Emerging Technologies

Ammonia/CO_2 Refrigeration System Evaluation at a Food Processing Facility

ET13SCE1230

DR13.07

Prepared by:

Emerging Products
Customer Service
Southern California Edison

April 2017

SOUTHERN CALIFORNIA EDISON
An EDISON INTERNATIONAL® Company

Acknowledgements

Southern California Edison's Emerging Products (EP) group is responsible for this project. It was developed as part of Southern California Edison's Emerging Technologies Program under internal project number ET13SCE1230. This technology evaluation was conducted by Scott Mitchell of Southern California Edison and researchers at the Electric Power Research Institute (EPRI) with overall guidance and management from Paul Delaney. Contact paul.delaney@sce.com for more information on this project.

Disclaimer

EXECUTIVE SUMMARY

The HVAC&R world is undergoing significant changes, with regulatory bodies moving to reduce the use of refrigerants with high global warming potential (GWP). In commercial and industrial applications, ammonia refrigerant is a common solution which has zero GWP, but its applicability is limited because of mild flammability and toxicity concerns. In these applications R404A and R507A are common refrigerants today, but because of their high GWPs they face growing regulatory restrictions.

This report describes a field test of an alternative technology: a system using ammonia as the primary fluid and CO_2 as a pumped volatile secondary fluid. This allows the ammonia charge to be much smaller than an equivalent all-ammonia system, while distributing only CO_2 into the building. The system was installed in a food production facility in Irvine, California and is monitored along with existing, baseline conventional-refrigerant equipment, to study performance. The study documents the findings of differences in energy and power consumption as well as observations and learnings from the process of transitioning from a baseline system to a new, alternative-refrigerant approach. The system evaluated is shown in Figure 1.

FIGURE 1 PHOTOGRAPH OF NH3/CO2 REFRIGERATION SKID DURING INSTALLATION (PHOTO CREDIT: CIMCO)

The new refrigeration system was installed to provide cooling for an existing a 2,100 square-foot, -20°F drive-in freezer. The existing refrigeration equipment, part of a R507A system, was left in place but shut off. Instrumentation equipment was installed to monitor the performance of both the new system and the existing equipment, and the new equipment was disabled periodically during the study to allow collection of baseline data.

The new system used less energy in similar operating conditions than the baseline equipment, as shown in the stacked-bar chart in Figure 2. This figure shows the total daily energy consumption of each sub-component. Comparing days in April with similar weather, the baseline system used 220 kWh more per day than the new equipment, a savings of 21%. During hotter weather in summer, the savings was 16-25%.

