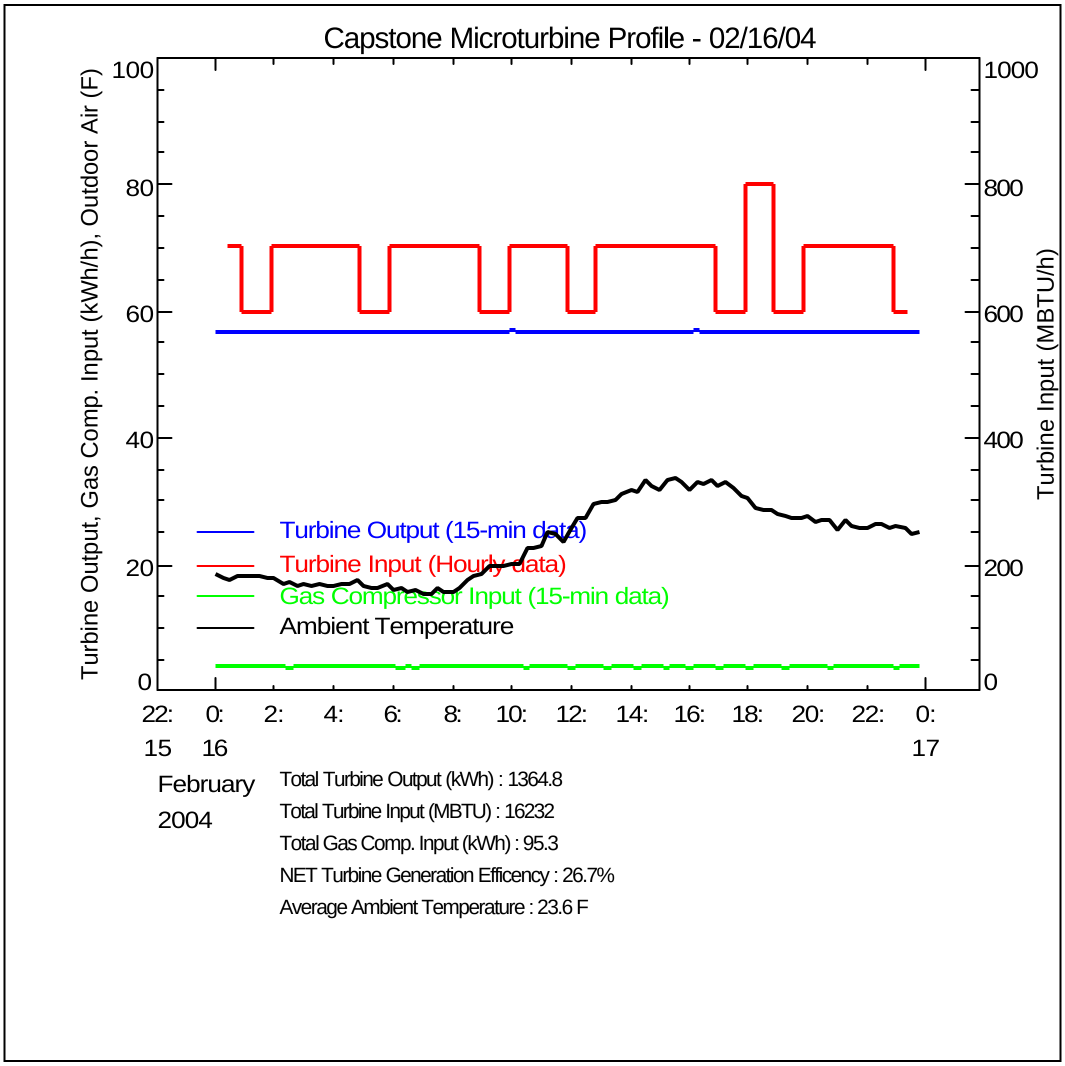


Figure 13. Capstone Microturbine Peak Heating Day Operation – February 16, 2004

Turbine gas use was measured using a pulse output from the dedicated billing meter. The pulse output had a resolution of 100 ft^3/pulse. This provided very coarse indication of gas use, even when summed into hourly intervals. On a daily basis the poor gas meter resolution has less impact. On the summer day shown in the Figure 12 above, the turbine consumed 17,692 MBTU of gas, assuming a higher heating value (HHV) of 1,003 BTU/ft^3 for that day. The range of values for the energy content of natural gas were determined from the Keyspan gas bills. Appendix E summarizes the range of energy content values or “therm factors” observed from the monthly bills. The appendix also compares those Keyspan reported values to the analytical results on gas samples taken from the site in June 2003 (by SRI) and September 2003 (by CDH). The laboratory analysis completed by Empact Analytical Systems was very close to Keypan values. Therefore, the analysis used in this report uses the periodically listed “therm factors” from Keyspan to convert the gas volume flowrate to energy use.

The gas compressor operates continuously while the turbine is running. The average power for the gas compressor was 3.9 kW and it consumed 93.5 kWh on July 5, 2003 and 95.3 kWh on February 16, 2004.

The "net" generation efficiency of the system should include the impact of parasitic power and heat recovery. The net efficiency is defined as:

$$EFF = \frac{W_{output} - W_{parasitic} + Q_{hr}}{G_{input}}$$

where:

W_{output} = Turbine power output (kWh) $\times$ 3.413 MBTU/kWh
G_{input} = Turbine gas input (MBTU, HHV)
$W_{parasitic}$ = Parasitic systems energy input (kWh) $\times$ 3.413 MBTU/kWh
Q_{hr} = Useful heat recovery (MBTU)

When calculating the "net" efficiency for the turbine alone, the only parasitic energy use is the gas compressor (the "net" CHP efficiency presented in the next section also includes the glycol pump power). For the day shown in Figure 12, the net efficiency of the microturbine on a daily basis was 21.0% and for Figure 13 it was 26.7%. The gross turbine generation efficiencies (ignoring the gas compressor) were 22.8 and 28.7% respectively.

Figure 14 shows of turbine power output and gas input with ambient temperature. The hourly gas use data is scattered due to the coarse resolution of the billing meter. The hourly trend shows power dropping with ambient at temperatures above 60-70°F, which was slightly sooner than expected based on manufacturer's specifications.

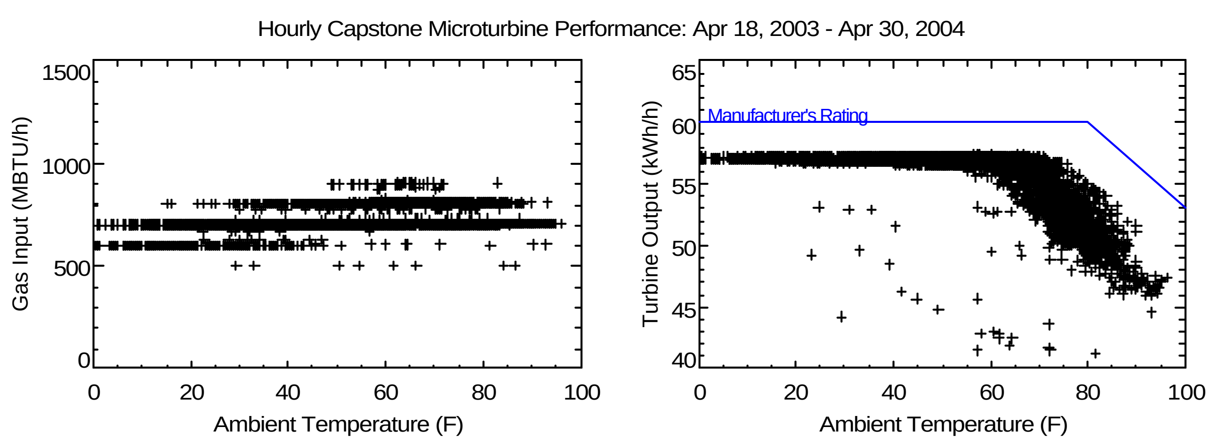


Figure 14. Hourly Turbine Performance Trends with Ambient Temperature

The power output of the microturbine is slightly lower than expected in part because of electric line losses in the wiring from the microturbine to the main distribution panel (MDP). The results from the ETV testing in June 2003 demonstrated that the voltage losses from the microturbine to the MDP were equivalent to 1.4 kW. Figure 15 shows how the voltage drop in the turbine wiring changes with turbine current output. The high losses are a result of the extra long wiring run (about 600 ft) that had to extend to a ground level disconnect in order to satisfy LIPA interconnection requirements. Adding in the 1.4 kW losses increases the maximum turbine output to 59 kW. Appendix A provides the background for this analysis of line losses.

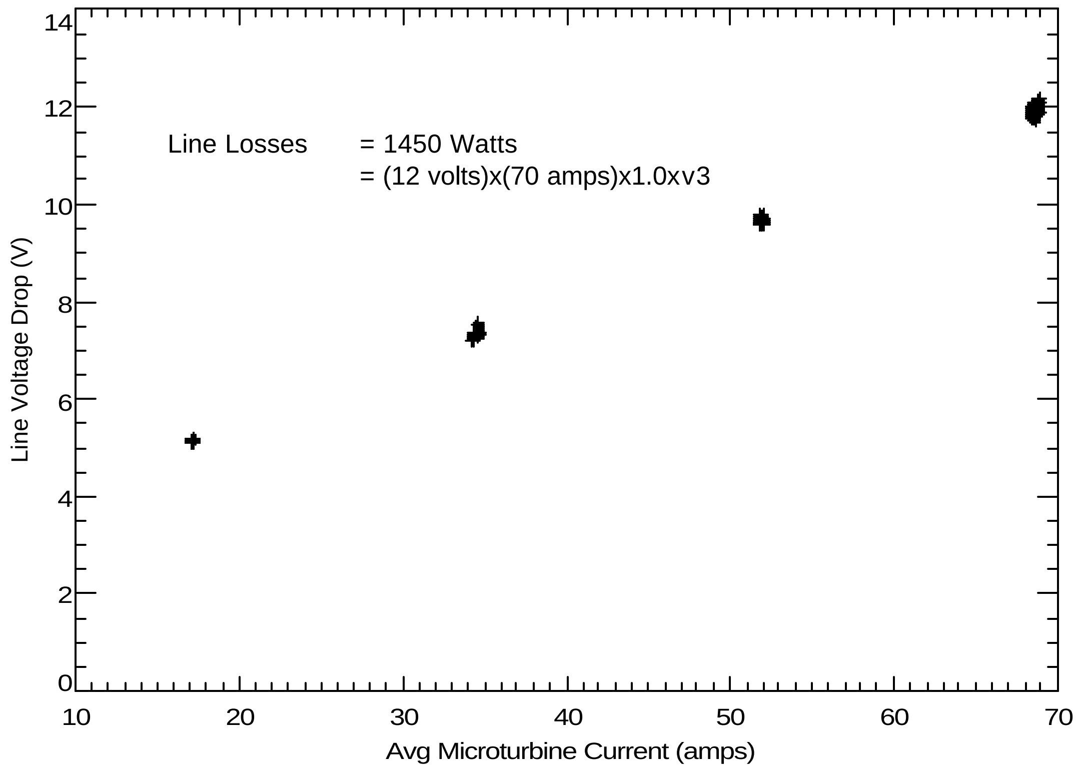


Figure 15. Variation of Phase-Phase Voltage Drop in Turbine Wiring with Turbine Current

On a daily basis, the trend of microturbine efficiency with ambient temperature is more consistent and less scattered. The data in Figure 16 compares the measured efficiency trend with manufacturers data. Two distinct trends in the measured data were observed corresponding to before and after the engine module was replaced on December 29, 2003 (see the events listed in Table 7). The new engine data is shown as red diamonds on the plot. The regression lines fit to the measure trends indicate that gross turbine efficiency decreases by about 0.066 to 0.082% for every 1°F increase in ambient temperature. The regression model predicts a turbine efficiency of 25.0% at 59°F for the old engine and 26.1% at 59°F for the new engine. The performance specifications from Capstone indicate an efficiency of 25.3% at 59°F based on higher heating value (i.e., 25.3% = 28% x 930 / 1030). The results of the more precise SRI testing on June 4-5, 2003 indicated that the Capstone turbine is providing its rated output and efficiency, after compensating for temperature and barometric pressure impacts[2]. Applying the 2.4% correction for wiring losses would increase the measured efficiency values by 0.6 points.

The data in Figure 16 show that the measured efficiency continues to increase at lower air temperatures, at least after the new engine module installed on December 29, 2003. Efficiency continues to increase linearly at lower ambient temperatures instead of being capped at the upper limit predicted by the Capstone specifications. On the coldest days, turbine efficiencies approaching 30% HHV have been observed. Data for the old engine did not reach a temperature low enough to confirm the trend at low temperatures.

[2] As described in Section 4 of the SRI ETV report at www.sri-rtp.com/Capstone_Turbine_test.htm

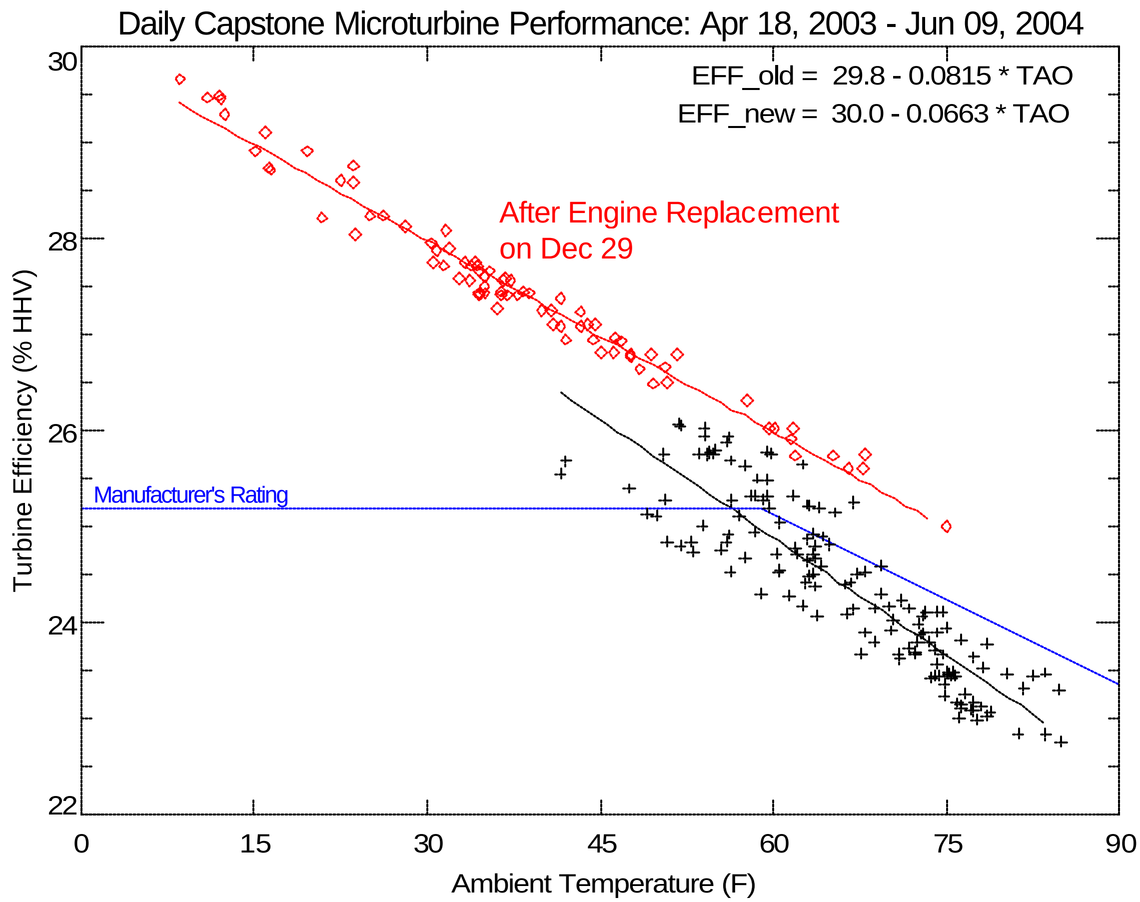


Figure 16. Trend of Daily Gross Turbine Efficiency with Ambient Temperature

4.4 DAILY OPERATING PATTERNS – HEAT RECOVERY AND CHP EFFICIENCY

Heat is recovered from the turbine exhaust using a Unifin exhaust-to-glycol heat exchanger. The recovered heat can be used for: 1) first stage heating in the store's main air handling unit or 2) to pre-heat regeneration air entering desiccant burner section. Because the Unifin glycol pump must operate continuously, the regeneration heat recovery coil – which is located on the outside of the Munters AHU – also acts as a passive heat dump coil when no space heating or desiccant regeneration is required (and the fans are off). The Unifin HX is equipped with a bypass damper that modulates to control the leaving glycol temperature to its control set point of 180°F.

Figure 17 and Figure 18 show operation of the heat recovery system for February 16, 2004 and July 5, 2003: the peak heating and cooling days for the monitoring period with turbine activity.

The top plot displays the system statuses related to the heat recovery system. The glycol pump operates continuously during turbine operation to prevent damage to the heat exchanger. The pump power was determined to use 750 Watts from handheld measurements. The status points indicate when the regeneration fan was activated. For this day there was no desiccant operation. The status labeled "Space Heating HR Operation" shows the operation of the valve sending glycol flow to the space heating HR coil. The Danfoss control system activated the valve when the space heating was required.

The middle plot displays the measured exhaust temperatures across the day. The entering exhaust temperature was between 520 and 550°F. The exhaust temperature leaving the Unifin HX ranged from 500°F while the bypass damper was open, to 150°F when the bypass damper was closed (and heat was being recovered to the glycol loop for space heating).

The bottom plot in Figure 17 illustrates the operation of the glycol side of the heat exchanger. The leaving glycol temperature is 116-130°F with a 13°F temperature drop when the space heating coil is active. The glycol temperature increases to about 190°F when the valve is open.

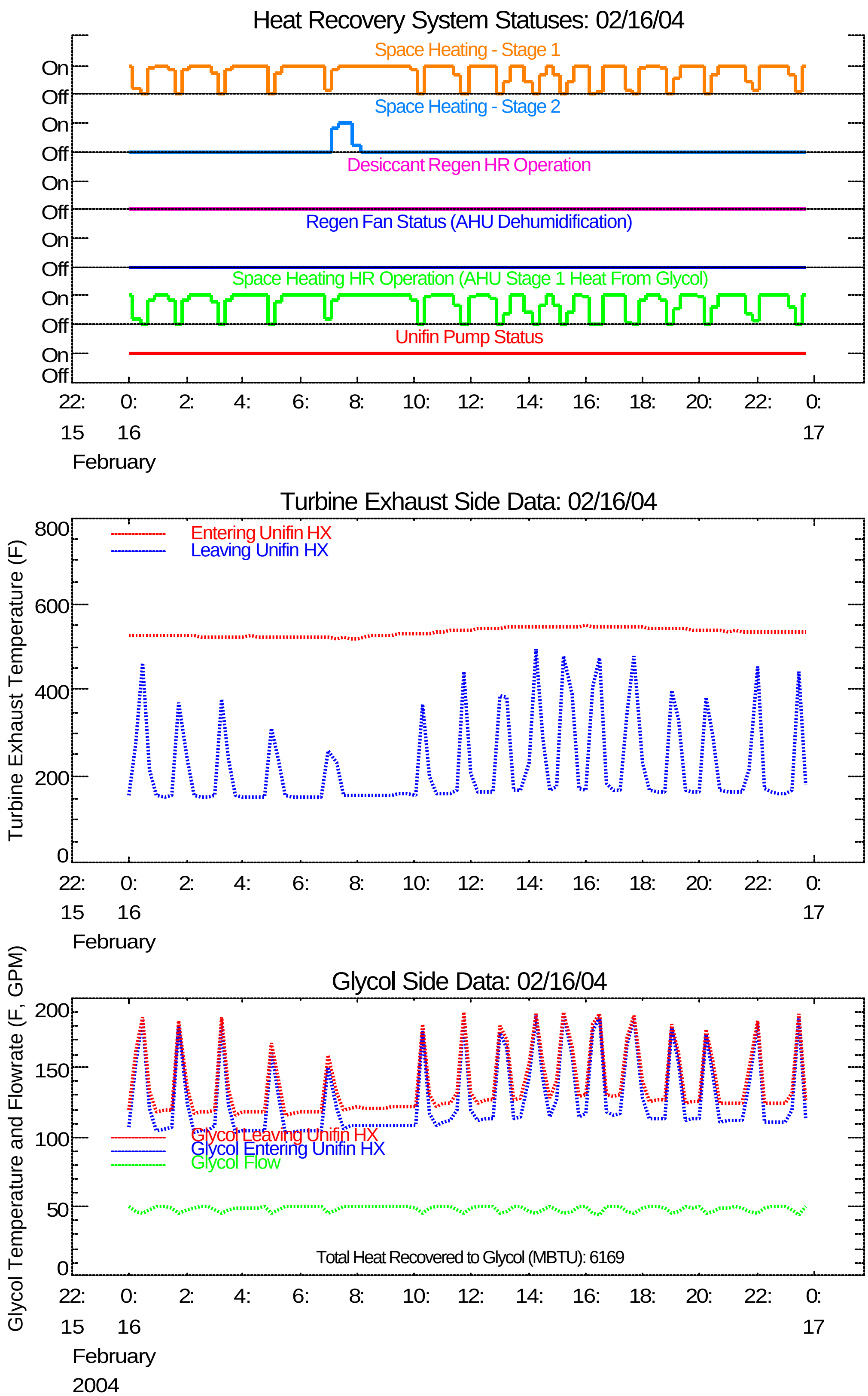


Figure 17. Heat Recovery System Operation on Peak Heating Day – February 16, 2004

Figure 18 shows the same data but for a typical summer day. For this day there was continuous desiccant operation after morning thermostat setup. Space heating was never required.