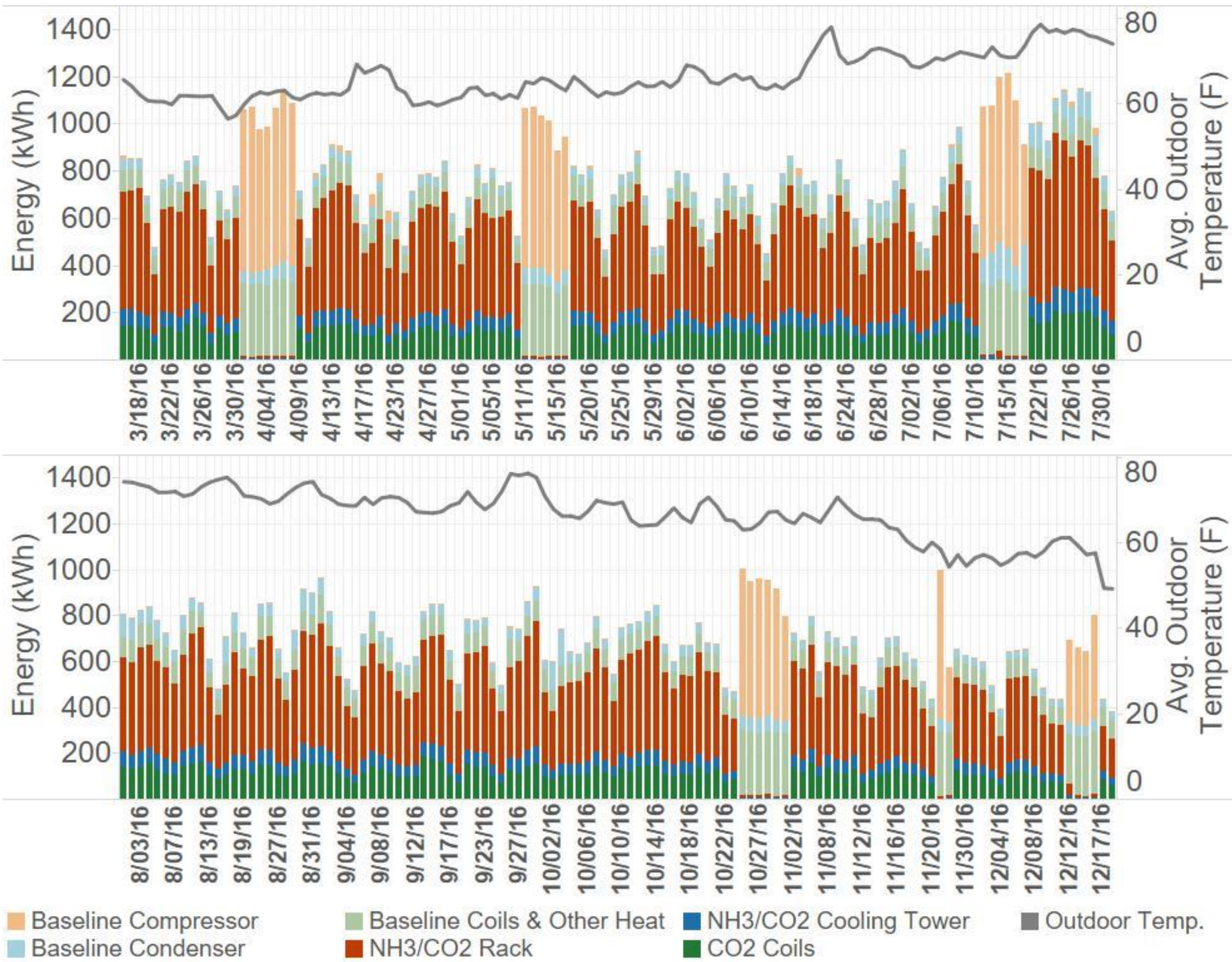

FIGURE 2 STACKED-BAR CHART OF DAILY ENERGY CONSUMPTION FOR ALL REFRIGERATION EQUIPMENT

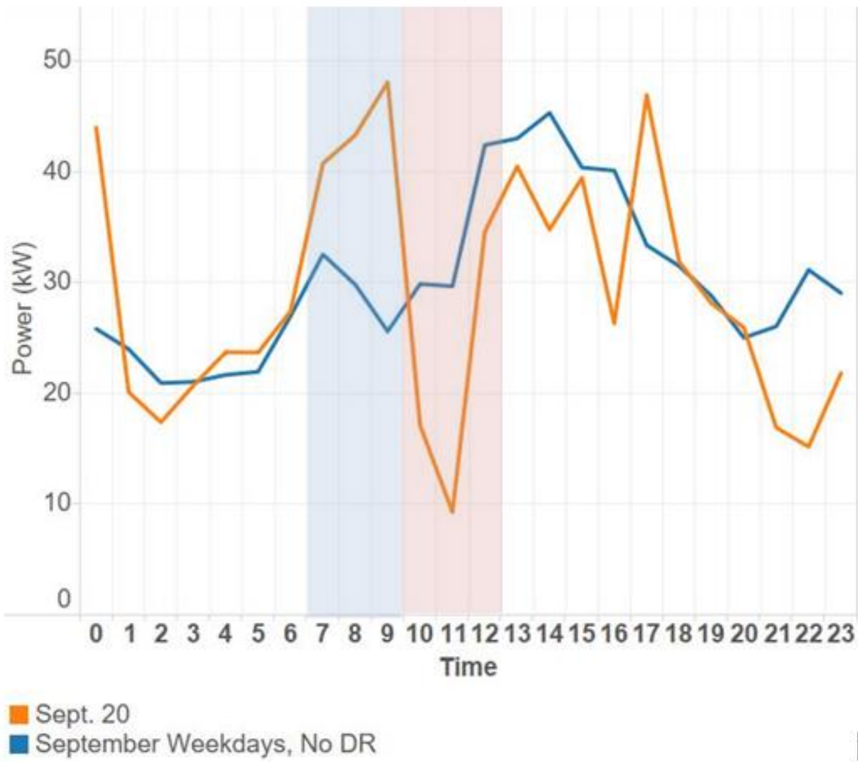

FIGURE 3 SEPTEMBER 20 DR EVENT COMPARED WITH SEPTEMBER NON-DR WEEKDAYS

The project also included simulated demand response events, including pre-cooling and load shedding. This was performed manually on-site with the help of the host facility personnel. The temperature set-points in the freezer were adjusted by about 5°F in either direction. The control was simple set-point adjustment. An example of a result comparing a pre-cool and shed event with typical baseline days is shown in Figure 3. Several different durations were evaluated. The average power during the events was 14-21 kW lower during the first hour than the same baseline hours.

The ability to pre-cool was limited by the pressure setting of the CO_2 receiver, which could not be quickly adjusted for these evaluations. Since the CO_2 liquid is held at a fixed pressure, the supply temperature in the freezer is limited by that pressure; subsequent research efforts should evaluate the effect of adjusting this pressure setting to further pre-cool.

The ammonia and carbon dioxide system evaluated here provided significant energy savings while using two refrigerants that are environmentally friendly and not subject to increasing regulatory and environmental pressures. The system remains in place and the host has, since project completion, added an additional freezer space which is only cooled using the new ammonia/carbon dioxide system, with no R507A backup.

CONTENTS

FIGURES

TABLES

INTRODUCTION

Rules around the acceptable use of refrigerants are rapidly changing in the United States and internationally, as well as locally in California. The U.S. EPA, acting under the Significant New Alternatives Policy (SNAP) program has recently changed the status of many high global warming potential (GWP) refrigerants. In the next several years, commonly-used refrigerants such as R404A, R507A, R134a and others will be prohibited for use in some types of new or retrofit commercial refrigeration installations[1]. The October, 2016 signing of the Kigali Amendment to the Montreal Protocol will further push the heating, ventilation, air conditioning and refrigeration (HVAC&R) industry to different refrigerant options as the amendment calls for the phasing out of hydrofluorocarbon (HFC) refrigerants.[2]

TABLE 1: TRAITS OF SOME COMMON REFRIGERANTS[3]

Refrigerant	Global warming potential (GWP)	Ozone depletion potential (ODP)	Critical Pressure (psia)	Critical Temperature (°F)	Normal Boiling Point (°F)	ASHRAE Safety Classification
R22	1790	0.040	723.7	205.1	-41.5	A1
R404A	3700	0	540.8	161.7	-51.2	A1
R507A	3800	0	537.4	159.1	-52.1	A1
R410A	2100	0	711.1	160.4	-60.6	A1
R407C	1774	0	671.5	186.9	-46.5	A1
R290 (propane)	~20	0	616.6	206.1	-44	A3
R744 (carbon dioxide)	1	0	1070.0	87.8	-109	A1
R717 (ammonia)	<1	0	1643.7	270.1	-28	B2L

The changing regulations have led many in the industry to search for alternatives. Some of the most common refrigerants today are shown in Table 1. Of the options shown in Table 1, the first five face regulatory challenges due to their environmental characteristics. The last three are considered alternative options for future use, but each faces some degree of challenge to growing adoption.

[1] 40 CFR Part 82 Protection of Stratospheric Ozone: Change of Listing Status for Certain Substitutes Under the Significant New Alternatives Policy Program; Final Rule, 80FR42870, July 20, 2015.

[2] United Nations Environment Programme Ozone Secretariat, The Montreal Protocol on Substances that Deplete the Ozone Layer http://ozone.unep.org/en/treaties-and-decisions/montreal-protocol-substances-deplete-ozone-layer

[3] 2013 ASHRAE Handbook – Fundamentals

Changes are also happening at a local level: California's Air Resources Board has recently approved a strategy document proposing aggressive changes which could include GWP limits as low as 150 for stationary refrigeration and 750 for stationary air conditioning[4]. The document generally outlines support for the Kigali Amendment, though the details of how that will be implemented are still to be determined at the time of this writing.

Natural refrigerants such as ammonia (R717), carbon dioxide (R744) and hydrocarbons such as propane (R290) are being used to meet the demand for very low GWP refrigeration equipment. Hydrocarbons have gained traction as the long-term solution for small-charge systems such as stand-alone refrigeration applications, where the charge level of flammable refrigerant is small and efficiency is very good. For larger industrial applications, ammonia has long been the refrigerant of choice, but due to toxicity and flammability, its use in large quantities near highly populated areas brings risks which must be accounted for, and may add cost. Carbon dioxide is gaining traction for supermarket refrigeration in the U.S., and current research and development efforts are focused on overcoming efficiency hurdles under transcritical operation, which is a particular challenge in warm climates. Solutions that can use these refrigerants while addressing the technical challenges and safety risks associated with their use could open new possibilities in significantly increasing the energy efficiency of the national refrigeration fleet, and greatly reducing the greenhouse gas impact of inevitable refrigerant release.