The middle plot displays the measured exhaust temperatures across the day. The entering exhaust temperature was 640°F. The exhaust temperature leaving the Unifin HX ranged from 600°F while the bypass damper was open, to 350°F when the bypass damper was closed and heat was being recovered to the glycol loop. The exhaust temperature leaving the HX is higher in this mode since less heat can be applied to preheat 90°F ambient air entering the burner.

The bottom plot in Figure 18 illustrates the operation of the glycol side of the heat exchanger. The leaving glycol temperature in this case is 186-188°F with a 9°F temperature drop when the regeneration coil is active. The glycol temperature increases slightly to 190°F when the valve is open. Again, glycol temperatures are much warmer in the summer due to the nature of the thermal load.

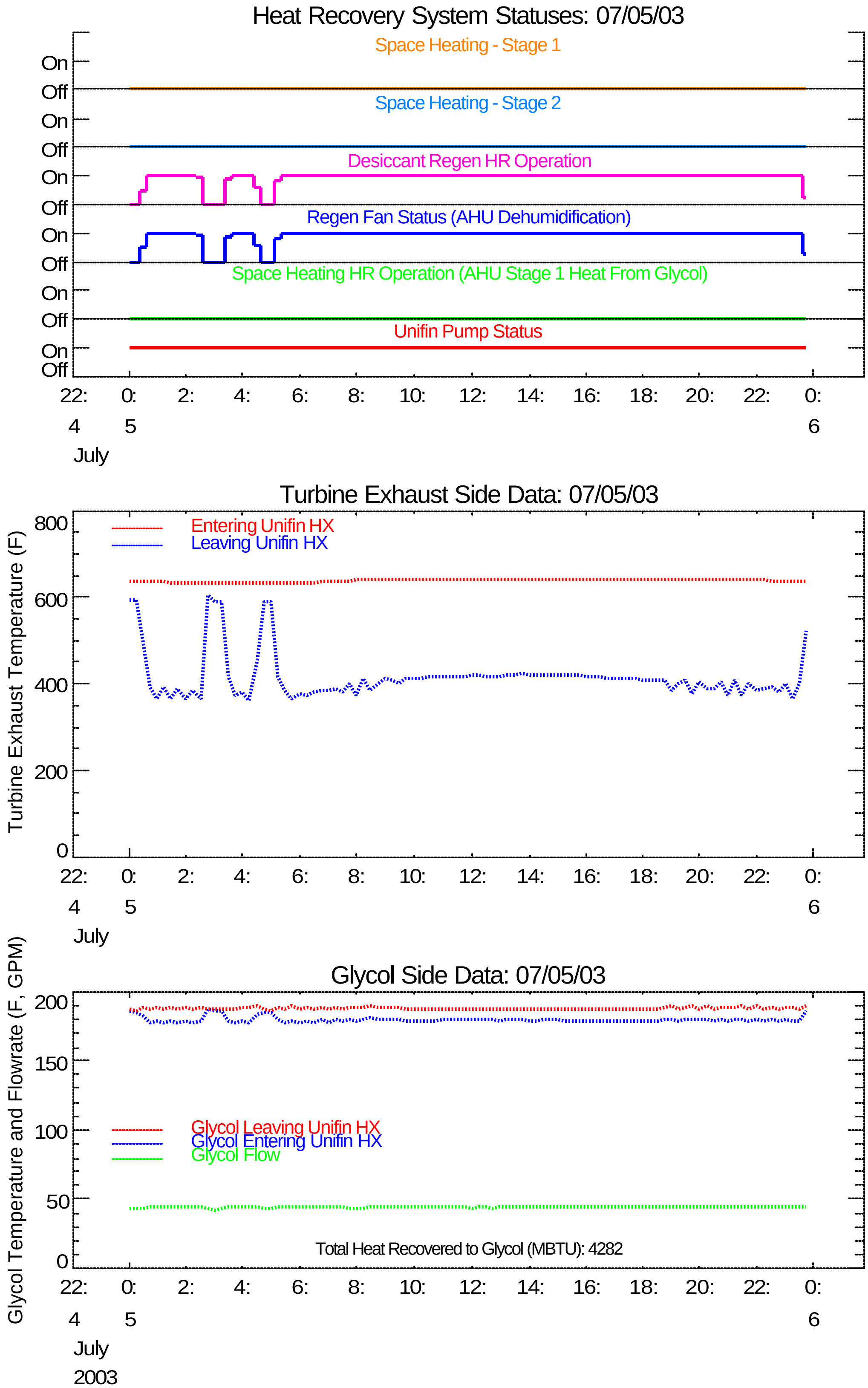


Figure 18. Heat Recovery System Operation on Peak Cooling Day – July 5, 2003

Using the measured heat recovery and turbine power output, the net CHP system efficiency can be determined using the equation on page 21 above.

Table 8 summarizes the monthly net generation and net CHP system performance since April 18, 2003. Tables of the daily CHP efficiency for each month are found in Appendix B. The heat recovery columns in both tables only include useful heating provided for space heating or desiccant regeneration.

Table 8. Microturbine Generation and CHP Performance

	[1]	[2]	[3]	[4]	[5]	[6]	[7] = [1-3] / [2]	[8] = [1-3-4+5+6] / [2]
	Turbine		Parasitic Loads		Heat Recovered			
Date	Power Output (kWh)	Gas Input (MBTU)	Gas Compressor (kWh)	Heat Recovery Glycol Pump (kWh)	Space Heating (MBTU)	Desiccant Regen (MBTU)	"Net" Turbine Generation Efficiency (%)	"Net" CHP Efficiency (%)
April-03	15,356	209,649	1,097.3	250.0	16,162	0	23.2%	30.5%
May-03	30,414	411,031	2,113.0	474.6	29,084	2,045	23.5%	30.7%
June-03	39,087	549,741	2,767.1	530.1	18	17,223	22.5%	25.4%
July-03	39,185	568,723	2,878.3	635.8	103	72,102	21.8%	34.1%
August-03	10,864	161,883	838.9	185.7	0	46,035	21.1%	49.2%
September-03	22,210	328,755	1,627.2	359.1	457	40,837	21.4%	33.6%
October-03	33,777	465,929	2,312.4	512.2	21,063	12,828	23.0%	29.9%
November-03	10,005	138,575	693.8	153.3	5,939	6,192	22.9%	31.3%
December-03	5,290	66,833	370.7	81.7	4,688	0	25.1%	31.7%
January-04	34,702	417,133	2,394.5	535.4	2,769	0	26.4%	26.7%
February-04	27,701	341,383	1,904.6	426.4	92,226	0	25.8%	52.4%
March-04	35,160	440,680	2,425.9	544.4	102,987	0	25.4%	48.3%
April-04	3,470	57,315	721.9	161.1	12,878	0	16.4%	37.9%
12-month Totals	***303,749***	***4,100,315***	***21,424***	***4,689***	***275,496***	***197,263***	***23.5%***	***34.6%***

Note: Actual natural gas HHV is used.

The amount of heat recovery varies with the operating mode. Figure 19 shows that the Unifin provides up to 400 MBtu/h when glycol is directed to the space-heating coil. When the glycol is diverted to the regeneration coil the heat recovery rate drops to about 230 MBtu/h (since the coil is smaller and the entering air temperature is warmer). When the regeneration fan is off (but glycol still flows to the regeneration coil) the system passively rejects about 20-30 MBTU/h. This passive heat loss is not classified as useful heat transfer and is therefore not included in any totals in the tables above.

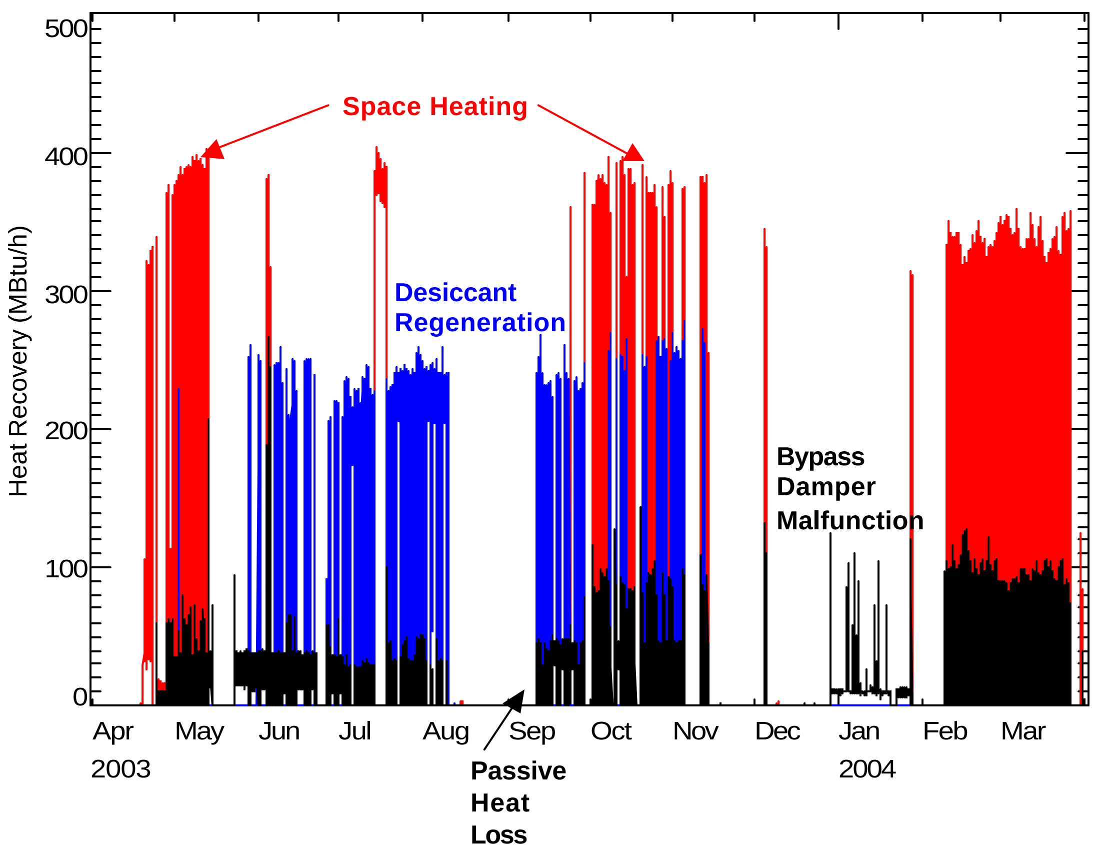


Figure 19. Useful Heat Recovery for Space Heating and Desiccant Dehumidification

The amount of available heat recovery load and the CHP efficiency are primarily driven by ambient conditions. Figure 20, Figure 21, and Figure 22 show the amount of useful heat recovery and CHP efficiency for a day varies with ambient temperature and humidity. The plots only include data for days where the turbine operated continuously and the heat recovery system functioned properly throughout the day. The maximum achievable CHP efficiency on a hot summer day is about 50%. The CHP efficiency for space heating has reached as high as 60% on days when the average temperature is 24°F. Figure 21 shows that the daily load on the space heating coil is driven by ambient temperature while Figure 22 shows that the regeneration heat load is a function of the ambient humidity level.

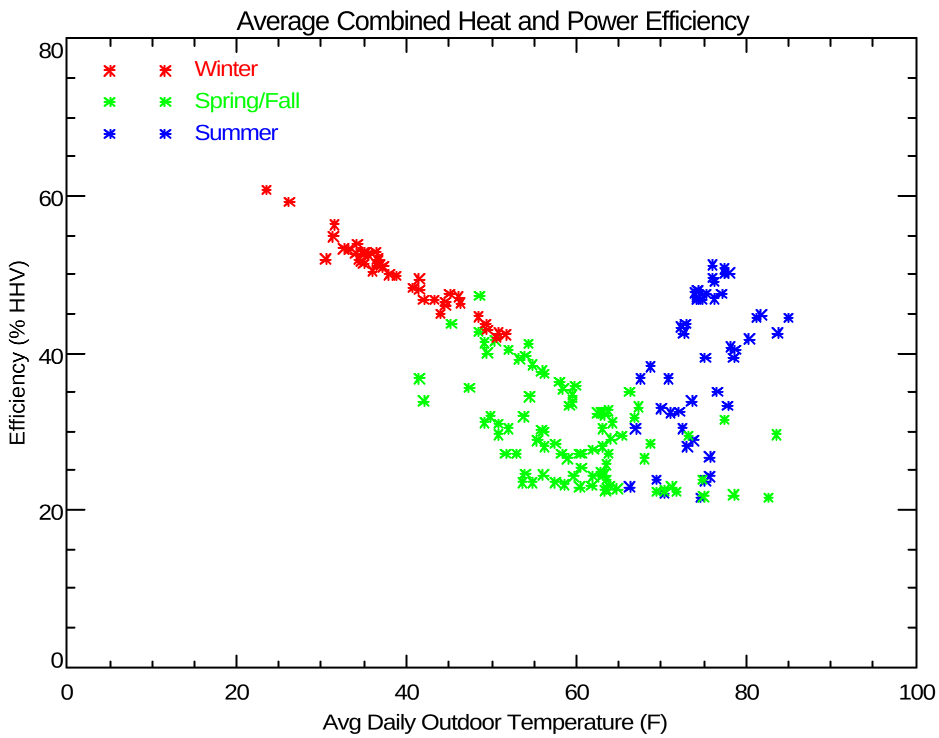


Figure 20. Impact of Ambient Temperature on CHP Efficiency

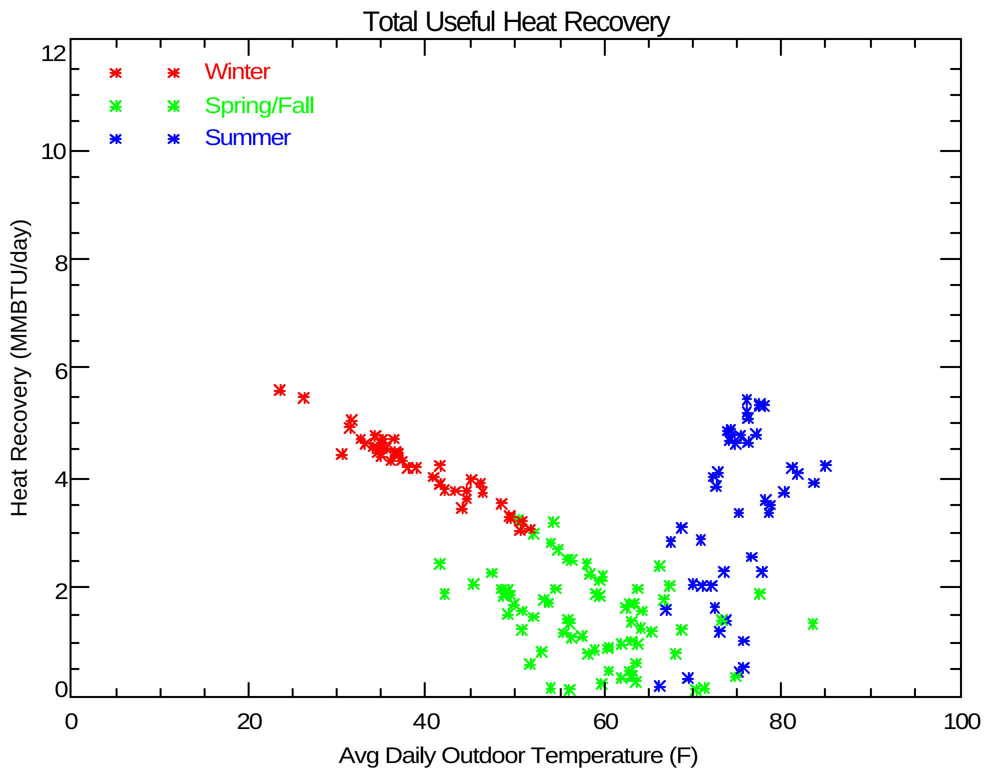


Figure 21. Impact of Ambient Temperature on Useful Heat Recovery by Both Coils

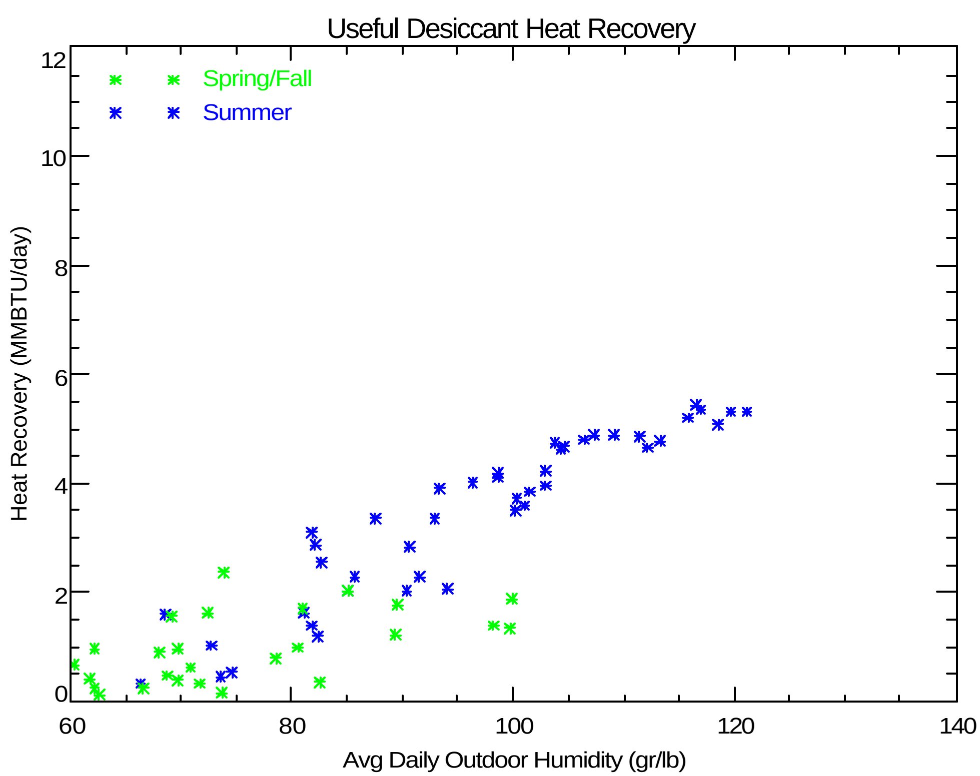


Figure 22. Impact of Ambient Humidity on Useful Heat Recovery by Regeneration Coil

4.5 TURBINE EXHAUST TEMPERATURES, FLOW AND BACK PRESSURE

Several data points were monitored to evaluate the exhaust stream from the microturbine. Figure 23 shows that the turbine exhaust temperature was affected by ambient air temperature. The trend indicates that the exhaust temperature increases by almost 2°F with each 1°F increase in ambient temperature. The exhaust temperature reached 640°F at peak ambient conditions (95°F).

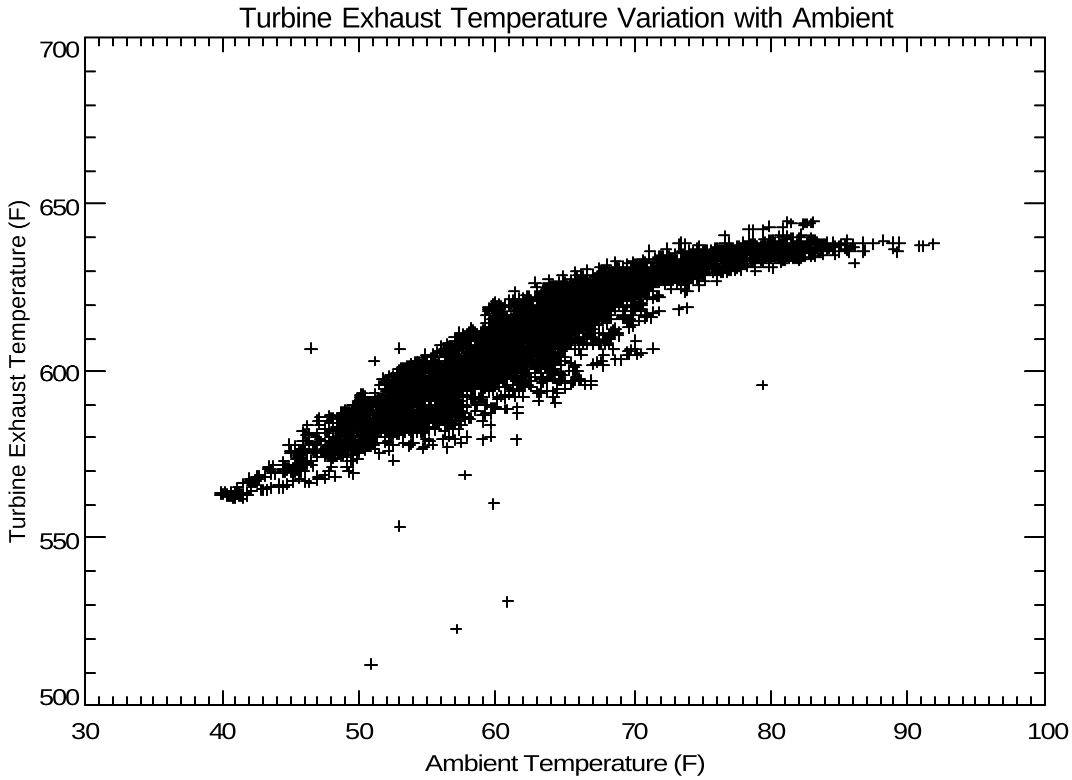


Figure 23. Variation of Turbine Exhaust Temperature with Ambient Temperature (April-June 2003)

Figure 24 shows the velocity pressure measured by the pitot tube in 8 inch diameter turbine exhaust duct. The pressure transducer was only valid for the first few weeks of operation. The sensor was checked and fixed several times over the summer of 2003, however, it continually failed because the tubing quickly filled with condensate after a few days of operation. For the initial period shown in Figure 24, the average velocity pressure was 0.495 inches. Using this velocity pressure, the turbine exhaust flow is estimated to be 3,108 lbm/h using the flow-pressure equation given in Appendix A (and assuming an exhaust gas density of 0.037 lbm/ft^3 at 605°F). This value matches reasonably well with the nominal exhaust flow rating of 1.06 lb/s (or 3,816 lbm/h) from the Capstone specifications.

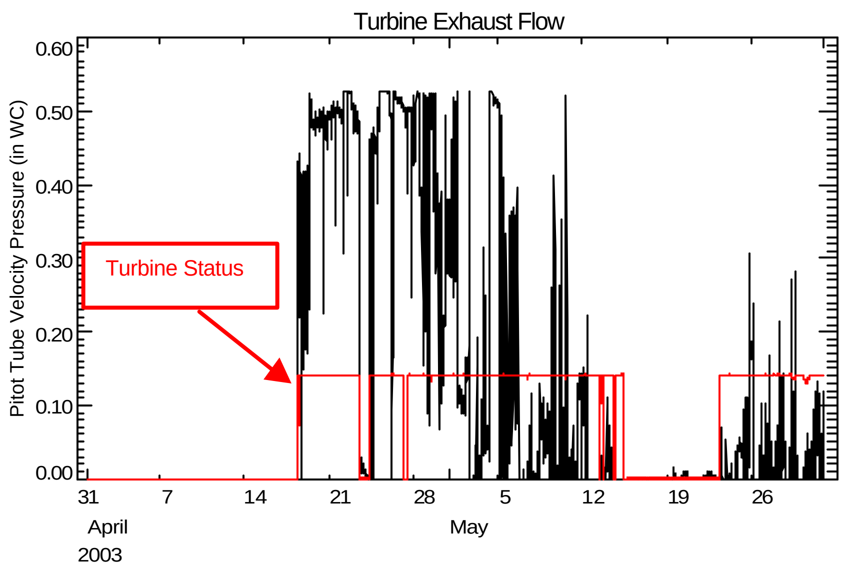


Figure 24. Pitot Tube Velocity Pressure Measurement in Turbine Exhaust Duct

Another method is estimate the exhaust gas flowrate was to use the measured heat recovery rate and the two exhaust gas temperatures (and assuming a specific heat of 0.25 Btu/lb-°F-h for the exhaust gases). Figure 25 shows the result of this approach. The plot only includes data for the Unifin meeting the space heating load since that mode provided consistent operation without the Unifin damper cycling. The data points on the plot are the mass flow determined from the heat balance. The average mass flow is about 4000 lb/h. The lines on the plot are the pitot-tube measured flow and the nominal rated flow from Capstone. All these values are in good agreement.

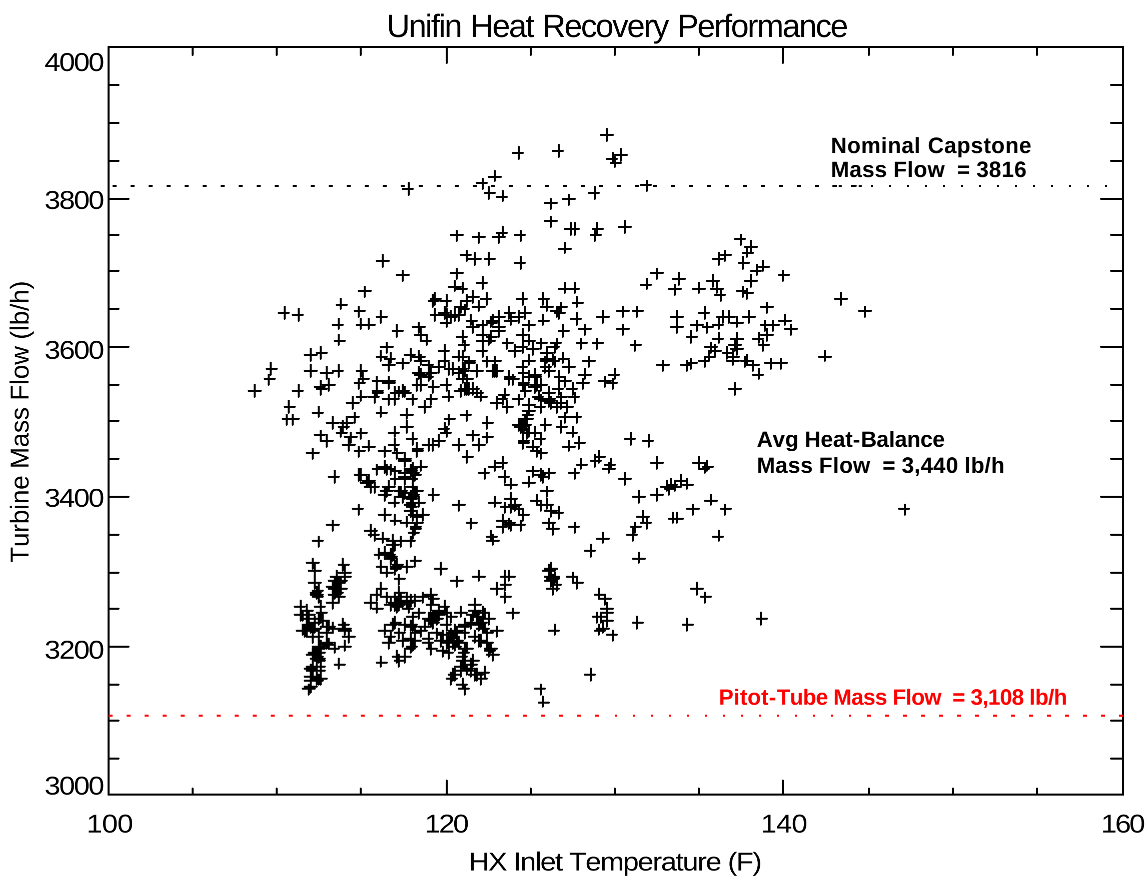


Figure 25. Pitot Tube Velocity Pressure Measurement in Turbine Exhaust Duct

Figure 26 shows the turbine exhaust back pressure variations for a typical day. Figure 27 shows the longer term average values. The back pressure on the turbine is a result of the pressure drop through the Unifin heat exchanger. The average back pressure is about 4.5 inches, though the pressure does change with ambient temperature and the position of the bypass damper. The bypass damper is open when the glycol temperatures are high (near 180°F) and the temperature difference between the glycol entering and leaving temperature approaches zero. The data in Figure 26 show that when the bypass damper is closed, the turbine exhaust pressure typically decreases by 0.25 inches.

The exhaust gas pressure drop on this Unifin unit was probably higher than expected. The incorrect heat exchanger unit was shipped to the site. This unit was identical to the correct unit except that the duct connection on the top of the cabinet was 5 inches in diameter instead of 8 inches. Unifin provided a custom-made transition fitting to complete the connection. However, this "neckdown" in the exhaust ducting probably increased the static pressure.

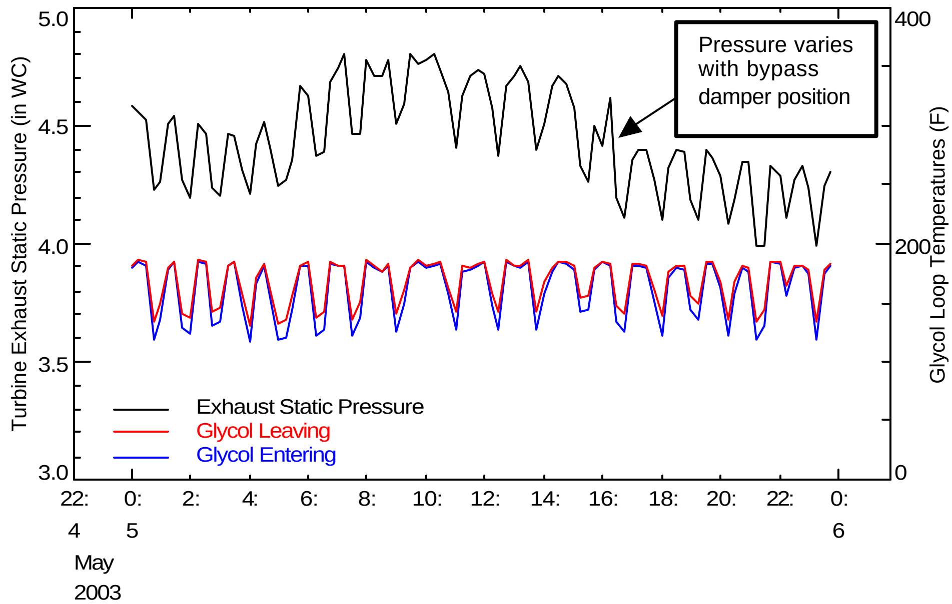


Figure 26. Turbine Exhaust Static Pressure – May 5, 2003

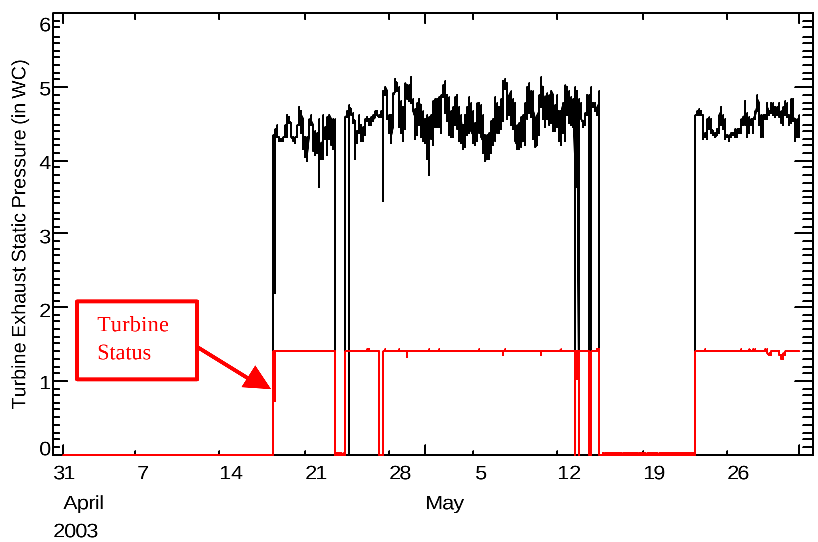


Figure 27. Turbine Exhaust Static or Back Pressure – April and May 2003

4.6 UNIFIN HEAT EXCHANGER PERFORMANCE

Figure 28 and Figure 29 compare the measured performance of the Unifin HX to the manufacturer's specifications. The data are only shown for intervals when the system was in the full heat recovery mode. Data are shown with different symbols for periods when the Unifin meets either the space heating or desiccant regeneration loads. The manufacturer's data use the average measured glycol flow rates in the two modes (as shown on the plots). The data in the space heating mode are closer to manufacturer's trends since the unit was constantly loaded with the bypass damper in the Unifin unit fully open. In the desiccant mode, the loop temperatures were typically near the control point where the bypass damper cycled. The cycling damper in the desiccant mode degraded the average heat recovery rate. The measured heat transfer rate in the space heating mode was about 10% lower than the rated capacity for the unit.

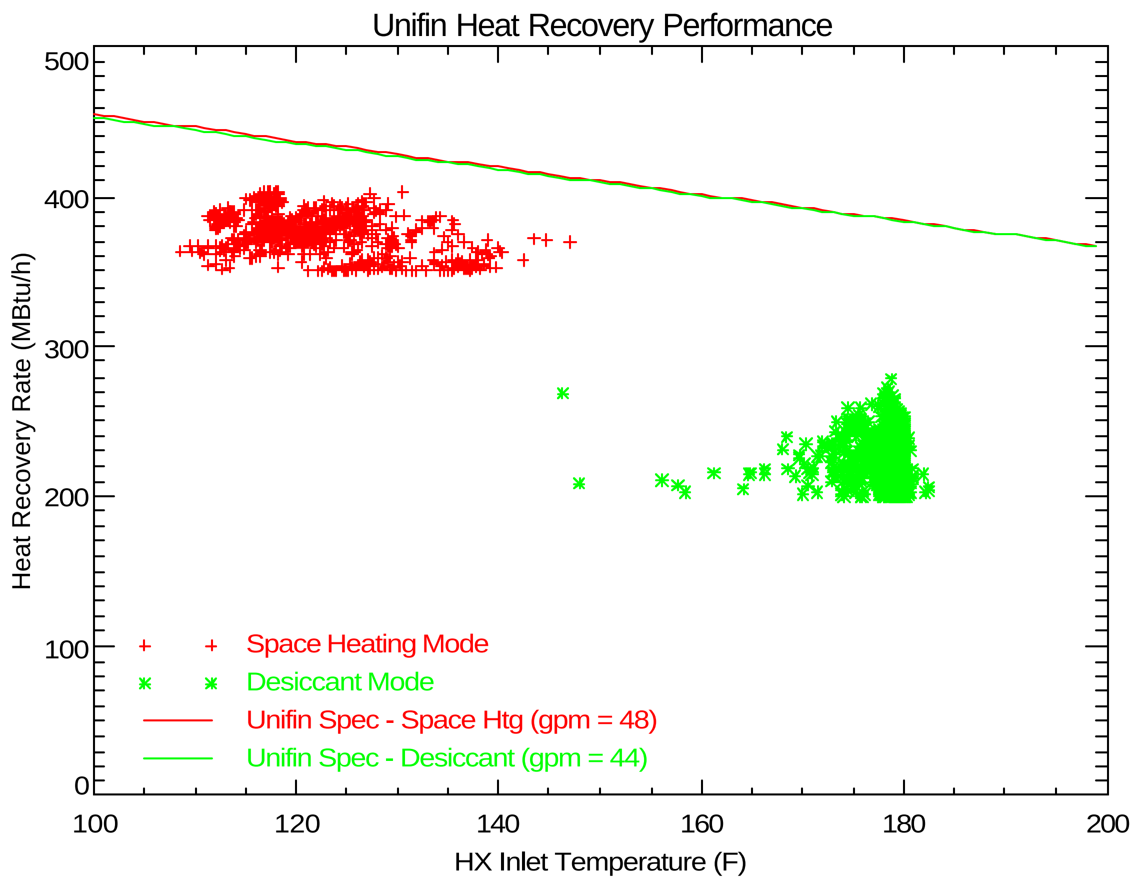


Figure 28. Comparing Measured Heat Recovery to Unifin Performance Specification

The effectiveness of the Unifin heat exchanger is the actual heat transferred divided by the maximum possible heat transfer. The exhaust side of the heat exchanger was the "C_{min}" side in this case, so the temperatures from that side of the system were used. As for the heat transfer rates, the effectiveness was just short of the Unifin specifications in the space heating mode when the unit was full loaded.

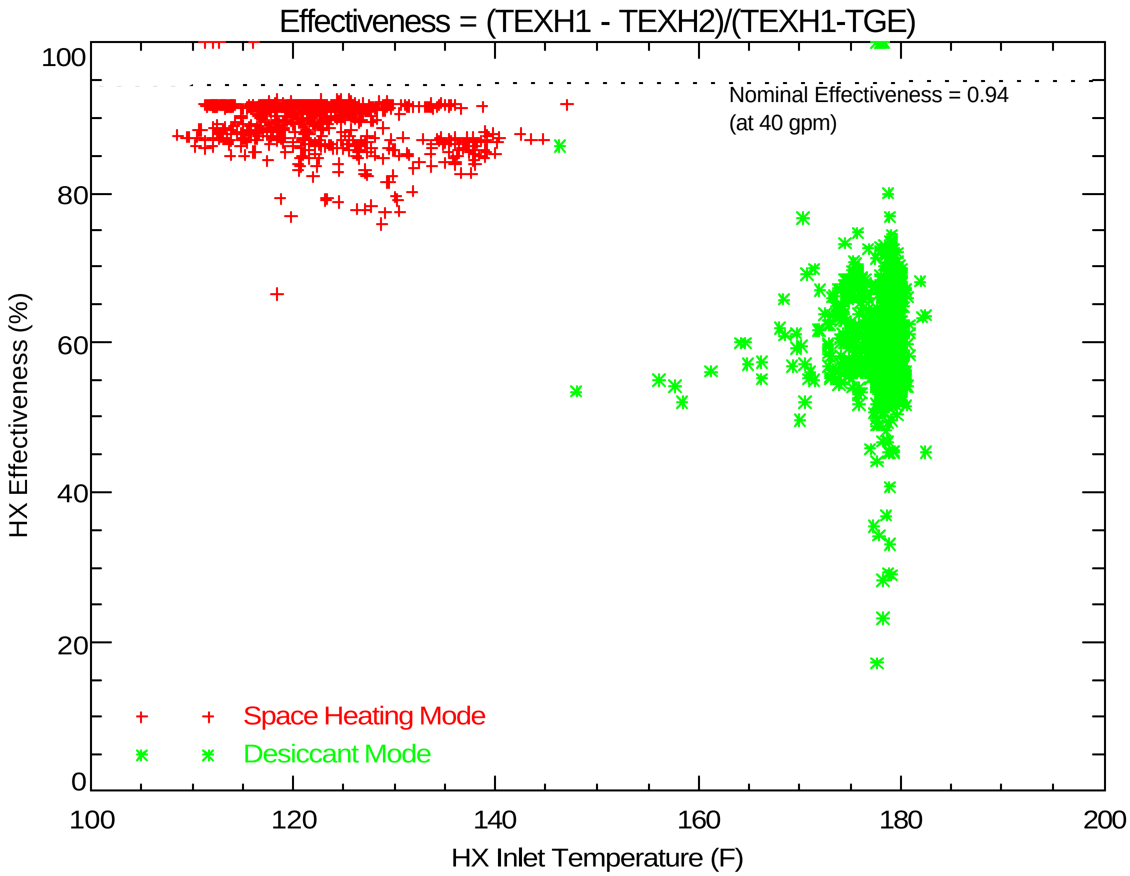


Figure 29. Comparing Measured Heat Transfer Effectiveness to (Inferred) Unifin Specifications

4.7 Store Main AHU and Desiccant Unit Operation

The three shade plots below display the operating patterns observed for the store's main air handling unit (AHU) fans. The Munters AHU contains heating, cooling and desiccant sections. The runtime of each component is shown with shades of gray. Dark areas indicate when the component was on. The supply fan operates continuously to provide heating, cooling and ventilation to the store. The desiccant module process and regeneration fans only run during dehumidification operation.