Ammonia is subject to restrictions on the federal, state and local levels, due to toxicity at high concentrations. In particular, the U.S. EPA has different regulations applying for site inventory thresholds of 500 pounds and 10,000 pounds, and requires emergency release notification in the event of leaks exceeding 100 pounds in a 24-hour period. Similarly, OSHA requirements apply to ammonia facilities, with additional requirements when exceeding the 10,000 pounds threshold. State level programs are common, too: most notable (and relevant to the field study discussed here) is California, where the quantity for increased scrutiny is 500 pounds. Inspections and reporting are required at regular intervals, and compliance audits must also be undertaken at regular intervals[5]. Further restrictions may be applied at the local level, particularly considering ammonia systems in highly populated areas. For these reasons, ammonia charge quantity reduction is becoming an increasingly hot topic in the industry. Most ammonia regulations were intended to deal with large charge systems. There are a number of efforts currently underway to develop regulations specifically for low charge ammonia systems that can take advantage of ammonia's high efficiency while minimizing the risk of harm due to leaks.

For some applications, a combined approach using ammonia and carbon dioxide may provide beneficial performance while limiting risk factors and performance issues. Many readers are likely familiar with the cascade cycle, a type of two-stage cooling cycle where a high stage fluid (ammonia in this case) is paired with a low-stage fluid (CO_2); heat rejection from the low stage is absorbed by the evaporating high-stage refrigerant. Cascade systems are useful particularly with

[4] California Air Resources Board, Short Lived Climate Pollutant Reduction Strategy Final Report. March, 2017 https://www.arb.ca.gov/cc/shortlived/meetings/03142017/final_slcp_report.pdf

[5] Chapp, T. 2014, Low Ammonia Charge Refrigeration Systems for Cold Storage: White Paper. International Association of Refrigerated Warehouses.

a large temperature difference and can offer good efficiency, but can be expensive and the cost and complexity is not needed for moderate cold storage and freezing temperature applications.

Another increasingly popular approach is to use CO_2 as a volatile secondary fluid, pumped to the evaporators where it partially evaporates, and condenses in a chiller. This can be achieved using ammonia as the primary working fluid. Compared with the common approach of using water/glycol as a secondary fluid, pumping power for volatile CO_2 is drastically lower, about 5% of the power required to pump water or glycol as pointed out in a 2012 ASHRAE Journal article[6].

This report describes a field study of a system using ammonia as the primary fluid and CO_2 as a pumped volatile secondary fluid. The system was installed in a food production facility in Irvine, California and is monitored along with existing, baseline conventional-refrigerant equipment, to study performance. The study documents the findings of differences in energy and power consumption as well as observations and learnings from the process of transitioning from a baseline system to a new, alternative-refrigerant approach[7].

[6] Pearson, S. 2012, Using CO2 to Reduce Refrigerant Charge. ASHRAE Journal, October 2012.

[7] Bush, J. and Mitchell, S., Reaching Near-Zero GWP with Packaged Ammonia/Carbon Dioxide Systems. ASHRAE Journal, February 2017.

BACKGROUND

For this research effort, Southern California Edison and EPRI investigated an ammonia/carbon dioxide refrigeration system for use in an industrial food application in Irvine, California. This section will detail the site and technology.

SITE OVERVIEW

The Irvine, California site was identified by Southern California Edison and Mayekawa, and was already in the process of considering a transition from their existing R-507A refrigeration system to an ammonia/carbon dioxide solution. The host is a Japanese-owned company that makes food products. The facility includes food processing, freezing, and storage.

FIGURE 4: PHOTO OF THE FREEZER UNDER INVESTIGATION

The food products are produced, then stored in a 2,100 square-foot, -20°F drive-in freezer. The refrigeration load, approximately 12.2 tons of refrigeration (TR), was met with a single R507A reciprocating compressor with a three-step mechanical unloader, installed in 2010. There are three additional R507A compressors for various other refrigerated spaces, all sharing an evaporative condenser. The production line typically operates 12 hour shifts 5 days per week, with a morning shift on Saturdays depending on workload. The freezer, shown in Figure 4, is used for short term product storage before shipping. The facility is situated in a busy, mixed-use area, in close proximity to retail and residential areas. This facility was chosen because of the ability to keep the existing equipment running for baseline comparisons throughout the test period.

TECHNOLOGY OVERVIEW

The system evaluated in this effort is a new packaged ammonia (NH_3)/carbon dioxide (CO_2) system manufactured by Mayekawa. The system, given the model name Newton 3000, uses ammonia as the primary stage and pumped, volatile CO_2 as the secondary stage. It will be referred to as the NH_3/CO_2 system, or similar, throughout this reporting. A simplified schematic of the system configuration is shown in Figure 5.

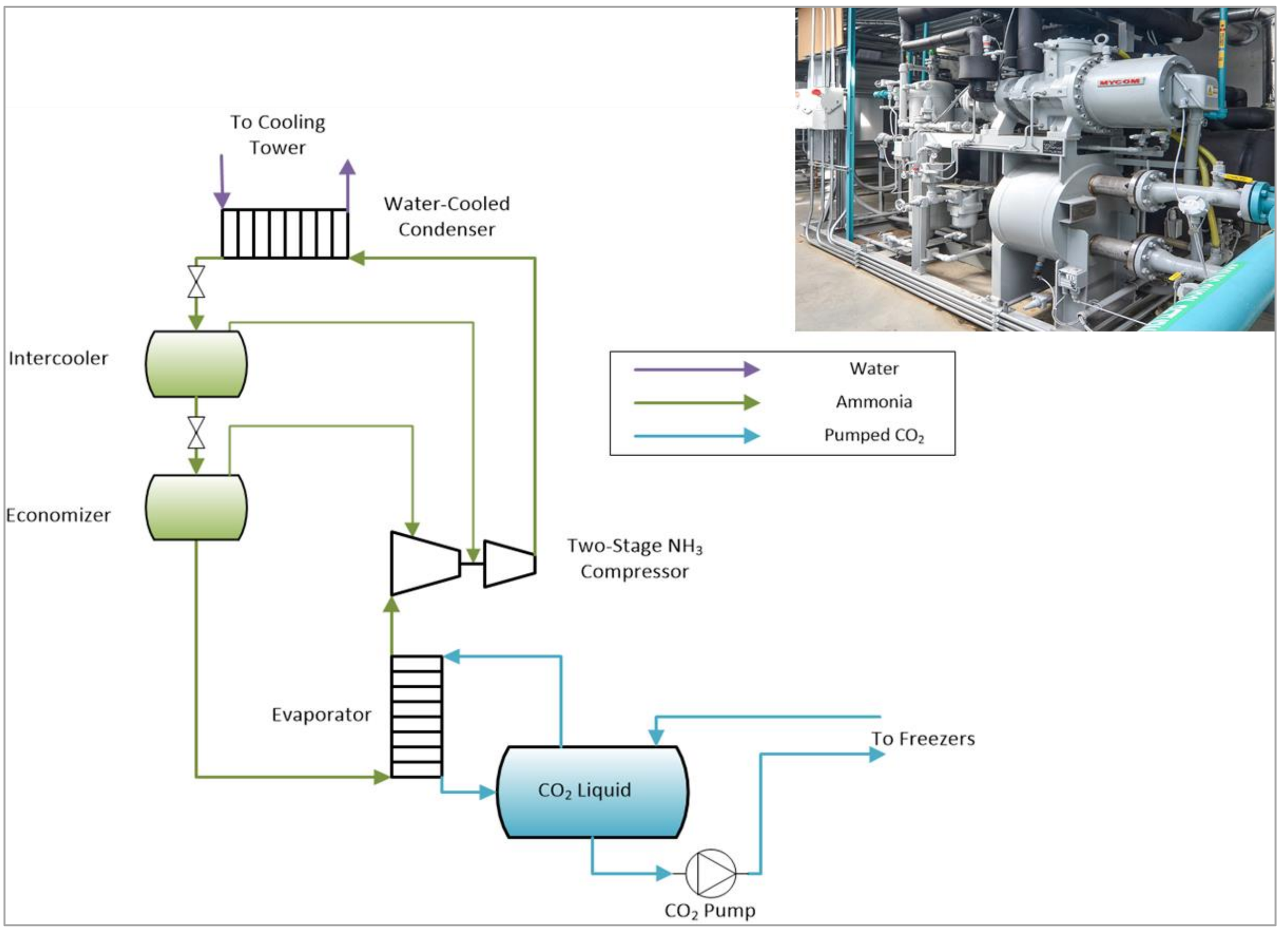

FIGURE 5 HIGH-LEVEL SCHEMATIC OF THE NH3/CO2 REFRIGERATION SYSTEM UNDER EVALUATION

The main ammonia circuit has a two-stage, screw compressor. The discharge of the screw compressor goes to a water-cooled condenser. From there, the refrigerant goes to an intercooler which separates liquid from vapor. The vapor goes to the compressor, mixing and cooling inlet refrigerant to the second stage of compression. The liquid goes to an economizer. The economizer works similarly, as vapor refrigerant goes to an economizer port in the first stage of compression, and liquid goes to the evaporator. The evaporator is an ammonia-to-carbon dioxide heat exchanger, which cools carbon dioxide. The CO_2 is in a liquid tank, from which sub-cooled liquid CO_2 is pumped to the process evaporators.

This configuration has a few advantages. First, since the system is skid-mounted, on-site install time can be significantly less than a site-built system. The centralized ammonia loop is relatively compact and has a low charge of ammonia. Liquid CO_2 is pumped to the evaporators and undergoes sensible heating and partial phase change. It is cooled by the ammonia system, which essentially acts as a chiller. This contains the charge of hazardous refrigerant to the central location. There are other advantages to this pre-engineered skit-mount approach. The intercooler and economizer stages improve compressor performance and send low-vapor quality refrigerant to the evaporator enhancing system efficiency. The integrated package allows the system controls to integrate each of these components and coordinate compressor speed and valve timing effectively and with little or no on-site tuning required.

NEW EQUIPMENT AND INSTALLATION

The Newton package is made in several configurations, having different compressor sizing and configurations for larger capacity, or specialized configurations for industrial freezing or ice rinks. The device tested in this effort was the Newton R-3000, for which the manufacturer provides the following specifications:

- CO_2 Supply Temperature: -25.6°F
- Cooling Capacity (with cooling water at 89.6°F): 26.9 TR
- Motor Power: 45 kW
- Ammonia Charge: 55.1 lbs.
- Power Source: AC 400/440V @ 50/60Hz for motor; AC 200/220V @ 50/60 Hz for controls
- Compressor: Semi-hermetic, compound screw, VFD driven with IPM motor
- Outer Dimensions: L 9ft 2in; W 6ft 5in; H 7ft 11in (excludes cooling tower)

The manufacturer also uses an interior permanent magnet (IPM) synchronous motor for the compressor; these motors have higher efficiency than conventional induction motors, and maintain high efficiency even at low compressor speed.

The installation was performed by CIMCO Refrigeration and included:

- Mayekawa Newton R-3000 NH_3/CO_2 package
- Closed-circuit cooling tower with water treatment
- Two CO2 evaporator fan coils with electric defrost
- Pump for circulating condenser water
- Installation of EPRI data monitoring equipment

The installation process took place over approximately four weeks in February and March, 2016. The total time on-site was reduced compared with a conventional installation because of the skid-mounted system. The installation was not without surprises, however. Satisfying the city permitting and inspection requirements led to several unanticipated additions to the site plan. A higher surrounding facade (for aesthetic purposes only) and additional ammonia leak containment measures (the addition of an ammonia diffusion tank for the vent discharge) were required. The surrounding facade is shown in Figure 6.

FIGURE 6 PHOTO SHOWING NEW EXTERIOR FAÇADE SURROUNDING REFRIGERATION SYSTEM (PHOTO CREDIT: CIMCO)

FIGURE 7 PHOTOGRAPH OF NH3/CO2 REFRIGERATION SKID DURING INSTALLATION (PHOTO CREDIT: CIMCO)

The Newton package is shown in Figure 7. The compressor (above) and condenser (below) are visible in this image, with the condenser connected to water/glycol lines to and from the cooling tower. This photograph was taken from adjacent to the cooling tower, which is shown in Figure 8.

FIGURE 8 PHOTO OF THE NEW COOLING TOWER FOR THE NH3/CO2 REFRIGERATION SYSTEM (PHOTO CREDIT: CIMCO)

FIGURE 9 NEW CARBON DIOXIDE EVAPORATOR COIL BEING INSTALLED IN THE FREEZER

Two new CO_2 evaporators were installed in the freezer. Figure 9 shows one of the coils during the installation process. Similar, R507A evaporators remained installed in the freezer during testing.

The baseline compressor, in the mezzanine compressor room, is shown in Figure 10. The compressor room features four R507A compressors for the various refrigeration end loads; an evaporative condenser for the R507A systems is outside the frame of this photo.

FIGURE 10 EXISTING COMPRESSOR IN MEZZANINE COMPRESSOR ROOM

An overall schematic of the installation is shown in Figure 11.