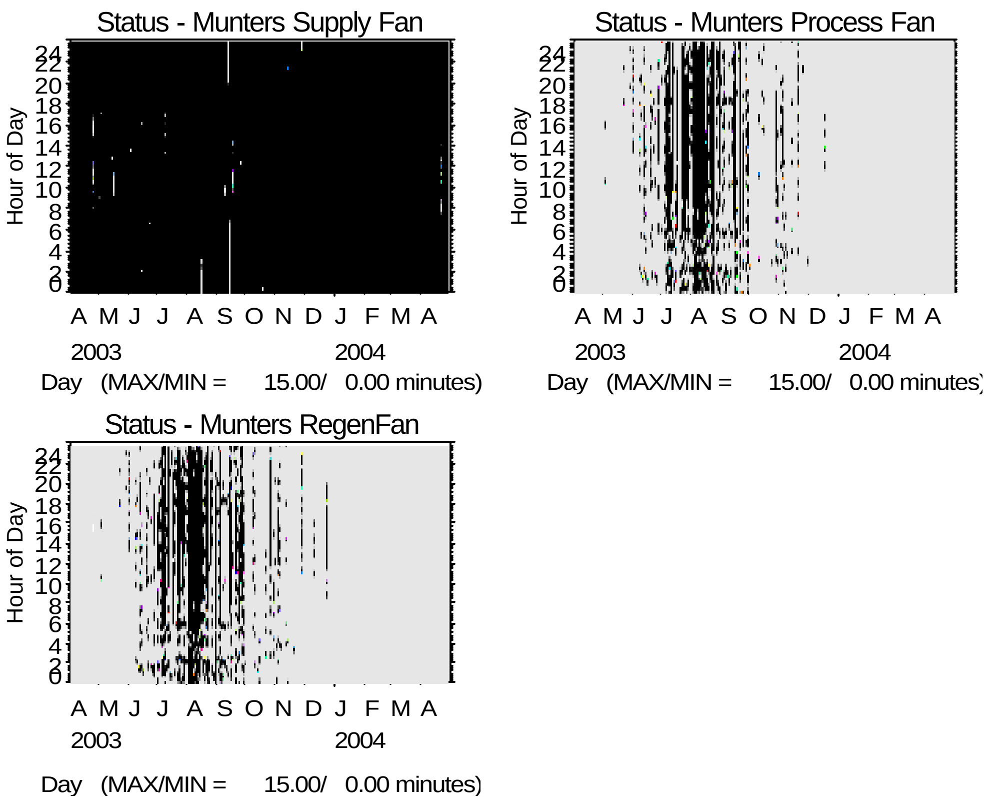


Figure 30. AHU Fan Operating Patterns

Figure 31 displays the heating and cooling operation of the AHU. The heating statuses were repaired in late April 2003, and accurately recorded the runtime of the two gas fired duct heaters since (we were only recording gas use before that date). The horizontal trends on the heating and cooling status plots (and the gas use plot in Figure 32) that occur near 6:00 AM each day indicate that the store is using nighttime thermostat setup/setback. Typically, each non-summer morning both stages of the duct heater operate to recover the store's space temperature.

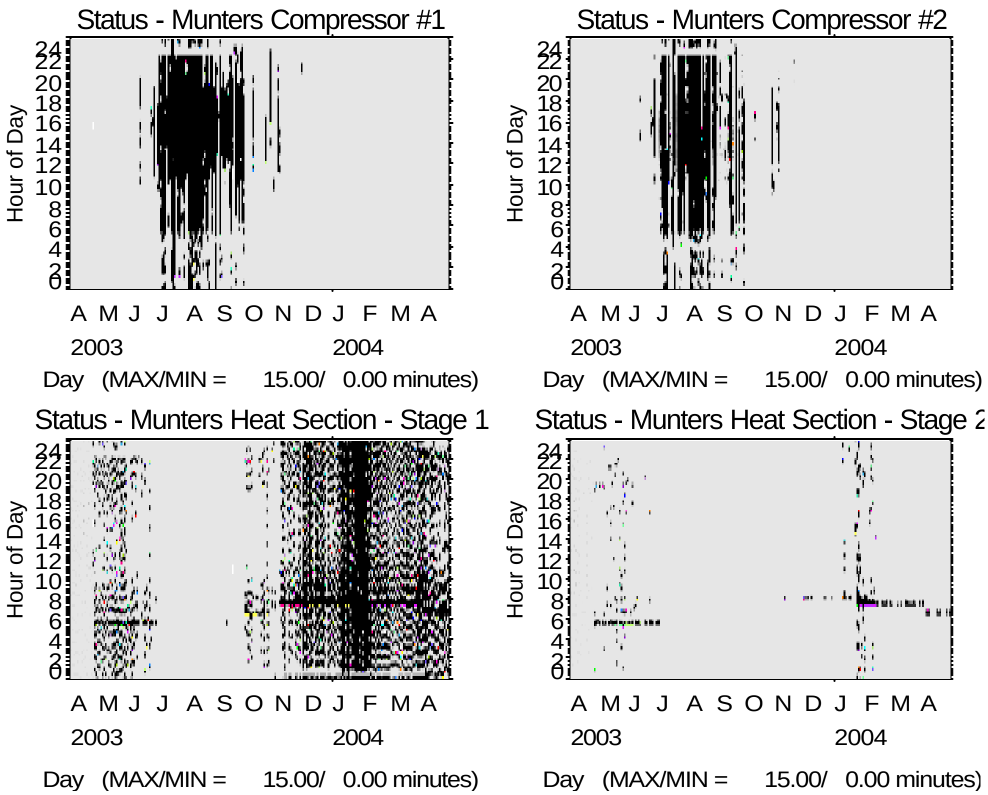


Figure 31. AHU Cooling/Heating Operating Patterns

Figure 32 shows the AHU electricity and gas use patterns. The morning warm-up from nighttime setback is apparent in the gas use patterns. Cooling setup is also apparent from the electric use.

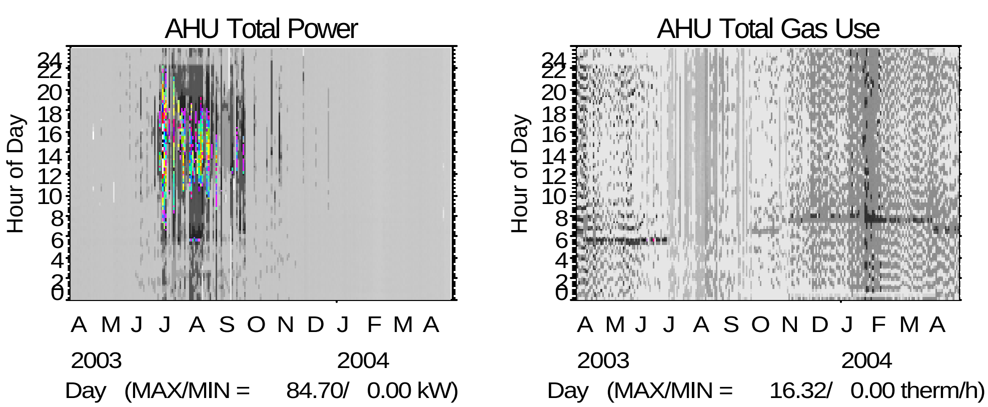


Figure 32. AHU Electricity and Gas Use Patterns

Table 9 and Table 10 summarize the monthly runtime and energy consumption of the AHU components.

Table 9. Monthly AHU Component Runtime

Month	Supply Fan Runtime (hours)	Dehumid. Runtime (hours)	Cooling Stage 1 Runtime (hours)	Cooling Stage 2 Runtime (hours)	Gas Heat Stage 1 Runtime (hours)	Gas Heat Stage 2 Runtime (hours)	Percent Data Collected (%)
Aug-02	288.7	156.5	135.2	111.4	n/a	n/a	40%
Sep-02	720.0	328.5	296.2	234.7	n/a	n/a	100%
Oct-02	739.9	133.3	77.7	70.7	n/a	n/a	100%
Nov-02	717.1	25.7	0.0	0.0	n/a	n/a	100%
Dec-02	743.9	6.5	0.0	0.0	n/a	n/a	100%
Jan-03	743.0	0.0	0.0	0.0	n/a	n/a	100%
Feb-03	661.9	0.0	0.0	0.0	n/a	n/a	99%
Mar-03	720.3	6.0	0.0	0.0	n/a	n/a	100%
Apr-03	714.6	0.0	0.0	0.0	42.9	8.2	100%
May-03	741.8	9.9	0.0	0.0	206.5	43.7	100%
Jun-03	719.8	106.6	42.1	41.8	67.2	16.8	100%
Jul-03	743.2	365.4	489.9	390.5	0.9	0.6	100%
Aug-03	732.6	517.3	519.3	482.9	0.0	0.0	100%
Sep-03	706.9	250.3	292.9	187.8	4.4	0.0	100%
Oct-03	744.0	58.3	35.6	22.3	84.5	0.0	100%
Nov-03	716.2	50.8	17.8	9.7	225.5	1.6	100%
Dec-03	744.0	15.1	0.0	0.0	433.9	4.5	100%
Jan-04	744.0	0.0	0.0	0.0	600.7	62.8	100%
Feb-04	696.0	0.0	0.0	0.0	468.2	28.5	100%
Mar-04	744.0	0.0	0.0	0.0	397.1	6.6	100%
Apr-04	718.4	1.0	0.0	0.0	371.5	8.5	100%
12-Month	***8,747***	***1,374***	***1,398***	***1,135***	***2,532***	***173***	

Table 10. Monthly AHU Component Energy Consumption

	Electricity Use					Gas Use			
Month	Supply Fan Energy (kWh)	Dessicant Process Fan Energy (kWh)	Dessicant Regen Fan Energy (kWh)	Condensing Section Energy (kWh)	Total AHU Energy (kWh)	Dehumid. Gas Use (therms)	Space Heating Gas Use (therms)	Total AHU Gas Use (therms)	Percent Data Collected (%)
Aug-02	3,132	947	829	4,273	9,181	710.4	0.0	710	40%
Sep-02	7,812	1,987	1,742	10,758	22,299	1,741.3	0.0	1,741	100%
Oct-02	8,028	807	704	2,953	12,492	706.7	721.2	1,428	100%
Nov-02	7,780	155	136	0	8,072	138.6	2,526.8	2,665	100%
Dec-02	8,071	39	35	0	8,145	55.6	3,784.1	3,840	100%
Jan-03	8,062	0	0	0	8,062	0.0	4,794.3	4,794	100%
Feb-03	7,182	0	0	0	7,182	0.0	4,122.0	4,122	99%
Mar-03	7,816	37	32	0	7,884	31.2	2,992.7	3,024	100%
Apr-03	7,754	0	0	114	7,868	0.0	2,346.1	2,346	100%
May-03	8,049	60	53	181	8,342	43.1	1,516.2	1,559	100%
Jun-03	7,810	645	565	2,089	11,109	414.1	474.3	888	100%
Jul-03	8,064	2,211	1,938	18,585	30,797	1,092.6	8.3	1,101	100%
Aug-03	7,948	3,129	2,742	20,616	34,436	2,129.4	0.0	2,129	100%
Sep-03	7,670	1,514	1,328	9,078	19,589	905.0	27.2	932	100%
Oct-03	8,072	353	310	459	9,194	199.1	514.4	714	100%
Nov-03	7,771	308	270	483	8,832	218.7	1,389.0	1,608	100%
Dec-03	8,072	91	80	0	8,244	88.3	2,710.9	2,799	100%
Jan-04	8,072	0	0	0	8,072	0.0	4,188.0	4,188	100%
Feb-04	7,552	0	0	0	7,552	0.0	3,068.8	3,069	100%
Mar-04	8,072	0	0	0	8,072	0.0	2,470.1	2,470	100%
Apr-04	7,795	6	5	0	7,806	11.5	2,340.4	2,352	100%
12-Month	***94,907***	***8,311***	***7,284***	***51,605***	***162,107***	***5,090***	***18,713***	***23,804***	
	37%	***3%***	***3%***	***20%***	***64%***	***10%***	***39%***	***49%***	

The next three figures show the variation of AHU electricity use, space heating gas use, and dehumidification gas use with ambient conditions. Figure 33 shows that space cooling operation for the AHU starts when the daily ambient temperature reaches 63-64°F.

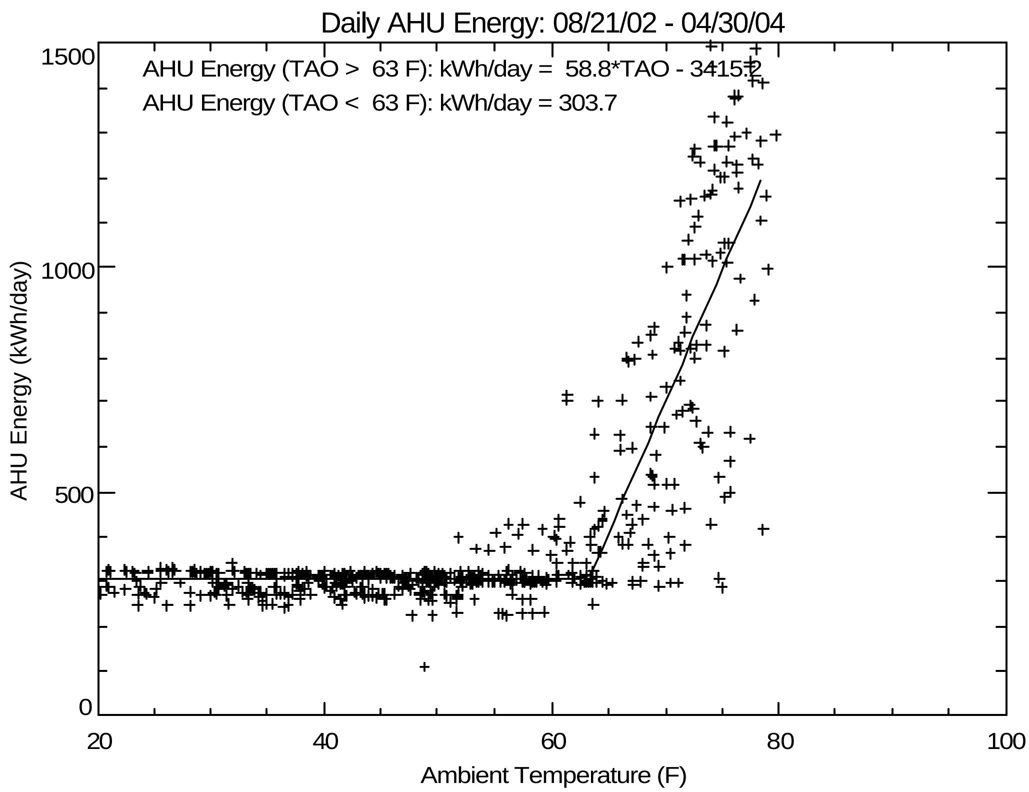


Figure 33. AHU Electricity Use Variation With Ambient Temperature

The space heating load line in Figure 34 shows that heating operation starts to occur when the space temperature is only a few degrees less than the ambient temperature. The plot uses the indoor-to-outdoor temperature difference to compensate for the fact that three different heating set points were used over the monitoring period. The impact of heat recovery on gas use has apparent in the plot. The trend indicates that, at a 20°F temperature difference, heat recovery lowers AHU gas use by 30 therms/day. We are seeing heat recovery savings around 50-60 therms/day at a temperature difference of 50°F, which is consistent with the 5-6 MMBtu/day of measured heat recovery supplied by the Unifin (as shown the Appendix B tables).

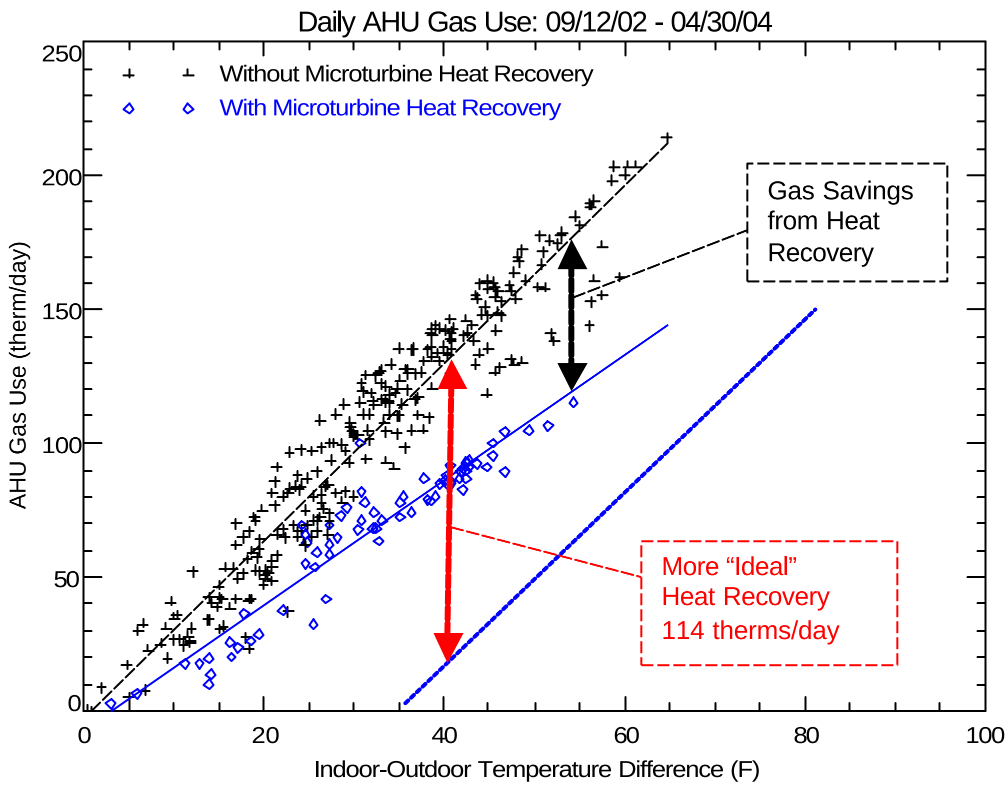


Figure 34. AHU Space Heating Gas Use Variation With Indoor-Outdoor Temperature Difference

On a daily basis, the heat recovery system did not perform as expected in the space heating mode. Figure 19 showed that the heat recovery coil could deliver about 380 MBtu/h, which should displace about 4.75 therms per hour from the furnace section (assuming 80% efficiency). This should equate to as much as 114 therms per day for a peak heating day. The measured displacement in Figure 34 was about half this amount. The more ideal heat recovery trend is shown on the plot as a dotted line.

In order to realize the additional heat recovery in the "ideal mode" the space heating controls would have to modified from the current settings. For the current system, the heat recovery coil was activated at a space temperature about 1°F below the furnace set point[3]. However, even though the heat recovery coil comes on as first stage, the furnace and coil frequently run together. Simultaneous furnace and coil operation decreases the amount of heat recovery that could be provided to the load. Increasing the differential between the heat recovery and furnace set points would give the heat recovery coil more chance to meet the space heating load and more closely approach the ideal mode.

The dehumidification trend indicates that dehumidification operation will occur when the daily average ambient humidity ratio rises above 50 gr/lb. When the heat recovery system was active, dehumidification gas use was reduced since the air was preheated to about 150-160°F. The regression analysis in the figure below predicts that heat recovery decreases daily gas use by 47 therms at 120 gr/lb and by 28 therms at 65 gr/lb. These savings are "on the order" of the total measured heat recovery of 5 MMBTU (or 50 therms) per day. The 10% higher heat recovery energy is partially explained by thermal losses from the glycol piping. The data in Figure 19 show that thermal losses are about 20-30 MBtu/h or about 0.5-0.7 MMBTU per day. Factoring in the impact of losses puts the heat recovery energy slightly below the gas use, as would be expected.