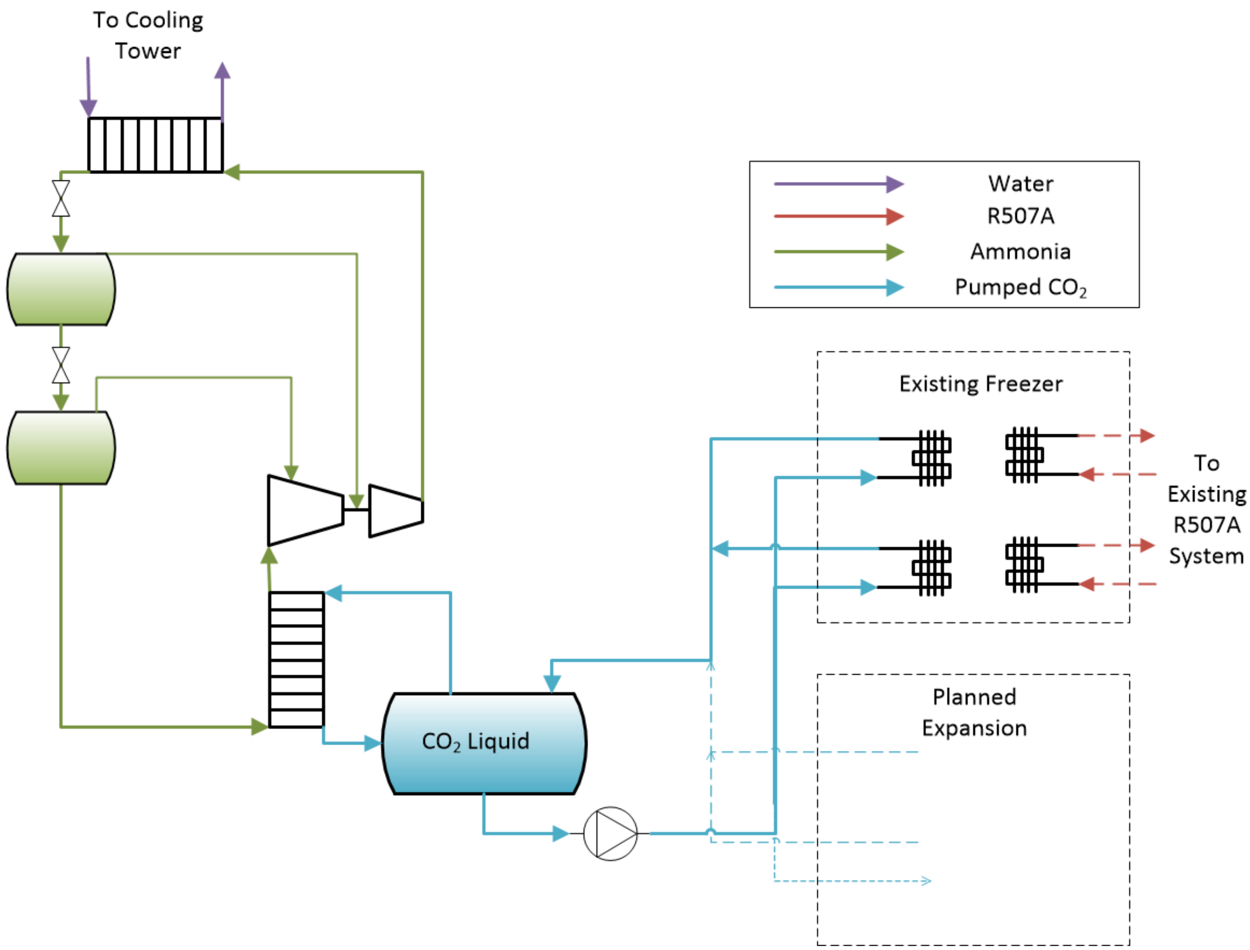

FIGURE 11 SIMPLIFIED SCHEMATIC OF (EVAPORATOR & CO2 TANK SHOWN AS ONE)

TESTING OVERVIEW

OBJECTIVES

The objectives of this work can be summarized as follows:

- Measure the performance of an NH_3/CO_2 refrigeration system in a real-world application. Considering factors such as weather and schedule, analyze system performance for new learnings and to enhance understanding of the system characteristics.
- Evaluate performance for the Southern California region. The Irvine, CA site provides only a relatively narrow band of weather compared with other cities in SCE's territory, but expected performance in other cities can be estimated based on the data collected in Irvine.
- Compare performance against existing, baseline equipment. Using the equipment already installed on-site as baseline, the energy and demand reduction from the new system can be determined. Since the system has excess capacity for the current loading, estimates can also be extrapolated to a possible expansion of the refrigerated space.
- Test the equipment in demand response operation. With the support of on-site personnel, the system was tested for "pre-cool" and "load shed" operation over several days of testing. The goal of this effort was to assess the current or near-term capability of this equipment to perform demand response in a freezer or warehouse type application.

APPROACH

The approach for this evaluation is field-monitoring of equipment under normal operation, with periodic baseline testing which is performed by disabling the system under test, and turning on the baseline equipment. This was done several times during the test period, in an effort to get data across a range of weather conditions while respecting the scheduling/availability of on-site personnel to make and monitor the changes. Demand response tests were performed similarly, with several days of testing during which on-site personnel executed changes to the equipment set-points to simulate a demand response event.

TEST OVERVIEW

The testing described above was performed over the course of 2016 starting after the installation was complete. To generally sort data, the days are filtered into:

- Baseline - the R507A system is the only one cooling the space
- New Equipment - the NH_3/CO_2 system is the only one cooling the space

- Transition - the day is split, usually because of a mid-day switch between baseline and new equipment, but also possibly including some limited maintenance which was not identified

- Other - days where on-site maintenance was taking place, demand response testing was performed, or other known aberrations from the test schedule

The days tagged as "other" are identified by communication with the host site manager. In initial data processing, each day is flagged as "baseline" if the baseline compressor uses >100 kWh and the Newton uses <75 kWh. A day is flagged as "New Equipment" if the Newton uses >100 kWh and the baseline compressor uses <75 kWh. "Transition" and "Other" days are removed from the bulk data analysis.

Baseline data was collected on the following dates:

- April 1 - 7
- May 11 - 16
- July 12 - 19
- October 25 - 30
- November 22 - 27
- December 12 - 15

The following dates were filtered as "other" based on a known maintenance, interruption, inspection, demand response test, or other known interruption:

- May 24
- June 21, 23
- July 17, 18, 20
- August 10, 11, 17, 18, 21, 22, 23, 24
- September 13, 20, 21, 30
- November 1, 6, 7, 23, 24, 25, 26

Also, one other change took place during the testing which will affect the results: the suction pressure of the R507A system (which includes the compressor monitored here, and others) was reduced to try to improve an ice cream making process elsewhere in the plant. This change, which took place around July 20 (just after the July baseline data collection) does not affect the new equipment, but does change the performance with the R507A compressor running. As will be discussed in the results section, the main difference was a lower average temperature in the freezer when the R507A system was running after this change.