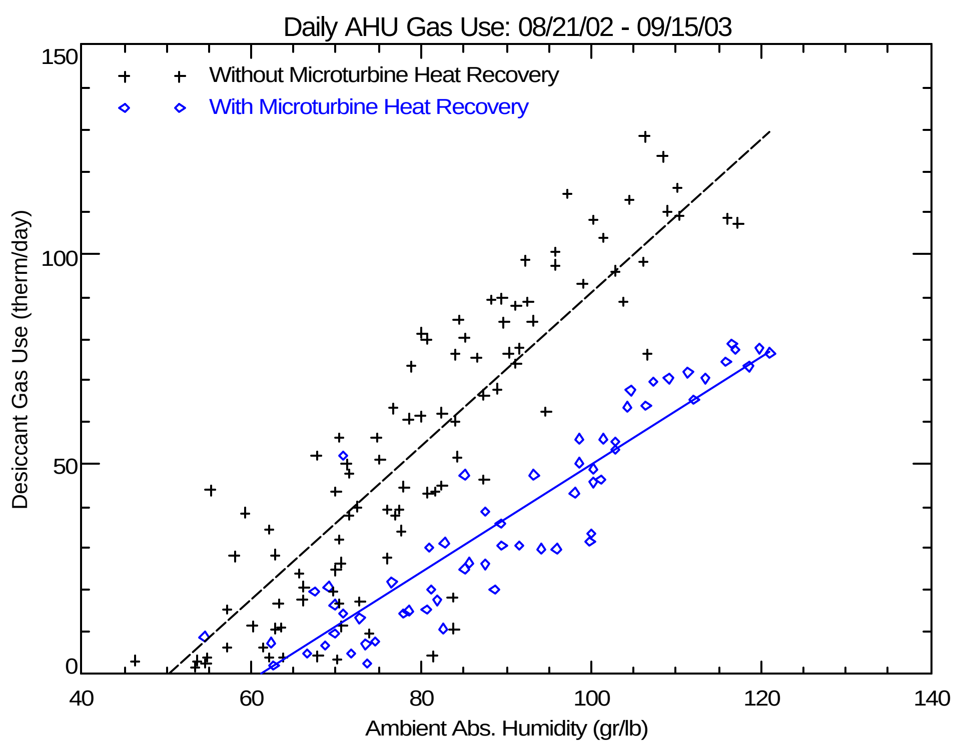


Figure 35. AHU Dehumidification Gas Use Variation With Ambient Humidity

[3] Due to various set point changes in the store, the heat recovery set point was sometimes even the same or higher than the furnace set point.

The AHU electricity use trend with ambient in Figure 33 shows more scatter than we typically observe for cooling equipment in conventional buildings. This additional scatter is in part caused by operation of the desiccant module. To demonstrate this impact, Figure 36 below shows how compressor energy use (i.e., AHU power with supply, process and regeneration power removed) varies with the ambient temperature. The days in the plot are also grouped according to the number of hours the desiccant unit operated. The regression analysis on the plot shows that compressor energy use is driven by both outdoor temperature (TAO) and desiccant runtime (RD).

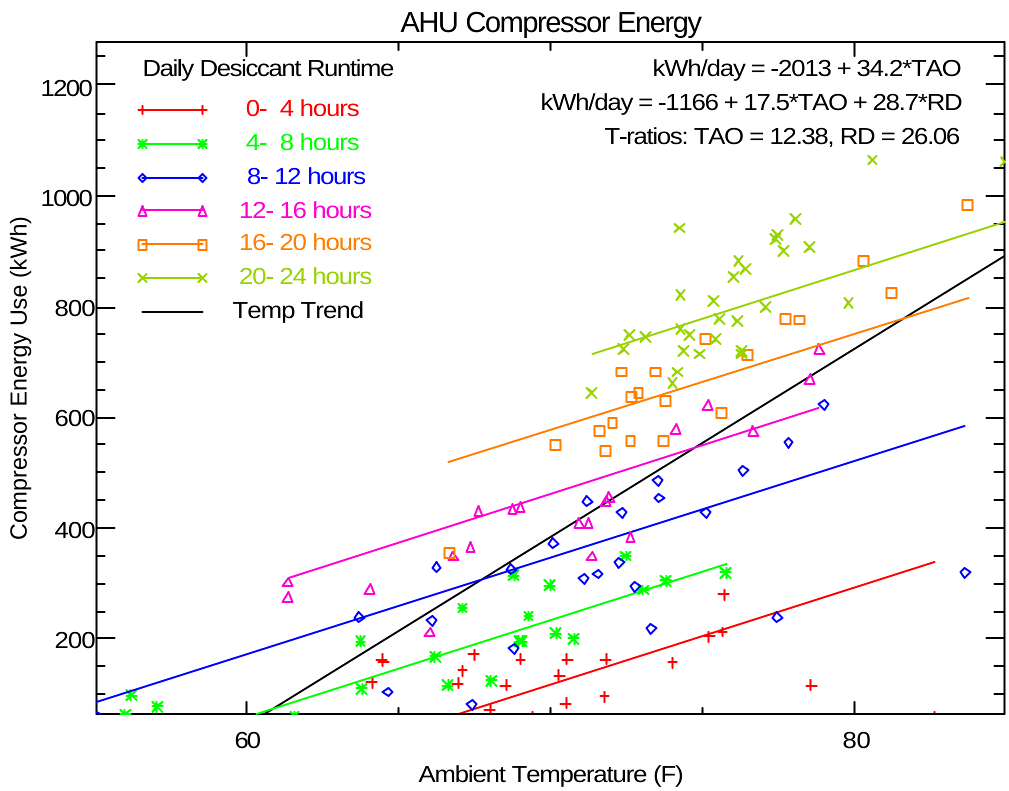


Figure 36. AHU Compressor Electricity Use with Ambient Temperature and Desiccant Runtime

The regression analysis in Figure 36 above attempts to discern the impact of these two factors. The regression results show that the compressor power increases by 34.2 kWh/day for each 1°F increase in ambient temperature, if the desiccant runtime is ignored (the black line). However, more than half of this variation is linked to increased desiccant operation. When considering days with similar amounts of desiccant operation, the compressor power slope decreases to 17.5 kWh/day per °F (as shown by the multiple colored lines). The t-ratios from the multi-linear regression coefficients are both much larger than 2, which indicate that the trends are statistically significant. Therefore, the operation of the desiccant unit has a significant impact on compressor runtime, since it adds sensible heat into the store. When the desiccant unit runs all the time, it increases compressor energy use by as much as 680 kWh/day.

Table 11. Energy Use "Slope" with Ambient: Various Scenarios

	Slope (kWh/day per °F)
Total AHU (Compressor and Supply, Process & Regen Fans) (Eqn in Figure 33)	58.6
Compressor Only: ignoring desiccant operation (1^{st} eqn in Figure 36)	34.2
Compressor Only: considering desiccant operation (2^{nd} eqn in Figure 36)	17.5

While operation of the desiccant unit increases energy use of the cooling system, it is expected to decrease refrigeration system energy use by lowering space humidity levels. On balance, the net impact of the desiccant system should be to lower energy costs for the total store. The refrigeration savings from operating at lower humidity levels are expected to more than offset the added air conditioning and dehumidification costs. While we are not measuring refrigeration system energy use at this store, measured data from other similar stores have confirmed the impact of lower humidity levels on refrigeration system energy use.

4.8 DESICCANT MODULE PERFORMANCE

Several aspects of desiccant module performance were measured at this site. These findings were presented is a separate report on the desiccant module that is included here as Appendix F. The report summarizes dehumidification performance details such as desiccant wheel grain depression, air flow rates, store ventilation rates, space humidity levels and other issues related to that system.

One result that was relevant to CHP operation was that the ability of the modulating burner controls to modulate gas use when heat recovery was available. The Munters burner controls were shown to successfully maintain the required reactivation temperature of 120°F leaving the wheel most of the time. This allowed the system to successfully take advantage of all the available heat recovery. The leaving temperature only exceeded the 120°F control point by about 5°F for a few hours at the hottest times of the year. At extreme conditions the burner gas valve was at the minimum setting and was unable to further reduce gas use.

4.9 MICROTURBINE EMISSIONS MEASUREMENTS

At part of the ETV testing, SRI took emissions measurements at the exhaust stack of the Unifin HX. Testing took place on June 4-5, 2003. This data is fully reported in Section 2 of the ETV report[4]. In addition CDH Energy staff took followup readings of NO_x and CO at the same location in the stack in September 2003 and June 2004. The CDH readings were taken using a hand-held Testo 350XL that was rented from Clean Air Engineering (www.cleanair.com). The Testo 350 XL was only able to measure Nitrogen Oxides (NO_x), and Carbon Monoxide (CO). Total Hydrocarbons (THC) were not measured. Appendix E includes the raw data and the accuracy details for the Testo instrument.

Table 12 compares the measured emissions data to the rated performance from Capstone. All readings were corrected to 15% O_2. The CDH readings in September 2003 were taken with a higher resolution version of the instrument and were very close to the higher accuracy data collected by SRI. The second CDH readings in June 2004 were higher than the previous readings but were also taken with a lower accuracy instrument. The carbon monoxide (CO) showed a significant change compared to the previous readings in September. This change may have been linked to the change out of the engine 6 months prior to testing. The slight change in the measured oxygen (O_2) concentration (from 17.3 to 17.8%) between September and June also might imply a slight change in the combustion settings when the engine was replaced. In all cases the measured emissions levels were consistently below the values provided by Capstone.

Table 12. Summary of Measured Emissions Compared to Capstone Specifications

		Pollutant Concentration (ppmv @ 15% O_2)		
	% O_2	**NO_x**	**CO**	**THC**
Capstone Rated Performance		*< 9*	*< 40*	*< 9*
SRI Testing on 6/4/03 (from ETV report, runs 1-6, reported on a dry basis)	17.8	3.1	3.7	0.9
CDH Testing on 9/17/03 Testo 350XL (w/ low NO_x & low CO ranges, ±2 ppm)	17.2	4.6	3.1	na
CDH Testing on 6/9/04 Testo 350XL (std NO_x & CO range, ±5 ppm)	17.8	6	19	na

Notes: See Appendix E for raw data and instrument accuracy details

[4] ETV report at www.sri-rtp.com/Capstone_Turbine_test.htm

5 ANNUAL CHP ANALYSIS

The data presented in the previous sections showed that the CHP system was often down or disabled during the monitoring period. This section uses the performance trends for building loads and equipment performance developed from the measured data in Section 4 to predict the annual performance of the system. The energy, environmental and economic performance of the system is evaluated using typical meteorological year (TMY) data for LaGuardia Airport.

5.1 MODELING APPROACH

The analysis in the previous sections determined how loads and equipment performance varies with driving factors such as ambient temperature and humidity. This information is driven by TMY weather data for New York and utility rate information to evaluate system performance on an annual basis.

Store Demand Profile

The variation of daily store energy use with ambient temperature was given by the regression model on Figure 10 in the previous section. This linear model is used with TMY data to predict average energy use for each hour. Then to account for the hourly variations in demand across the day, the power is adjusted to reflect on and off peak consumption. Based on measured data from the site, the power use typically increases by 20 kW for hours between 7 am to midnight, and decreased by 40 kW from midnight to 6 am. These adjustments result in roughly the same daily average energy use but mimic the daily demand profile we have observed for the building. Figure 37 shows the predicted power for each hour of the day using this approach. The two trends corresponding to on-peak and off-peak periods is shown on the plot. This “synthetic” building power profile is used to predict the building demand for each hour.

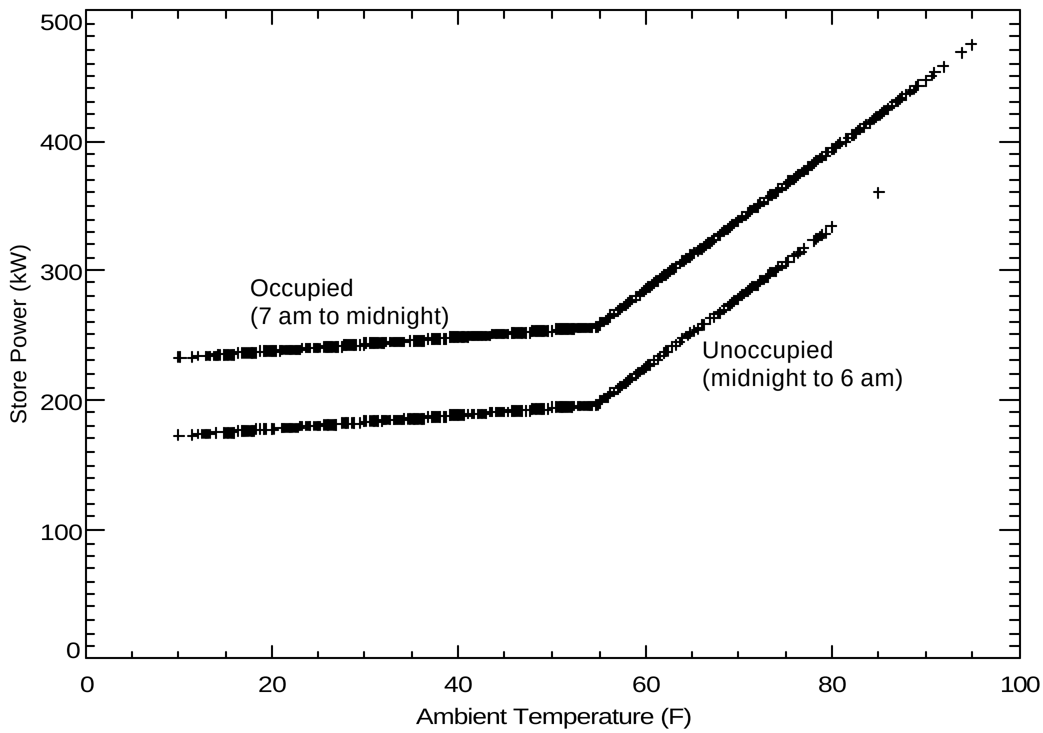


Figure 37. Predicted Variation in Supermarket Power with Ambient Temperature

Microturbine Performance

The trends of gas consumption and power output with ambient temperature for the Capstone microturbine were given in the previous section in Figure 14. The blue line on the plot showed the expected power output based on the manufacturer's specifications. The measured turbine output is 57 kW for ambient temperatures below 70°F compared to the expected power output of 60 kW. The measured output is low in part because of the voltage drop in the long wiring run to the main distribution panel. Based on the ETV test results, we estimated the voltage losses to be 1.4 kW. If a more normal wiring run (or a larger wire size) had been used, the losses would have been closer to zero. Therefore, for this analysis, we have added 1.4 kW to the measured output to make the peak output 58.4 kW.

The microturbine efficiency is the ratio of the power output divided by gas input (based on higher heating value). Figure 16 in the previous section shows the trend of measured efficiency with ambient temperature. The blue line shows the expected relationship from the manufacturer (also corrected to higher heating value). The measured turbine efficiency trend shows more degradation at higher ambient temperatures. For the manufacturer's data, the efficiency is not allowed to exceed 25.2%. For the measured trend, we have let the efficiency reach as high as 26.7%, consistent with measured data for the new engine. The analysis uses the measured trends for the new engine except where otherwise stated. Table 13 summarizes the turbine performance models used in the annual analysis.

Table 13. Turbine Performance Curves Used for Annual Analysis

Capstone Power Output		
-	Factory Specifications:	60 kW for TAO = 80°F Decrease linearly to 53 kW at 100°F
-	Measured Data:	58 kW for TAO = 70°F Decrease linearly to 46 kW at 95°F
Capstone Efficiency (HHV)		
-	Factory Specifications:	25.2% for TAO = 59°F Decrease linearly to 22.8% at 100°F
-	Measured (OLD Engine):	EFF = 29.78 - 0.0815*TAO EFF = 25.7% (capped at 50°F)
-	Measured (NEW Engine):	EFF = 30.0 -0.066*TAO EFF = 26.7% (capped at 50°F)

Heat Recovery Performance

The heat recovery system provides heat for both space heating and desiccant regeneration. Figure 19 in the previous section shows the heat recovery rate for space heating is about 380 MBtu/h while the recovery rate is only 220 MBtu/h for the desiccant regeneration. The heat recovery rate is higher for space heating since the air temperature entering the coil is lower and the air flow rate is higher than for the regeneration coil.

More important is the displaced gas use resulting from heat recovery operation. This was determined in the previous section by comparing gas use of the AHU in each season with and without the heat recovery system operating. Figure 34 from the previous showed the impact of heat recovery on gas use for the furnace section. Gas use with and without heat recovery operation showed a strong linear trend with ambient temperature. Figure 35 shows that desiccant gas use demonstrates a strong linear trend with ambient humidity (in gr/lb). Multi-linear regression analysis was used to discern the impact of desiccant heat recovery. This model, shown as the lines on Figure 35, is used to predict the net gas savings due to regeneration heat recovery at various ambient conditions.

Desiccant Gas Use:

therms/day = -93.4374 + W*1.84148 + HR*[14.0044 - W*0.549256]

where:
W - Ambient humidity level (gr/lb)
HR - Heat recovery flag (1=ON, 0=OFF)

For space heating gas use, the two regression lines on from Figure 34 are:

Space Heating Gas Use:

therms/day	=	$-3.03 + (T_{in}-T)* 3.3333639$	*(without heat recovery)*
therms/day	=	$-7.28 + (T_{in}-T)* 2.3450950$	*(with heat recovery)*

where: T - Ambient Temperature (°F)
T_{in} - Indoor Temperature (°F)

As discussed above, the amount of gas use displaced by space heating heat recovery was much lower than expected due to the furnace and heat recovery coil set points selected by the store. At steady state the heat delivered to the space heating heat recovery coil is about 380 MBtu/h. We can assume that the furnace combustion efficiency is about 80%, so the displaced gas use should be about 4.75 therms per hour (or 114 therms/day). The dotted blue line on Figure 34 shows the assumed gas use trend with more "ideal" heat recovery that would be realized with more optimal space heating set points. The resulting linear model for ideal heat recovery is:

therms/day $= -(3.03+114.0) + (T_{in}-T)* 3.3333639$ *(with "ideal" heat recovery)*

The trend above in Figure 34 and Figure 35 are based on daily average gas use. To properly predict the hourly variations in space heating gas use, the impact of thermostat setback in the store needs to be considered. The store has a 4°F temperature setback for approximately 8 hours each night (11 pm-7 am). Temperature setback was simulated by shifting the daily average space heating line (with a balance point of 72°F) and higher and lower for the occupied and unoccupied periods, respectively. For the eight hour set back period the balance point was lowered to 69.4°F and for the 16 hour occupied period the balance point was increased to 73.4°F. This shift in the load line closely mimicked the hourly impact of thermostat setback while still providing the proper average temperature across the day.

The model assumed that both heating and dehumidification could occur in the same hour. Simultaneous operation occurs at temperatures near 72°F (the heating threshold) and humidity levels above 50 gr/lb (the dehumidification threshold). While simultaneous operation is not likely to occur in practice, this approach mimicked the overall measured heating and dehumidifier runtimes.

Gas and Electric Utility Rates

The microturbine at the store has its own gas service that is currently on Keyspan Rate 260 (High Load Factor Service). The natural gas rate includes transportation charges as well as commodity. Gas use for the rest of the store is currently on Keyspan Rate 170 (Space Heating) and applies to the Munters AHU gas service as well as the rest of the store. The schedule for both Rates 260 and 170 summarized in Appendix G.

The supermarket is currently purchasing power under LIPA Rate 285 (Secondary). The rate has three energy periods (peak, off-peak and intermediate) for peak demand and energy charges. There is also a service charge and meter charge per day. The details of this rate are given in Appendix G.

Annual Simulation Approach

The annual simulations were driven by hourly typical meteorological year (TMY) data for New York LaGuardia Airport. The hourly ambient temperature and humidity data are used with the relationships described above to predict turbine power output and gas use as well as displaced gas use for space heating and desiccant drying. The synthetic electric demand profile was used with electric utility rates to predict the impact of the CHP system on monthly and annual costs. We also added in a base gas load of 2000 therms per month to account for ovens and other internal loads in the facility. This simulation approach was applied various operating scenarios to estimate the energy use, environmental impacts, and cost benefits associated with the CHP system.

For the economic analysis the turbine (and CHP system) maintenance cost was estimated to be $0.01/kWh based on feedback from the manufacturer.

The annual analysis also quantified the impact of CHP operation on net emissions from the store. This analysis requires estimates of NO_x and CO_2 emissions from the microturbine (in lb per kWh out), the burners being displaced by heat recovery (lb per therm input), and the local utility power plants (lb per kWh). The assumptions used in the analysis are given in Table 14. The displaced utility emissions change slightly during the on-peak periods since more gas turbines are included in the state-wide mix for those hours.

Table 14. Assumed Emission Rates for Annual Analysis

System	**NO_x**	**CO_2**	**Source of Data**
Capstone Emissions	0.148 lb/MWh	1.52 lb/kWh	SRI measurements on June 2003
Regen Burner/Furnace	0.01 lb/therm	11.6 lb/therm	Assumed in SRI analysis
NY Utility Grid – On-Peak	2.1 lb/MWh	1.42 lb/kWh	Used by SRI (from DOE/EIA)
NY Utility Grid – Off-Peak	2.8 lb/MWh	1.64 lb/kWh	Used by SRI (from DOE/EIA)

Notes: On-peak times for the utility are assumed to be 8 am to 6 pm, Monday-Friday

5.2 SIMULATION RESULTS: SUPERMARKET IN HAUPPAUGE

The analysis was completed for a number of hardware configurations, operating scenarios, and utility costs. Table 15 shows the results for the case which includes the standard assumptions described above, and most closely matches the store as it operated. The main assumptions include:

- Measured turbine performance for the original engine,
- Measured heat recovery trends,
- Continuous 24-hour operation,
- KeySpan rates with commodity costs listed in Appendix G.

With this system , the annual CHP efficiency for the systems was 38% and the net annual savings were a negative $11,592 per year. Operating the microturbine continuously reduces electric costs by $45,009 per year and heat recovery displaces $13,590 in annual gas costs. The average value of the displaced electric and gas consumption is 9.8¢/kWh and 1.136/therm, respectively. However, these savings are offset by annual turbine gas costs of $65,176 and annual maintenance costs of $5,016. The average cost of gas consumed by the turbine is $0.948/therm.

The main reason for no savings was the modest heat recovery savings predicted for space heating. If space heating is assumed to be more ideal, as described above, the annual savings improve substantially. This case is shown in Table 16. The annual CHP efficiency increases to 52% and the net savings for the year are a positive $2,488.

Hardware Configuration Options

The change in the performance characteristics with the new engine installed also had a big impact. Table 17 shows the results for this case. The annual CHP efficiency increases to 54% and the annual savings increase more than $2,800 to $5,307.

As a reality check, we also used the manufacturers published specifications for power output and efficiency in the simulations. This case is shown in Table 18. The results were very similar to the original engine curves based on measured data. The results are slightly worse with the Capstone curves. The catalog turbine performance is worse because the efficiency is assumed to be capped at lower ambient temperatures while the measured efficiency curves get slightly better lower at colder conditions.

Table 15. Annual Simulation Results Basecase (Old Engine, Measured HR)

CHP Efficiency

		[1]	[2]	[3]	[4]	[5]	[6]		
	Demand Reduction (min kW)	Turbine Output (kWh)	Gas Comp & Pump Parasitics (kWh)	Turbine Gas Input (therms)	QDES Heat Recovery (therms)	QHT Heat Recovery (therms)	TOTAL Displaced Gas Use (therms)	Gross Turbine Eff (-)	CHP Eff (-)
January	58	43,152	3,553	5,733	0	1,095	1,369	26%	43%
February	58	38,976	3,209	5,176	-	949	1,186	26%	42%
March	58	43,152	3,553	5,742	2	884	1,107	26%	39%
April	55	41,741	3,438	5,605	35	622	814	25%	35%
May	54	43,075	3,553	5,899	285	415	818	25%	35%
June	46	40,534	3,438	5,763	659	130	856	24%	36%
July	47	40,815	3,553	5,907	921	11	983	24%	37%
August	48	41,045	3,553	5,911	900	58	1,020	24%	38%
September	51	41,106	3,438	5,796	635	165	875	24%	36%
October	52	43,074	3,553	5,884	279	438	841	25%	35%
November	56	41,750	3,438	5,596	44	654	864	25%	36%
December	58	43,152	3,553	5,734	14	971	1,228	26%	41%
ANNUAL		***501,572***	***41,829***	***68,747***	***3,773***	***6,391***	***11,961***	***25%***	***38%***

Notes: Gas Use and Efficiency Based on Higher Heating Value (HHV)

Turbine Efficiency = 100x(3.413x**[1]** / (100x**[3]**)

CHP Efficiency = 100x(3.413x(**[1]** - **[2]**) + 100x(**[4]** + **[5]**)) / (100x**[3]**)

CHP Emissions

	Nitrogen Oxides (NO_x) Emissions				Carbon Dioxide (CO_2) Emissions			
	Displaced by Capstone Power (lbs)	Displaced by Heat Recovery (lbs)	Total Displaced (lbs)	Site Impact	Displaced by Capstone Power (lbs)	Displaced by Heat Recovery (lbs)	Total Displaced (lbs)	Site Impact
January	95.6	13.7	109.3	11%	1,720	15,890	17,610	2.0%
February	86.7	11.9	98.5	11%	1,660	13,760	15,420	1.8%
March	96.4	11.1	107.5	11%	1,990	12,840	14,830	1.7%
April	92.6	8.1	100.7	10%	1,700	9,440	11,140	1.2%
May	95.8	8.2	104.0	10%	1,860	9,490	11,350	1.2%
June	90.3	8.6	98.9	9%	1,870	9,930	11,800	1.2%
July	90.2	9.8	100.1	8%	1,710	11,400	13,110	1.3%
August	91.5	10.2	101.7	9%	1,950	11,830	13,780	1.4%
September	91.1	8.8	99.9	9%	1,710	10,150	11,860	1.2%
October	95.4	8.4	103.9	10%	1,730	9,760	11,490	1.2%
November	93.4	8.6	102.1	10%	1,960	10,020	11,980	1.3%
December	95.6	12.3	107.9	11%	1,720	14,240	15,960	1.8%
ANNUAL	***1,114.7***	***119.6***	***1234.3***	***10%***	***21,580***	***138,750***	***160,330***	***1.4%***

Notes: Assumed NOx emission rates are 0.148/MWh for Capstone, 2.1 & 2.8 lb/MWh for ON & OFF-Peak Grid

Assumed CO_2 emission rates are 1.52/kWh for Capstone, 1.42 & 1.64 lb/kWh for ON & OFF-Peak Grid

Space Heating and desiccant burners assumed to be: NOx - 0.01lb/therm, CO_2 - 11.6 lb/therm

	Basecase Building			Building with CHP			Savings				Microturbine				
Month	**Peak Demand (kW)**	**Grid Import (kWh)**	**Electric Costs**	**Peak Demand (kW)**	**Grid Import (kWh)**	**Electric Costs**	**Total Electric Savings (kWh)**	**Total Electric Cost Savings**	**Gas Savings (therms)**	**Gas Cost Savings**	**Turbine Output (kWh)**	**Maint. Cost**	**Turbine Gas Use (therms)**	**Turbine Gas Cost**	**Net Savings**
January	288.5	167,780	$15,164	235.3	128,180	$11,714	39,599	$3,450	1,369	$1,473	43,152	$432	5,733	$5,092	($600)
February	274.8	152,008	$13,820	221.5	116,240	$10,673	35,767	$3,147	1,186	$1,237	38,976	$390	5,176	$4,444	($450)
March	324.2	172,878	$15,788	271.0	133,279	$12,337	39,599	$3,450	1,107	$1,275	43,152	$432	5,742	$5,536	($1,243)
April	369.6	178,241	$16,490	319.4	139,938	$13,160	38,303	$3,330	814	$999	41,741	$417	5,605	$5,835	($1,923)
May	378.5	203,653	$18,596	329.3	164,131	$15,175	39,522	$3,421	818	$918	43,075	$431	5,899	$5,426	($1,519)
June	450.7	233,948	$33,770	409.5	196,852	$29,345	37,096	$4,425	856	$992	40,534	$405	5,763	$5,430	($419)
July	436.9	259,240	$35,614	394.7	221,978	$31,159	37,262	$4,455	983	$1,281	40,815	$408	5,907	$6,590	($1,262)
August	432.9	254,498	$35,099	389.6	217,006	$30,593	37,492	$4,506	1,020	$1,274	41,045	$410	5,911	$6,273	($903)
September	401.2	225,707	$31,585	355.0	188,039	$26,955	37,668	$4,630	875	$947	41,106	$411	5,796	$5,083	$83
October	396.4	201,411	$18,508	349.2	161,890	$15,099	39,521	$3,409	841	$890	43,074	$431	5,884	$5,093	($1,225)
November	355.8	176,117	$16,224	304.6	137,805	$12,888	38,312	$3,337	864	$957	41,750	$418	5,596	$5,156	($1,280)
December	283.7	170,215	$15,330	230.5	130,616	$11,880	39,599	$3,450	1,228	$1,347	43,152	$432	5,734	$5,217	($851)
Annual	*450.7*	*2,395,696*	*$265,988*	*409.5*	*1,935,953*	*$220,979*	*459,743*	*$45,009*	*11,961*	*$13,590*	*501,572*	*$5,016*	*68,747*	*$65,176*	*($11,592)*
			$0.111			$0.114		$0.098		$1.136				$0.948	

Table 16. Annual Simulation Results (Old Engine, <u>Ideal HR</u>)

CHP Efficiency

	[1]	[2]	[3]	[4]	[5]	[6]		
	Demand Reduction (min kW)	Turbine Output (kWh)	Gas Comp & Pump Parasitics (kWh)	Turbine Gas Input (therms)	QDES Heat Recovery (therms)	QHT Heat Recovery (therms)	TOTAL Displaced Gas Use (therms)	Gross Turbine Eff (-)	CHP Eff (-)
January	58	43,152	3,553	5,733	0	2,683	3,354	26%	70%
February	58	38,976	3,209	5,176	-	2,416	3,020	26%	70%
March	58	43,152	3,553	5,742	2	2,410	3,014	26%	66%
April	55	41,741	3,438	5,605	35	1,683	2,140	25%	54%
May	54	43,075	3,553	5,899	285	991	1,538	25%	44%
June	46	40,534	3,438	5,763	659	224	973	24%	37%
July	47	40,815	3,553	5,907	921	13	986	24%	37%
August	48	41,045	3,553	5,911	900	93	1,064	24%	38%
September	51	41,106	3,438	5,796	635	310	1,055	24%	38%
October	52	43,074	3,553	5,884	279	1,059	1,618	25%	46%
November	56	41,750	3,438	5,596	44	1,783	2,276	25%	56%
December	58	43,152	3,553	5,734	14	2,573	3,230	26%	69%
ANNUAL		***501,572***	***41,829***	***68,747***	***3,773***	***16,238***	***24,269***	***25%***	***52%***

Notes: Gas Use and Efficiency Based on Higher Heating Value (HHV)
Turbine Efficiency = 100x(3.413x**[1]** / (100x**[3]**)
CHP Efficiency = 100x(3.413x(**[1]** - **[2]**) + 100x(**[4]** + **[5]**)) / (100x**[3]**)

CHP Emissions

	Nitrogen Oxides (NO_x) Emissions				Carbon Dioxide (CO_2) Emissions			
	Displaced by Capstone Power (lbs)	Displaced by Heat Recovery (lbs)	Total Displaced (lbs)	Site Impact	Displaced by Capstone Power (lbs)	Displaced by Heat Recovery (lbs)	Total Displaced (lbs)	Site Impact
January	95.6	33.5	129.1	13%	1,720	38,910	40,630	4.6%
February	86.7	30.2	116.9	13%	1,660	35,040	36,700	4.3%
March	96.4	30.1	126.6	13%	1,990	34,970	36,960	4.2%
April	92.6	21.4	114.0	12%	1,700	24,830	26,530	3.0%
May	95.8	15.4	111.2	11%	1,860	17,840	19,700	2.1%
June	90.3	9.7	100.1	9%	1,870	11,290	13,160	1.3%
July	90.2	9.9	100.1	8%	1,710	11,430	13,140	1.3%
August	91.5	10.6	102.2	9%	1,950	12,340	14,290	1.4%
September	91.1	10.6	101.7	9%	1,710	12,240	13,950	1.4%
October	95.4	16.2	111.6	11%	1,730	18,770	20,500	2.2%
November	93.4	22.8	116.2	12%	1,960	26,400	28,360	3.2%
December	95.6	32.3	127.9	13%	1,720	37,470	39,190	4.4%
ANNUAL	***1,114.7***	***242.7***	***1357.4***	***11%***	***21,580***	***281,530***	***303,110***	***2.7%***

Notes: Assumed NOx emission rates are 0.148/MWh for Capstone, 2.1 & 2.8 lb/MWh for ON & OFF-Peak Grid
Assumed CO_2 emission rates are 1.52/kWh for Capstone, 1.42 & 1.64 lb/kWh for ON & OFF-Peak Grid
Space Heating and desiccant burners assumed to be: NOx - 0.01lb/therm, CO_2 - 11.6 lb/therm

	Basecase Building			Building with CHP			Savings				Microturbine				
Month	**Peak Demand (kW)**	**Grid Import (kWh)**	**Electric Costs**	**Peak Demand (kW)**	**Grid Import (kWh)**	**Electric Costs**	**Total Electric Savings (kWh)**	**Total Electric Cost Savings**	**Gas Savings (therms)**	**Gas Cost Savings**	**Turbine Output (kWh)**	**Maint. Cost**	**Turbine Gas Use (therms)**	**Turbine Gas Cost**	**Net Savings**
January	288.5	167,780	$15,164	235.3	128,180	$11,714	39,599	$3,450	3,354	$3,624	43,152	$432	5,733	$5,092	$1,552
February	274.8	152,008	$13,820	221.5	116,240	$10,673	35,767	$3,147	3,020	$3,198	38,976	$390	5,176	$4,444	$1,511
March	324.2	172,878	$15,788	271.0	133,279	$12,337	39,599	$3,450	3,014	$3,528	43,152	$432	5,742	$5,536	$1,011
April	369.6	178,241	$16,490	319.4	139,938	$13,160	38,303	$3,330	2,140	$2,701	41,741	$417	5,605	$5,835	($221)
May	378.5	203,653	$18,596	329.3	164,131	$15,175	39,522	$3,421	1,538	$1,768	43,075	$431	5,899	$5,426	($669)
June	450.7	233,948	$33,770	409.5	196,852	$29,345	37,096	$4,425	973	$1,132	40,534	$405	5,763	$5,430	($278)
July	436.9	259,240	$35,614	394.7	221,978	$31,159	37,262	$4,455	986	$1,285	40,815	$408	5,907	$6,590	($1,258)
August	432.9	254,498	$35,099	389.6	217,006	$30,593	37,492	$4,506	1,064	$1,332	41,045	$410	5,911	$6,273	($845)
September	401.2	225,707	$31,585	355.0	188,039	$26,955	37,668	$4,630	1,055	$1,153	41,106	$411	5,796	$5,083	$289
October	396.4	201,411	$18,508	349.2	161,890	$15,099	39,521	$3,409	1,618	$1,765	43,074	$431	5,884	$5,093	($351)
November	355.8	176,117	$16,224	304.6	137,805	$12,888	38,312	$3,337	2,276	$2,595	41,750	$418	5,596	$5,156	$357
December	283.7	170,215	$15,330	230.5	130,616	$11,880	39,599	$3,450	3,230	$3,589	43,152	$432	5,734	$5,217	$1,391
Annual	*450.7*	*2,395,696*	*$265,988*	*409.5*	*1,935,953*	*$220,979*	*459,743*	*$45,009*	*24,269*	*$27,671*	*501,572*	*$5,016*	*68,747*	*$65,176*	*$2,488*
			$0.111			$0.114		$0.098		$1.140				$0.948	

Table 17. Annual Simulation Results (New Engine, Ideal HR)

CHP Efficiency

	Demand Reduction (min kW)	[1] Turbine Output (kWh)	[2] Gas Comp & Pump Parasitics (kWh)	[3] Turbine Gas Input (therms)	[4] QDES Heat Recovery (therms)	[5] QHT Heat Recovery (therms)	[6] TOTAL Displaced Gas Use (therms)	Gross Turbine Eff (-)	CHP Eff (-)
January	58	43,152	3,553	5,519	0	2,683	3,354	27%	73%
February	58	38,976	3,209	4,983	-	2,416	3,020	27%	73%
March	58	43,152	3,553	5,525	2	2,410	3,014	27%	68%
April	55	41,741	3,438	5,383	35	1,683	2,140	26%	56%
May	54	43,075	3,553	5,640	285	991	1,538	26%	47%
June	46	40,534	3,438	5,462	659	224	973	25%	39%
July	47	40,815	3,553	5,576	921	13	986	25%	40%
August	48	41,045	3,553	5,586	900	93	1,064	25%	41%
September	51	41,106	3,438	5,505	635	310	1,055	25%	41%
October	52	43,074	3,553	5,629	279	1,059	1,618	26%	48%
November	56	41,750	3,438	5,376	44	1,783	2,276	27%	58%
December	58	43,152	3,553	5,519	14	2,573	3,230	27%	71%
ANNUAL		***501,572***	***41,829***	***65,703***	***3,773***	***16,238***	***24,269***	***26%***	***54%***

Notes: Gas Use and Efficiency Based on Higher Heating Value (HHV)
Turbine Efficiency = 100x(3.413x**[1]** / (100x**[3]**)
CHP Efficiency = 100x(3.413x(**[1]** - **[2]**) + 100x(**[4]** + **[5]**)) / (100x**[3]**)

CHP Emissions

	Nitrogen Oxides (NO_x) Emissions				Carbon Dioxide (CO_2) Emissions			
	Displaced by Capstone Power (lbs)	Displaced by Heat Recovery (lbs)	Total Displaced (lbs)	Site Impact	Displaced by Capstone Power (lbs)	Displaced by Heat Recovery (lbs)	Total Displaced (lbs)	Site Impact
January	95.6	33.5	129.1	13%	1,720	38,910	40,630	4.6%
February	86.7	30.2	116.9	13%	1,660	35,040	36,700	4.3%
March	96.4	30.1	126.6	13%	1,990	34,970	36,960	4.2%
April	92.6	21.4	114.0	12%	1,700	24,830	26,530	3.0%
May	95.8	15.4	111.2	11%	1,860	17,840	19,700	2.1%
June	90.3	9.7	100.1	9%	1,870	11,290	13,160	1.3%
July	90.2	9.9	100.1	8%	1,710	11,430	13,140	1.3%
August	91.5	10.6	102.2	9%	1,950	12,340	14,290	1.4%
September	91.1	10.6	101.7	9%	1,710	12,240	13,950	1.4%
October	95.4	16.2	111.6	11%	1,730	18,770	20,500	2.2%
November	93.4	22.8	116.2	12%	1,960	26,400	28,360	3.2%
December	95.6	32.3	127.9	13%	1,720	37,470	39,190	4.4%
ANNUAL	***1,114.7***	***242.7***	***1357.4***	***11%***	***21,580***	***281,530***	***303,110***	***2.7%***

Notes: Assumed NOx emission rates are 0.148/MWh for Capstone, 2.1 & 2.8 lb/MWh for ON & OFF-Peak Grid
Assumed CO_2 emission rates are 1.52/kWh for Capstone, 1.42 & 1.64 lb/kWh for ON & OFF-Peak Grid
Space Heating and desiccant burners assumed to be: NOx - 0.01lb/therm, CO_2 - 11.6 lb/therm