INSTRUMENTATION

To capture performance of the refrigeration systems, EPRI designed and built an instrumentation system for installation in the field. The monitoring points and equipment installed are detailed in Table 3.

TABLE 2 INSTRUMENTATION DESCRIPTION

Reading	Description	Location(s)	Accuracy	Device
Power	Multiple, see below	Six circuit breaker-level readings	+/- 0.2% power	Elkor WattsOn Power Meter
Temperature	Refrigerant Temperature	CO_2 circulating supply and return	+/- 0.9 °F	Type-T Thermocouple w/ Well
Temperature	Air Temperature	In freezer; cooling tower air inlet and outlet	+/- 0.4 °F	10K-3 Thermistor
Temperature	Water Temperature	Cooling tower inlet, outlet	+/- 0.4 °F	10K-3 Thermistor
Air Temperature/Humidity	Air Temperature /Humidity	In Freezer	+/- 0.9F @ 72F; +/- 2% RH	Dwyer RHT-R016
Air Temperature/Humidity	Air Temperature /Humidity	Outdoor	+/- 0.9F @ 72F; +/- 2% RH	Dwyer RHP-2R11
Refrigerant Flow	CO_2 flow rate	CO_2 circulating supply line	liquid +/- 0.10% reading; gas +/- 0.25% reading	Micro Motion Coriolis Flow Meter CMFS050M
Make-up Water Flow	Make-up Water Flow	Cooling tower make-up water	+/- 1.5% of reading	Seametrics MJNR-075-20P

The power meters were installed at the circuit breaker level for a total of six readings. They are:

- NH_3/CO_2 system
- NH_3/CO_2 system cooling tower and pump
- CO_2 evaporator coils including defrost heat
- Baseline compressor
- Baseline evaporative condenser
- Baseline evaporator coils, including defrost heat, under-floor heat, and other

There is some overlap in what power must be considered to compare systems, such as the heaters on the baseline evaporator coil circuit. So, in comparing total power or energy consumption, all readings are included.

In addition, the following data points were logged from the output of the Newton system's control board:

- Suction pressure
- Discharge pressure
- Intermediate Pressure
- Economizer Pressure

- Oil Pressure
- CO_2 Receiver Pressure
- Suction Temperature
- Discharge Temperature
- Oil Temperature
- Cooling Inlet Temperature
- Cooling Outlet Temperature
- Compressor Motor Current
- Compressor RPM
- CO_2 Differential Pressure
- CO_2 Pump Motor Current
- CO_2 Saturated Temperature

Since these values are from the manufacturer's equipment, they are considered as secondary measures and considered to augment the primary data stream.

RESULTS

Data acquisition began on March 14, 2016.

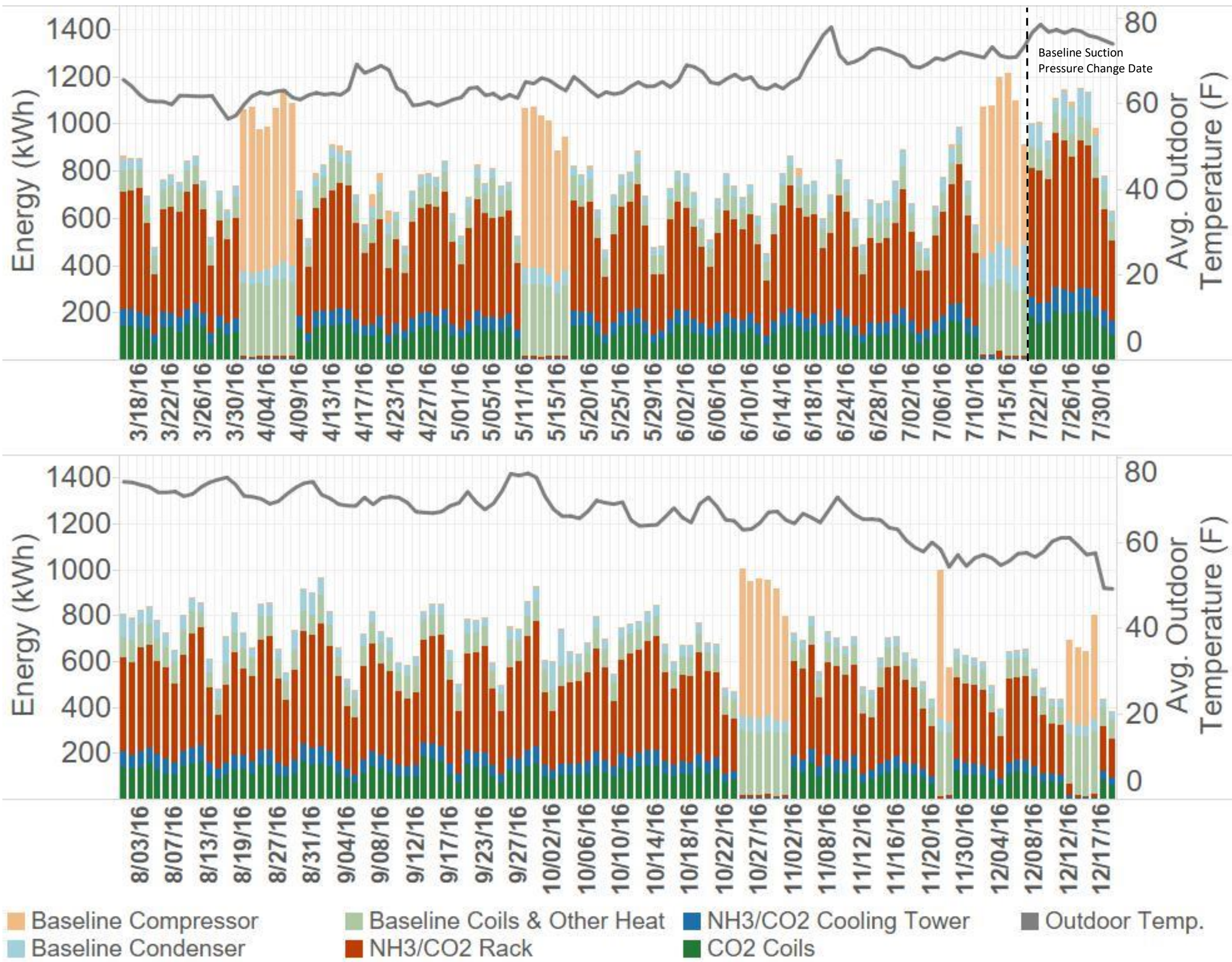

FIGURE 12 STACKED-BAR CHART OF DAILY ENERGY CONSUMPTION FOR ALL REFRIGERATION EQUIPMENT

From a high-level perspective Figure 12 shows the energy consumption of all equipment, for each baseline or "new equipment" day, with the "other" and "transition" days removed. The graph is a stacked bar chart, so the highest number is the total for all equipment. The baseline compressor is shown in orange, the baseline condenser in light blue and the baseline coils (as well as underfloor heat) in light green. The NH_3/CO_2 system is shown in red, the new cooling tower in dark blue and the CO2 evaporator coils in dark green. The outdoor temperature, averaged over each day, is also shown in gray. The results show that, while there is some overlap, for a given period of similar weather the energy consumption is considerably lower using the new NH_3/CO_2 system. This graph also shows that, during the new equipment operating days, some of the baseline equipment (the heaters captured by the baseline coil power meter, and the baseline condenser) still run. The baseline condenser energy typically is about the same to somewhat higher with the baseline compressor running. The measured "baseline coil and other heat" energy is much higher with the baseline compressor running, as would be expected since on "new equipment" days the coils are off and only the auxiliary heaters are running. All of the new equipment goes to very near zero energy on baseline days; the compressor does run briefly a few times per day to maintain the pressure level of the CO_2 receiver.

The data is visualized differently in Figure 13, which shows the total energy plotted against daily average outdoor temperature for baseline and new equipment days. Also, on this graph Sundays are shown with a hollow circle and all other days are filled. This graph shows an upward trend of energy with outdoor temperature for both cases, as might be expected. The baseline equipment generally used more energy across the whole temperature range. The energy consumption on Sundays is shown separately because it is considerably lower than other days: the plant is usually closed on Sundays.

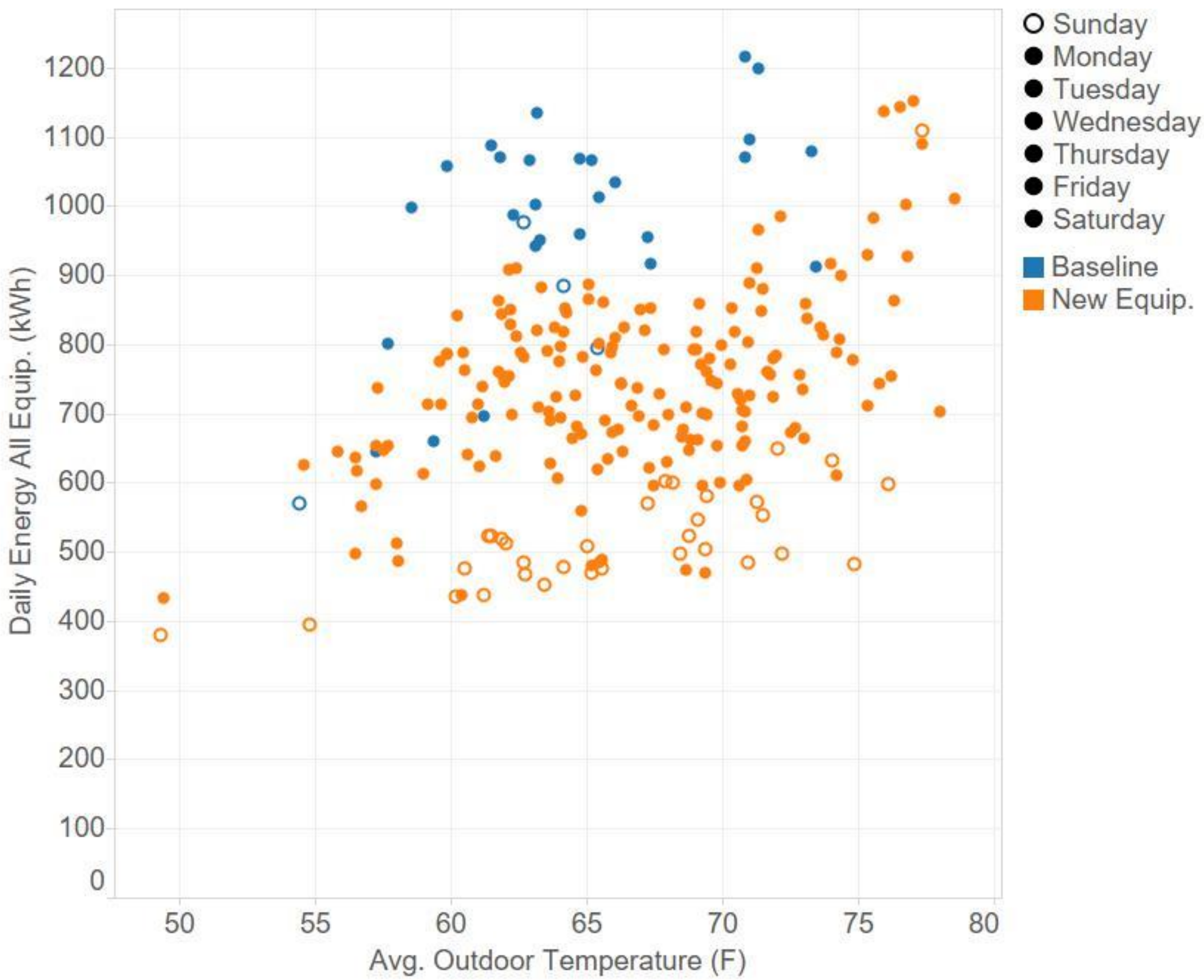

FIGURE 13 DAILY ENERGY CONSUMPTION VS. AVERAGE OUTDOOR TEMPERATURE FOR BASELINE AND NEW EQUIPMENT

The following tables show examples of baseline and new equipment daily summary data for spring and summer. Each shows the daily energy for each sub-meter, the maximum power recorded for each sub-meter (in one-minute intervals), and the average, maximum, and minimum temperature for the day.

First Table 3 shows late March and early April data. The same weekdays were selected for comparison, and time before and after the baseline period is shown. As shown on the table, there is some overlap of each system during the operation of the other; for instance, on April 13, the baseline coils and other heaters had higher energy draw than typical; close inspection shows that a defrost heater engaged at some point, even though the compressor was not running at that time. Similarly, for each system there is occasional brief cycling on of the compressor during "off" times. Comparing April 4-6 and April 11-13, where average temperatures were similar, the new equipment (on April 11-13) used 21% less energy, or 220 kWh per day less than the baseline.