	Basecase Building			Building with CHP			Savings				Microturbine				
Month	**Peak Demand (kW)**	**Grid Import (kWh)**	**Electric Costs**	**Peak Demand (kW)**	**Grid Import (kWh)**	**Electric Costs**	**Total Electric Savings (kWh)**	**Total Electric Cost Savings**	**Gas Savings (therms)**	**Gas Cost Savings**	**Turbine Output (kWh)**	**Maint. Cost**	**Turbine Gas Use (therms)**	**Turbine Gas Cost**	**Net Savings**
January	288.5	167,780	$15,164	235.3	128,180	$11,714	39,599	$3,450	3,354	$3,624	43,152	$432	5,519	$4,908	$1,736
February	274.8	152,008	$13,820	221.5	116,240	$10,673	35,767	$3,147	3,020	$3,198	38,976	$390	4,983	$4,285	$1,670
March	324.2	172,878	$15,788	271.0	133,279	$12,337	39,599	$3,450	3,014	$3,528	43,152	$432	5,525	$5,334	$1,213
April	369.6	178,241	$16,490	319.4	139,938	$13,160	38,303	$3,330	2,140	$2,701	41,741	$417	5,383	$5,610	$4
May	378.5	203,653	$18,596	329.3	164,131	$15,175	39,522	$3,421	1,538	$1,768	43,075	$431	5,640	$5,195	($438)
June	450.7	233,948	$33,770	409.5	196,852	$29,345	37,096	$4,425	973	$1,132	40,534	$405	5,462	$5,156	($4)
July	436.9	259,240	$35,614	394.7	221,978	$31,159	37,262	$4,455	986	$1,285	40,815	$408	5,576	$6,230	($899)
August	432.9	254,498	$35,099	389.6	217,006	$30,593	37,492	$4,506	1,064	$1,332	41,045	$410	5,586	$5,937	($509)
September	401.2	225,707	$31,585	355.0	188,039	$26,955	37,668	$4,630	1,055	$1,153	41,106	$411	5,505	$4,836	$536
October	396.4	201,411	$18,508	349.2	161,890	$15,099	39,521	$3,409	1,618	$1,765	43,074	$431	5,629	$4,880	($137)
November	355.8	176,117	$16,224	304.6	137,805	$12,888	38,312	$3,337	2,276	$2,595	41,750	$418	5,376	$4,960	$554
December	283.7	170,215	$15,330	230.5	130,616	$11,880	39,599	$3,450	3,230	$3,589	43,152	$432	5,519	$5,028	$1,580
Annual	*450.7*	*2,395,696*	*$265,988*	*409.5*	*1,935,953*	*$220,979*	*459,743*	*$45,009*	*24,269*	*$27,671*	*501,572*	*$5,016*	*65,703*	*$62,357*	*$5,307*
Avg Cost ($/kWh or $/therm)			$0.111			$0.114		$0.098		$1.140				$0.949	

Table 18. Annual Simulation Results (Capstone Engine Specs, Ideal HR)

CHP Efficiency

	[1]	[2]	[3]	[4]	[5]	[6]		
	Demand Reduction (min kW)	Turbine Output (kWh)	Gas Comp & Pump Parasitics (kWh)	Turbine Gas Input (therms)	QDES Heat Recovery (therms)	QHT Heat Recovery (therms)	TOTAL Displaced Gas Use (therms)	Gross Turbine Eff (-)	CHP Eff (-)
January	60	44,640	3,553	6,046	0	2,683	3,354	25%	68%
February	60	40,320	3,209	5,461	-	2,416	3,020	25%	67%
March	60	44,640	3,553	6,047	2	2,410	3,014	25%	63%
April	60	43,200	3,438	5,862	35	1,683	2,140	25%	52%
May	60	44,640	3,553	6,087	285	991	1,538	25%	44%
June	54	42,997	3,438	5,994	659	224	973	24%	37%
July	55	44,337	3,553	6,258	921	13	986	24%	37%
August	56	44,263	3,553	6,226	900	93	1,064	24%	38%
September	58	43,172	3,438	5,984	635	310	1,055	25%	38%
October	58	44,633	3,553	6,081	279	1,059	1,618	25%	45%
November	60	43,200	3,438	5,859	44	1,783	2,276	25%	54%
December	60	44,640	3,553	6,046	14	2,573	3,230	25%	66%
ANNUAL		***524,682***	***41,829***	***71,950***	***3,773***	***16,238***	***24,269***	***25%***	***51%***

Notes: Gas Use and Efficiency Based on Higher Heating Value (HHV)
Turbine Efficiency = 100x(3.413x**[1]** / (100x**[3]**)
CHP Efficiency = 100x(3.413x(**[1]** - **[2]**) + 100x(**[4]** + **[5]**)) / (100x**[3]**)

CHP Emissions

	Nitrogen Oxides (NO_x) Emissions				Carbon Dioxide (CO_2) Emissions			
	Displaced by Capstone Power (lbs)	Displaced by Heat Recovery (lbs)	Total Displaced (lbs)	Site Impact	Displaced by Capstone Power (lbs)	Displaced by Heat Recovery (lbs)	Total Displaced (lbs)	Site Impact
January	99.2	33.5	132.7	14%	1,790	38,910	40,700	4.6%
February	89.9	30.2	120.1	13%	1,720	35,040	36,760	4.3%
March	100.0	30.1	130.2	13%	2,060	34,970	37,030	4.2%
April	96.1	21.4	117.5	12%	1,760	24,830	26,590	3.0%
May	99.6	15.4	115.0	11%	1,920	17,840	19,760	2.1%
June	96.1	9.7	105.8	9%	1,910	11,290	13,200	1.3%
July	98.5	9.9	108.4	9%	1,800	11,430	13,230	1.3%
August	99.2	10.6	109.8	9%	2,070	12,340	14,410	1.4%
September	96.0	10.6	106.6	10%	1,760	12,240	14,000	1.4%
October	99.2	16.2	115.4	11%	1,790	18,770	20,560	2.2%
November	96.9	22.8	119.7	12%	2,040	26,400	28,440	3.2%
December	99.2	32.3	131.5	14%	1,790	37,470	39,260	4.4%
ANNUAL	***1,170.0***	***242.7***	***1412.7***	***11%***	***22,410***	***281,530***	***303,940***	***2.7%***

Notes: Assumed NOx emission rates are 0.148/MWh for Capstone, 2.1 & 2.8 lb/MWh for ON & OFF-Peak Grid
Assumed CO_2 emission rates are 1.52/kWh for Capstone, 1.42 & 1.64 lb/kWh for ON & OFF-Peak Grid
Space Heating and desiccant burners assumed to be: NOx - 0.01lb/therm, CO_2 - 11.6 lb/therm

	Basecase Building			Building with CHP			Savings				Microturbine				
Month	**Peak Demand (kW)**	**Grid Import (kWh)**	**Electric Costs**	**Peak Demand (kW)**	**Grid Import (kWh)**	**Electric Costs**	**Total Electric Savings (kWh)**	**Total Electric Cost Savings**	**Gas Savings (therms)**	**Gas Cost Savings**	**Turbine Output (kWh)**	**Maint. Cost**	**Turbine Gas Use (therms)**	**Turbine Gas Cost**	**Net Savings**
January	288.5	167,780	$15,164	233.3	126,692	$11,584	41,087	$3,580	3,354	$3,624	44,640	$446	6,046	$5,360	$1,398
February	274.8	152,008	$13,820	219.5	114,896	$10,555	37,111	$3,265	3,020	$3,198	40,320	$403	5,461	$4,679	$1,380
March	324.2	172,878	$15,788	269.0	131,791	$12,208	41,087	$3,580	3,014	$3,528	44,640	$446	6,047	$5,821	$840
April	369.6	178,241	$16,490	314.4	138,479	$13,015	39,762	$3,475	2,140	$2,701	43,200	$432	5,862	$6,095	($351)
May	378.5	203,653	$18,596	323.3	162,566	$15,016	41,087	$3,580	1,538	$1,768	44,640	$446	6,087	$5,594	($693)
June	450.7	233,948	$33,770	401.5	194,389	$28,891	39,559	$4,879	973	$1,132	42,997	$430	5,994	$5,642	($60)
July	436.9	259,240	$35,614	386.7	218,456	$30,607	40,784	$5,007	986	$1,285	44,337	$443	6,258	$6,971	($1,123)
August	432.9	254,498	$35,099	381.6	213,788	$30,074	40,710	$5,025	1,064	$1,332	44,263	$443	6,226	$6,598	($684)
September	401.2	225,707	$31,585	348.0	185,973	$26,574	39,734	$5,012	1,055	$1,153	43,172	$432	5,984	$5,242	$491
October	396.4	201,411	$18,508	343.2	160,331	$14,940	41,080	$3,568	1,618	$1,765	44,633	$446	6,081	$5,258	($372)
November	355.8	176,117	$16,224	300.6	136,355	$12,749	39,762	$3,475	2,276	$2,595	43,200	$432	5,859	$5,390	$247
December	283.7	170,215	$15,330	228.5	129,128	$11,750	41,087	$3,580	3,230	$3,589	44,640	$446	6,046	$5,492	$1,231
Annual	*450.7*	*2,395,696*	*$265,988*	*401.5*	*1,912,843*	*$217,962*	*482,853*	*$48,026*	*24,269*	*$27,671*	*524,682*	*$5,247*	*71,950*	*$68,143*	*$2,306*
								$0.099		$1.140				$0.947	

The subsequent runs in this section use the performance curves for the new engine and also assume more ideal heat recovery for space heating. (the basecase is given in Table 17 above).

System Operation Options

The monthly results for continuous operation in Table 17 show that net savings are actually negative in the summer months when the amount of useful heat recovery is less. The seasonal variations imply that operating scenarios to schedule turbine operation may provide greater savings than continuous operation. Table 19 and Table 20 show the impact that scheduling CHP system operation has on overall economics. The first scenario (Table 19) assumes that the turbine only operates from 8 am to midnight each day, when the building is occupied. The net annual savings increase only slightly to $5,370. The second scenario (Table 20) assumes that the CHP system is scheduled to operate 24 hours a day in the winter so that heat recovery can be applied to the nighttime space heating loads in the store. However, operation is not allowed from midnight to 8 am each night, from April to October when space heating loads are low. This operating scenario results in net annual savings of $7,625, an increase of nearly $2,500 over continuous 24-hour operation.

Table 19. Annual Simulation Costs Results – No Night-Time Operation (midnite-7 am)

	Basecase Building			Building with CHP			Savings				Microturbine				
Month	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Total Electric Savings (kWh)	Total Electric Cost Savings	Gas Savings (therms)	Gas Cost Savings	Turbine Output (kWh)	Maint. Cost	Turbine Gas Use (therms)	Turbine Gas Cost	Net Savings
January	288.5	167,780	$15,164	279.6	141,380	$12,901	26,400	$2,263	2,258	$2,428	28,768	$288	3,679	$3,327	$1,076
February	274.8	152,008	$13,820	252.2	128,163	$11,691	23,845	$2,129	2,004	$2,090	25,984	$260	3,322	$2,912	$1,047
March	324.2	172,878	$15,788	271.0	146,479	$13,266	26,400	$2,522	1,981	$2,281	28,768	$288	3,686	$3,614	$902
April	369.6	178,241	$16,490	319.4	152,712	$14,059	25,529	$2,432	1,377	$1,709	27,821	$278	3,597	$3,804	$58
May	378.5	203,653	$18,596	329.3	177,331	$16,103	26,323	$2,492	978	$1,107	28,691	$287	3,777	$3,534	($222)
June	450.7	233,948	$33,770	409.5	209,536	$30,345	24,412	$3,425	626	$714	26,704	$267	3,622	$3,474	$398
July	436.9	259,240	$35,614	394.7	234,839	$32,165	24,402	$3,449	649	$846	26,770	$268	3,677	$4,164	($138)
August	432.9	254,498	$35,099	392.3	229,910	$31,632	24,588	$3,467	709	$886	26,956	$270	3,689	$3,977	$105
September	401.2	225,707	$31,585	355.8	200,764	$28,063	24,943	$3,522	667	$711	27,235	$272	3,666	$3,275	$685
October	396.4	201,411	$18,508	349.2	175,089	$16,028	26,322	$2,481	1,037	$1,111	28,690	$287	3,766	$3,319	($15)
November	355.8	176,117	$16,224	304.6	150,579	$13,786	25,538	$2,438	1,522	$1,705	27,830	$278	3,588	$3,366	$499
December	283.7	170,215	$15,330	270.7	143,815	$13,043	26,400	$2,287	2,171	$2,381	28,768	$288	3,680	$3,407	$973
Annual	*450.7*	*2,395,696*	*$265,988*	*409.5*	*2,090,597*	*$233,082*	*305,099*	*$32,906*	*15,977*	*$17,968*	*332,985*	*$3,330*	*43,748*	*$42,174*	*$5,370*
								$0.108		$1.125				$0.964	

Table 20. Annual Simulation Costs Results – No Night-Time Operation (midnite-7 am), April-October

	Basecase Building			Building with CHP			Savings				Microturbine				
Month	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Total Electric Savings (kWh)	Total Electric Cost Savings	Gas Savings (therms)	Gas Cost Savings	Turbine Output (kWh)	Maint. Cost	Turbine Gas Use (therms)	Turbine Gas Cost	Net Savings
January	288.5	167,780	$15,164	235.3	128,180	$11,714	39,599	$3,450	3,354	$3,624	43,152	$432	5,519	$4,908	$1,736
February	274.8	152,008	$13,820	221.5	116,240	$10,673	35,767	$3,147	3,020	$3,198	38,976	$390	4,983	$4,285	$1,670
March	324.2	172,878	$15,788	271.0	133,279	$12,337	39,599	$3,450	3,014	$3,528	43,152	$432	5,525	$5,334	$1,213
April	369.6	178,241	$16,490	319.4	152,712	$14,059	25,529	$2,432	1,377	$1,709	27,821	$278	3,597	$3,804	$58
May	378.5	203,653	$18,596	329.3	177,331	$16,103	26,323	$2,492	978	$1,107	28,691	$287	3,777	$3,534	($222)
June	450.7	233,948	$33,770	409.5	209,536	$30,345	24,412	$3,425	626	$714	26,704	$267	3,622	$3,474	$398
July	436.9	259,240	$35,614	394.7	234,839	$32,165	24,402	$3,449	649	$846	26,770	$268	3,677	$4,164	($138)
August	432.9	254,498	$35,099	392.3	229,910	$31,632	24,588	$3,467	709	$886	26,956	$270	3,689	$3,977	$105
September	401.2	225,707	$31,585	355.8	200,764	$28,063	24,943	$3,522	667	$711	27,235	$272	3,666	$3,275	$685
October	396.4	201,411	$18,508	349.2	175,089	$16,028	26,322	$2,481	1,037	$1,111	28,690	$287	3,766	$3,319	($15)
November	355.8	176,117	$16,224	304.6	137,805	$12,888	38,312	$3,337	2,276	$2,595	41,750	$418	5,376	$4,960	$554
December	283.7	170,215	$15,330	230.5	130,616	$11,880	39,599	$3,450	3,230	$3,589	43,152	$432	5,519	$5,028	$1,580
Annual	*450.7*	*2,395,696*	*$265,988*	*409.5*	*2,026,301*	*$227,887*	*369,395*	*$38,101*	*20,937*	*$23,617*	*403,049*	*$4,030*	*52,715*	*$50,063*	*$7,625*
								$0.103		$1.128				$0.950	

Another issue is the impact that unexpected equipment outages would have on system economics. To evaluate that issue, we arbitrarily assume that the CHP system will not operate for the 10th and 20th day of each month (i.e., no power or heat recovery will be provided). Shutting the system down arbitrarily for these two days per month in intended shows the average demand and energy impact of a lower availability (93.4%).

Table 21 shows that the net impact of shutting the system down for these two days is to decrease savings by nearly $2,900 per year. Figure 38 shows that arbitrarily specifying the downtime on the 10th and 20th day of each month means that the outages do not always negate the demand savings. The demand reduction from operating the microturbine is only negated by the outage in some months (in this case January, February, August and September). On these months the outage days happened when the building was near its peak demand for the month because ambient temperatures were high. For the other months the building peak demand was low enough on the outage days so that the monthly peak was not affected. This exercise demonstrates the net demand and energy impacts of providing a lower availability of 93.4%.

Table 21. Annual Simulation Costs Results – Impact of Two Outages per Month

	Basecase Building			Building with CHP			Savings				Microturbine				
Month	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Total Electric Savings (kWh)	Total Electric Cost Savings	Gas Savings (therms)	Gas Cost Savings	Turbine Output (kWh)	Maint. Cost	Turbine Gas Use (therms)	Turbine Gas Cost	Net Savings
January	288.5	167,780	$15,164	251.7	130,735	$12,012	37,045	$3,152	3,137	$3,376	40,368	$404	5,163	$4,602	$1,522
February	274.8	152,008	$13,820	252.3	118,795	$11,056	33,212	$2,764	2,805	$2,958	36,192	$362	4,627	$3,990	$1,370
March	324.2	172,878	$15,788	271.0	135,834	$12,540	37,045	$3,248	2,797	$3,262	40,368	$404	5,169	$5,001	$1,105
April	369.6	178,241	$16,490	319.4	142,493	$13,363	35,748	$3,127	1,975	$2,487	38,957	$390	5,026	$5,250	($25)
May	378.5	203,653	$18,596	329.3	166,686	$15,378	36,968	$3,218	1,397	$1,601	40,291	$403	5,280	$4,875	($458)
June	450.7	233,948	$33,770	409.5	199,305	$29,580	34,642	$4,190	906	$1,051	37,851	$379	5,099	$4,824	$39
July	436.9	259,240	$35,614	394.7	224,395	$31,438	34,846	$4,176	905	$1,180	38,169	$382	5,215	$5,838	($864)
August	432.9	254,498	$35,099	419.1	219,420	$31,677	35,078	$3,422	991	$1,239	38,401	$384	5,226	$5,565	($1,289)
September	401.2	225,707	$31,585	396.4	190,520	$28,343	35,187	$3,242	985	$1,072	38,396	$384	5,139	$4,525	($595)
October	396.4	201,411	$18,508	349.2	164,444	$15,302	36,967	$3,206	1,493	$1,624	40,290	$403	5,270	$4,579	($152)
November	355.8	176,117	$16,224	304.6	140,360	$13,090	35,757	$3,134	2,061	$2,342	38,966	$390	5,020	$4,643	$443
December	283.7	170,215	$15,330	248.5	133,170	$12,187	37,045	$3,143	3,010	$3,332	40,368	$404	5,163	$4,714	$1,357
Annual	*450.7*	*2,395,696*	*$265,988*	*419.1*	*1,966,157*	*$225,966*	*429,538*	*$40,021*	*22,463*	*$25,523*	*468,617*	*$4,686*	*61,398*	*$58,404*	*$2,454*
								$0.093		$1.136				$0.951	

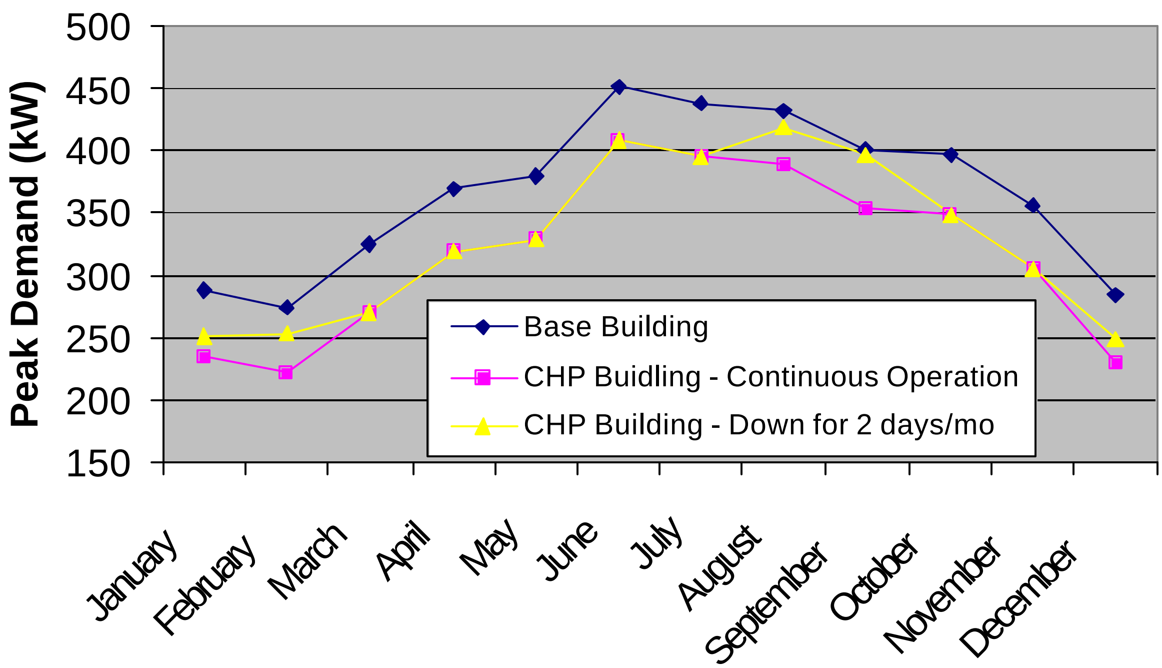


Figure 38. Impact of Two Days of Downtime per Month (Outage days on 10th and 20th)

Impact of Commodity Gas Costs

The cost of gas has a significant impact on the economics of the system. The base scenario in Table 17 assumed the commodity costs given in Appendix G, which were taken from the KeySpan gas bills for the store in 2002-2003. The average commodity gas cost was $0.714 per therm which resulted in net savings of $5,307.

Figure 39 shows the impact of different commodity gas prices on overall economics. The savings reach zero at a commodity gas cost of $0.84/therm. The annual savings increase by more than $4,500 for each $0.10 per therm drop in commodity prices. If commodity gas is $0.50/therm – a cost more consistent with the past few years – the savings exceed $15,000 per year, which is more in lines with the original project expectations.

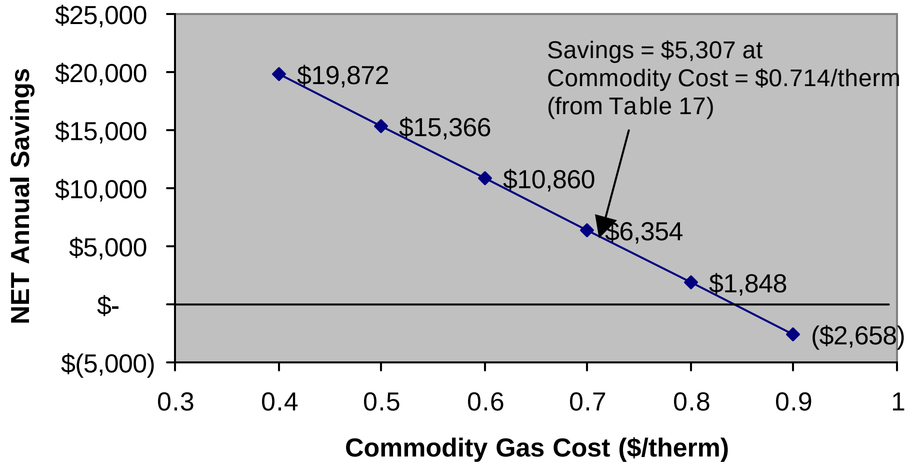


Figure 39. Impact of Commodity Gas Costs on Net Annual Savings

5.3 SIMULATION RESULTS: OTHER LOCATIONS

The model was used with other weather data and utility rates to determine the system economics for other locations around the US (see Table 22). This process is analogous to moving the store to other locations around the US and using local utility rates. TMY weather data for each location was used to drive the model.

Table 22. Summary of Weather Data and Utility Rates Used for Other Locations

Location	TMY Weather Data	Electric Rates Base / CHP	Gas Rates Store / Turbine
Long Island	NY - LaGuardia	LIPA 285	KeySpan 170 / 260
New York	NY - LaGuardia	Con Ed SC9-R1	KeySpan 170 / 260
Southern CA	Los Angeles	SCE TOU8	SoCal Gas 10
Chicago	Chicago	ComEd Gen / Standby	NICOR 4
Portland	Portland	PPL 30 / 36	NW Natural 31

Using the above locations and associated utility rates we made economic comparisons for the five locations. The turbine was assumed to run 24 hours per day, use the performance curves for the new turbine, and have ideal heat recovery (i.e., the same as Table 17 for Long Island). The savings for test site on Long Island was $5,307. Table 23 to Table 26 show the analogous results for these four other locations. Table 27 and Figure 40 compared the results from all the sites. The highest savings occur in Con Edison/KeySpan territory in New York and Southern California ($18,445 and $18,832, respectively). Chicago resulted in the next highest annual savings. In Portland the system increased operating costs by more than 20,000 per year.

Table 23. CHP Cost Savings: Supermarket located in Chicago, IL

	Basecase Building			Building with CHP			Savings				Microturbine				
Month	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Total Electric Savings (kWh)	Total Electric Cost Savings	Gas Savings (therms)	Gas Cost Savings	Turbine Output (kWh)	Maint. Cost	Turbine Gas Use (therms)	Turbine Gas Cost	Net Savings
January	251.7	164,394	$11,634	198.5	124,795	$9,308	39,599	$2,326	3,470	$3,254	43,152	$432	5,516	$4,874	$275
February	261.8	148,800	$10,942	208.6	113,033	$8,963	35,767	$1,980	3,152	$2,582	38,976	$390	4,982	$3,851	$321
March	329.1	169,543	$12,839	275.9	129,944	$10,485	39,599	$2,354	3,269	$3,183	43,152	$432	5,521	$5,307	($202)
April	369.6	178,426	$13,793	319.4	140,143	$11,718	38,283	$2,075	2,231	$1,556	41,721	$417	5,383	$4,007	($794)
May	436.9	209,998	$16,255	394.7	171,037	$13,864	38,961	$2,391	1,608	$1,134	42,514	$425	5,590	$4,060	($960)
June	455.6	236,486	$22,642	415.3	199,898	$15,934	36,588	$6,708	976	$800	40,026	$400	5,403	$4,259	$2,848
July	468.5	255,505	$24,095	429.3	218,197	$17,008	37,308	$7,087	995	$1,027	40,861	$409	5,564	$4,580	$3,125
August	445.8	250,945	$23,443	404.6	213,347	$15,931	37,598	$7,512	1,131	$1,081	41,151	$412	5,584	$4,247	$3,935
September	415.0	219,092	$20,868	369.8	181,584	$14,647	37,508	$6,221	1,163	$805	40,946	$409	5,454	$3,735	$2,882
October	374.5	196,843	$14,813	325.3	157,325	$12,276	39,518	$2,537	1,769	$1,006	43,071	$431	5,609	$3,498	($386)
November	351.8	170,703	$13,175	300.6	132,386	$10,900	38,317	$2,275	2,718	$1,819	41,755	$418	5,362	$3,598	$78
December	306.4	166,309	$12,396	253.2	126,709	$9,970	39,599	$2,425	3,422	$2,645	43,152	$432	5,519	$4,053	$585
Annual	*468.5*	*2,367,047*	*$196,896*	*429.3*	*1,908,399*	*$151,004*	*458,648*	*$45,892*	*25,904*	*$20,891*	*500,477*	*$5,005*	*65,488*	*$50,070*	*$11,708*
			$0.083			$0.079		$0.100		$0.807				$0.765	

Table 24. CHP Cost Savings: Supermarket located in Orange County, CA

	Basecase Building			Building with CHP			Savings				Microturbine				
Month	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Total Electric Savings (kWh)	Total Electric Cost Savings	Gas Savings (therms)	Gas Cost Savings	Turbine Output (kWh)	Maint. Cost	Turbine Gas Use (therms)	Turbine Gas Cost	Net Savings
January	369.6	193,452	$22,318	319.4	153,872	$18,020	39,580	$4,298	1,728	$1,468	43,133	$431	5,603	$4,227	$1,108
February	355.8	172,711	$20,105	304.6	136,952	$16,190	35,759	$3,915	1,627	$1,306	38,968	$390	5,053	$3,668	$1,163
March	346.9	192,072	$21,921	294.7	152,475	$17,630	39,597	$4,292	1,766	$1,666	43,150	$432	5,600	$4,792	$735
April	378.5	196,091	$22,650	329.3	157,846	$18,496	38,245	$4,154	1,398	$1,075	41,683	$417	5,454	$3,584	$1,228
May	396.4	213,739	$24,484	349.2	174,230	$20,226	39,509	$4,258	1,438	$1,207	43,062	$431	5,682	$3,912	$1,123
June	369.6	211,384	$33,440	319.4	173,152	$27,982	38,232	$5,458	1,447	$1,458	41,670	$417	5,518	$4,303	$2,196
July	383.4	230,450	$36,049	335.2	191,215	$30,520	39,235	$5,529	1,249	$1,329	42,788	$428	5,718	$4,276	$2,154
August	459.6	235,677	$37,121	419.4	196,817	$31,752	38,859	$5,369	1,236	$1,311	42,412	$424	5,690	$3,950	$2,306
September	392.3	222,570	$35,299	345.1	184,679	$29,929	37,891	$5,370	1,237	$1,285	41,329	$413	5,521	$4,055	$2,187
October	383.4	216,630	$24,742	335.2	177,195	$20,474	39,435	$4,268	1,372	$1,134	42,988	$430	5,685	$3,659	$1,314
November	369.6	196,931	$22,539	319.4	158,641	$18,394	38,290	$4,145	1,376	$975	41,728	$417	5,463	$3,249	$1,454
December	392.3	195,034	$22,656	345.1	155,532	$18,388	39,502	$4,269	1,664	$1,062	43,055	$431	5,599	$3,034	$1,865
Annual	*459.6*	*2,476,742*	*$323,324*	*419.4*	*2,012,605*	*$268,001*	*464,137*	*$55,323*	*17,538*	*$15,277*	*505,966*	*$5,060*	*66,584*	*$46,709*	*$18,832*
			$0.131			$0.133		$0.119		$0.871				$0.702	

Table 25. CHP Cost Savings: Supermarket located in New York (Con Ed/KeySpan territory)

	Basecase Building			Building with CHP			Savings				Microturbine				
Month	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Total Electric Savings (kWh)	Total Electric Cost Savings	Gas Savings (therms)	Gas Cost Savings	Turbine Output (kWh)	Maint. Cost	Turbine Gas Use (therms)	Turbine Gas Cost	Net Savings
January	288.5	167,780	$21,263	235.3	128,180	$16,931	39,599	$4,332	3,354	$3,624	43,152	$432	5,519	$4,908	$2,618
February	274.8	152,008	$19,506	221.5	116,240	$15,584	35,767	$3,922	3,020	$3,198	38,976	$390	4,983	$4,285	$2,445
March	324.2	172,878	$22,396	271.0	133,279	$17,608	39,599	$4,788	3,014	$3,528	43,152	$432	5,525	$5,334	$2,551
April	369.6	178,241	$23,731	319.4	139,938	$18,460	38,303	$5,271	2,140	$2,701	41,741	$417	5,383	$5,610	$1,946
May	378.5	203,653	$26,322	329.3	164,131	$21,323	39,522	$4,999	1,538	$1,768	43,075	$431	5,640	$5,195	$1,140
June	450.7	233,948	$30,524	409.5	196,852	$25,086	37,096	$5,438	973	$1,132	40,534	$405	5,462	$5,156	$1,009
July	436.9	259,240	$32,692	394.7	221,978	$27,793	37,262	$4,899	986	$1,285	40,815	$408	5,576	$6,230	($454)
August	432.9	254,498	$32,165	389.6	217,006	$27,250	37,492	$4,915	1,064	$1,332	41,045	$410	5,586	$5,937	($100)
September	401.2	225,707	$28,841	355.0	188,039	$23,995	37,668	$4,846	1,055	$1,153	41,106	$411	5,505	$4,836	$752
October	396.4	201,411	$26,430	349.2	161,890	$21,046	39,521	$5,384	1,618	$1,765	43,074	$431	5,629	$4,880	$1,838
November	355.8	176,117	$23,278	304.6	137,805	$18,119	38,312	$5,159	2,276	$2,595	41,750	$418	5,376	$4,960	$2,376
December	283.7	170,215	$21,408	230.5	130,616	$17,214	39,599	$4,193	3,230	$3,589	43,152	$432	5,519	$5,028	$2,323
Annual	*450.7*	*2,395,696*	*$308,555*	*409.5*	*1,935,953*	*$250,408*	*459,743*	*$58,148*	*24,269*	*$27,671*	*501,572*	*$5,016*	*65,703*	*$62,357*	*$18,445*
			$0.129			$0.129		$0.126		$1.140				$0.949	

Table 26. CHP Cost Savings: Supermarket located in Portland, OR

	Basecase Building			Building with CHP			Savings				Microturbine				
Month	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Peak Demand (kW)	Grid Import (kWh)	Electric Costs	Total Electric Savings (kWh)	Total Electric Cost Savings	Gas Savings (therms)	Gas Cost Savings	Turbine Output (kWh)	Maint. Cost	Turbine Gas Use (therms)	Turbine Gas Cost	Net Savings
January	270.7	171,155	$7,520	217.5	131,555	$6,548	39,599	$972	3,132	$2,662	43,152	$432	5,517	$4,434	($1,232)
February	293.4	156,702	$7,059	240.2	120,935	$6,174	35,767	$884	2,593	$2,199	38,976	$390	4,987	$3,978	($1,284)
March	324.2	176,177	$7,852	271.0	136,578	$6,759	39,599	$1,094	2,620	$2,207	43,152	$432	5,532	$4,370	($1,501)
April	360.7	176,871	$7,980	310.5	138,570	$6,852	38,301	$1,128	2,141	$1,837	41,739	$417	5,375	$4,240	($1,693)
May	405.3	196,028	$8,801	359.1	156,612	$7,533	39,416	$1,267	1,596	$1,423	42,969	$430	5,591	$4,392	($2,131)
June	410.2	208,457	$9,266	364.9	170,517	$8,041	37,940	$1,224	1,080	$1,039	41,378	$414	5,466	$4,298	($2,448)
July	459.6	231,862	$10,254	419.4	193,470	$8,904	38,392	$1,351	1,080	$1,137	41,945	$419	5,608	$4,502	($2,434)
August	436.9	221,515	$9,815	394.7	182,546	$8,494	38,969	$1,321	1,396	$1,385	42,522	$425	5,643	$4,541	($2,260)
September	432.9	207,258	$9,286	389.6	169,367	$8,012	37,891	$1,273	1,246	$1,174	41,329	$413	5,454	$4,301	($2,266)
October	378.5	189,461	$8,487	329.3	149,910	$7,274	39,551	$1,213	1,994	$1,738	43,104	$431	5,581	$4,403	($1,883)
November	297.5	170,614	$7,575	244.2	132,292	$6,589	38,322	$986	2,479	$2,099	41,760	$418	5,352	$4,234	($1,567)
December	279.6	171,724	$7,566	226.4	132,124	$6,574	39,599	$992	3,017	$2,573	43,152	$432	5,518	$4,431	($1,298)
Annual	*459.6*	*2,277,826*	*$101,460*	*419.4*	*1,814,477*	*$87,754*	*463,349*	*$13,706*	*24,376*	*$21,472*	*505,178*	*$5,052*	*65,625*	*$52,123*	*($21,997)*
			$0.045			$0.048		$0.030		$0.881				$0.794	

Figure 40 and Table 27 show that the variance of Net Savings for the different locations is nearly $40,000.

The thermal and electric loads also changed between the locations as shown in Table 27. As expected the different locations all had annual heat recovery savings ranging from 2,126 MMBtu in Chicago to 1,485 MMBtus in Southern California. The load for New York, Long Island, Chicago, and Portland did not vary significantly.

Southern California has the highest electric rate and also shows the highest Net Savings for the CHP system. Savings are generally highest in the locations with the highest electric costs, or more specifically, the ratio of gas to electric prices.

Table 27. Comparison of Supermarket CHP Performance and Economics in Different Locations

Location	Average Electric Rate ($/kWh)	Gas Rate ($/therm)	G/E Ratio	Net Savings	Desiccant Heat Recovery (MMBtu)	Space Heating Heat Recovery (MMBtu)	Total Heat Recovery (MMBtu)
Southern California	$0.1305	$0.8711	6.7	$18,832	517.1	967.6	1,484.7
New York	$0.1288	$1.1401	8.9	$18,445	377.3	1,623.8	2,001.1
Chicago	$0.0832	$0.8065	9.7	$11,708	338.3	1,787.4	2,125.7
Long Island	$0.1110	$1.1401	10.3	$5,307	377.3	1,623.8	2,001.1
Portland	$0.0445	$0.8809	19.8	($21,997)	182.7	1,796.2	1,978.9

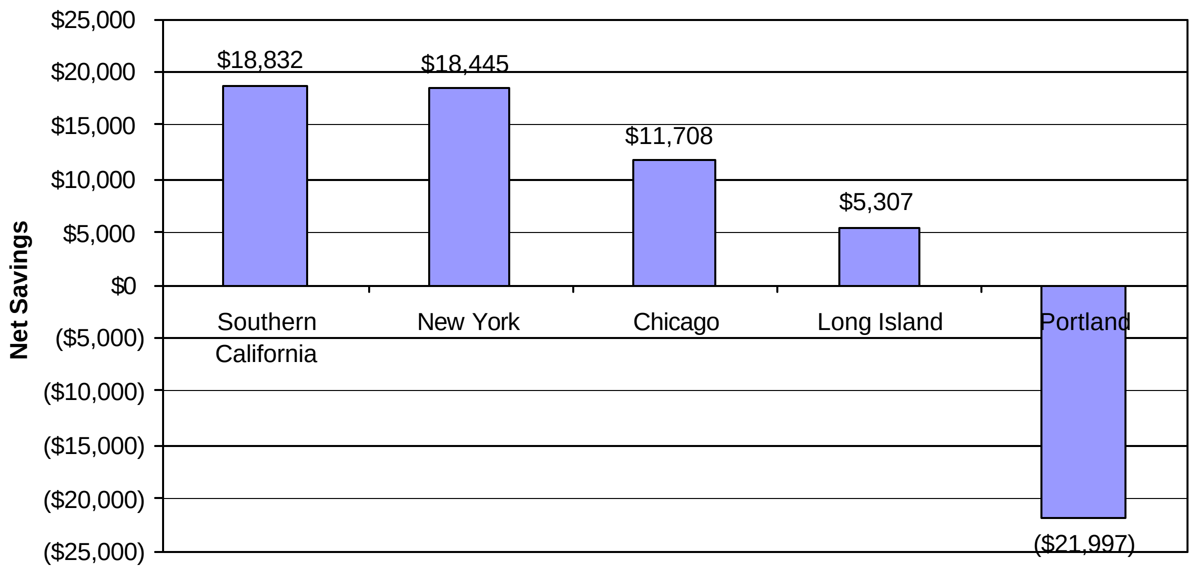


Figure 40. Net Savings for Supermarket CHP System By Location

6 LESSONS

Several lessons can be drawn from the experiences of installing and testing the CHP system at this site.

1. ***This system was successfully integrated into the store design in a cost effective manner***. The equipment was mounted on the supermarket roof near the main AHU, which allowed the plumbing to be close coupled. The microturbine was added into the store MDP without the need for additional electrical interconnection hardware (e.g, transformers, etc.). All the components and controls are similar in complexity to other refrigeration and HVAC equipment at the store. The CHP system could be easily and cost-effectively integrated the standard store design.

2. ***The cost effectiveness of the system strongly depends on gas rates***. The initial estimates when the project was conceived had assumed total gas costs of $0.75/therm (with commodity costs of about $0.50/therm). Actual gas costs for the monitoring period was $0.95/therm. The annual savings decreased by about $4,500 for each $0.10/therm increase in gas costs.

3. ***CHP systems and controls must be properly integrated into the facility to take full advantage of the available heat recovery***. In this store the controls worked well to preheat regeneration air for the desiccant wheel but did not always use all the available heat recovery for space heating. Across the monitoring period, minor changes were made to the space heating set points for the furnace section without making corresponding changes to the set points for the heat recovery coils. Care must be taken to integrate these control functions.

4. ***The capacity and efficiency of the microturbine-based CHP system varies substantially across the year***. The power output from the system drops off at higher temperatures. The measured decrease was more than predicted by the Capstone specifications. At 95°F, the power output from the turbine drops off by more than 10 kW compared to ISO conditions.

5. ***Retail applications will require CHP systems than can run unattended***. Supermarkets and other stores typically do not have onsite staff to operate a CHP system. Therefore, CHP systems for supermarkets must be as robust and reliable as HVAC and refrigeration systems. After several initial problems with the microturbine and Unifin HX, the system has run continuously for the last four months (May 2004 to September 2004). Many of the problems at the site have been related to the Unifin HX. The new Capstone 60 with an integrated HX addresses many of the reliability issues at the site will also reduce the installation costs.

6. ***CHP heat recovery integrated well with the desiccant wheel***. The modulating burner on the desiccant wheel was able to modulate to a lower firing rate and take full advantage of the available heat recovery. No modifications to the Munters unit or its controls were required.

7. ***The administrative details of interconnection with the electric utility were difficult in this retail application***. Like many retail applications, a third-party landlord owned this building and the facility included other tenants. As is fairly common in malls, the facility had multiple tenants on separate transformers, all served by the local utility on a single "non-radial" electric feed. These issues greatly complicated the technical and administrative/legal details of the electrical interconnection. Changes in the NY's Standard Interconnection Requirements (SIR) to include network or non-radial electric feeds should help to address some of these issues.