TABLE 3 BASELINE AND NEW EQUIPMENT DAILY SUMMARY DATA FOR SPRING

		3/27	3/28	3/29	3/30	4/3	4/4	4/5	4/6	4/10	4/11	4/12	4/13
		Sun	Mon	Tue	Wed	Sun	Mon	Tue	Wed	Sun	Mon	Tue	Wed
Baseline Compressor Energy	kWh	0	0	0	5	606	607	666	724	0	11	7	11
Baseline Condenser Energy	kWh	36	39	39	41	47	66	63	67	36	45	47	46
Baseline Coils & Other Heat Energy	kWh	81	86	87	89	305	301	321	331	81	90	92	138
NH3/CO2 Rack Energy	kWh	289	402	356	427	15	10	13	11	282	442	474	507
NH3/CO2 Cooling Tower Energy	kWh	40	55	49	58	3	3	3	3	41	61	65	65
CO2 Coils Energy	kWh	71	132	105	116	0	0	0	0	73	139	143	144
Energy Total	kWh	518	713	637	736	976	986	1066	1136	512	788	828	911
Baseline Condenser Max. Power	kW	2	10	12	12	10	13	13	14	5	13	14	14
Baseline Coils & Other Heat Max Power	kW	4	5	5	10	36	36	36	36	3	11	11	60
Baseline Compressor Max. Power	kW	0	0	0	38	38	38	37	39	0	38	30	31
NH3/CO2 Cooling Tower Max. Power	kW	6	5	5	5	4	4	4	4	5	6	6	6
CO2 Coils Max. Power	kW	44	44	45	45	0	0	0	0	44	45	44	45
NH3/CO2 Rack Max. Power	kW	47	49	50	48	42	41	41	41	48	48	49	50
Average Outdoor Temp.	F	62	59	57	57	63	62	63	63	62	63	62	62
Max. Outdoor Temp.	F	71	66	67	66	73	76	78	74	73	76	73	74
Minimum Outdoor Temp.	F	57	52	50	51	54	53	53	56	58	55	55	57

Table 4 shows the same data for days in July. Again, data from before and after the baseline period is shown. In this case, the weather prior to the baseline period was similar to the baseline period, and the weather after was considerably hotter. Comparing the days with similar weather, the energy consumption was 25% lower, or 287 kWh per day, than the baseline days. Comparing the hotter period of July 21-23 to the baseline period, the energy consumption was still 16% lower or 191 kWh per day. The energy consumption was on average 96 kWh per day higher for the hot days than the mild days with the NH_3/CO_2 system running.

TABLE 4 BASELINE AND NEW EQUIPMENT DAILY SUMMARY DATA FOR SUMMER

		7/7	7/8	7/9	7/14	7/15	7/16	7/21	7/22	7/23
		Thu	Fri	Sat	Thu	Fri	Sat	Thu	Fri	Sat
Baseline Compressor Energy	kWh	15	6	0	700	742	707	6	12	0
Baseline Condenser Energy	kWh	64	66	59	159	149	96	97	105	79
Baseline Coils & Other Heat Energy	kWh	89	87	86	305	308	280	88	91	86
NH3/CO2 Rack Energy	kWh	508	586	434	30	14	11	549	562	519
NH3/CO2 Cooling Tower Energy	kWh	71	82	63	6	3	3	85	84	83
CO2 Coils Energy	kWh	164	158	114	0	0	0	178	156	159
Energy Total	kWh	910	985	756	1200	1216	1097	1002	1010	928
Baseline Condenser Max. Power	kW	13	13	13	13	13	13	13	13	13
Baseline Coils & Other Heat Max Power	kW	10	10	5	36	36	36	10	11	5
Baseline Compressor Max. Power	kW	38	37	0	39	39	38	35	37	0
NH3/CO2 Cooling Tower Max. Power	kW	8	7	7	7	7	7	7	7	7
CO2 Coils Max. Power	kW	44	44	44	0	0	0	45	45	45
NH3/CO2 Rack Max. Power	kW	49	49	50	47	42	43	49	48	48
Average Outdoor Temp.	F	71	72	72	71	71	71	77	79	77
Max. Outdoor Temp.	F	81	82	82	79	81	81	92	94	88
Minimum Outdoor Temp.	F	65	66	66	67	64	66	67	66	68

A significant difference in the energy consumption is revealed by examining a 24-hour average load profile for each case, baseline and new equipment. Figure 14 shows the hourly average power and temperature profile, for data recorded in the month of July (excluding Sundays). The baseline data is shown in red, the new equipment in blue. The baseline days had slightly lower temperature on average. The average baseline power ranged from about 37 kW overnight (and again in the 10 to 11 AM window) to a high of 56 kW from 12 to 1 PM. For the new equipment, the power ranges from a low of about 24 kW overnight to a high of 48 kW in the 1 to 2 PM window. The most notable difference is that the baseline equipment has a relatively flat load profile in comparison. The new equipment power overnight is half of the peak, compared with about 66% for the new equipment.

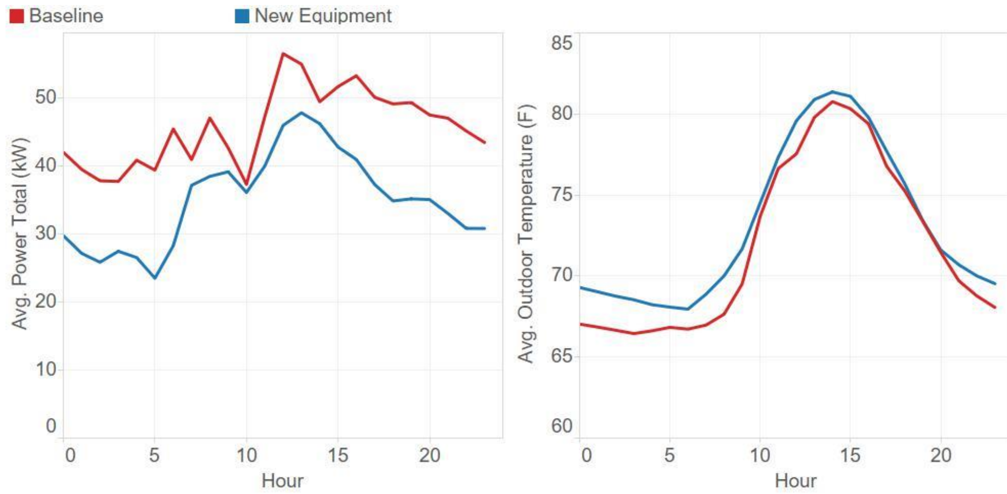

FIGURE 14 LOAD SHAPE OF EQUIPMENT WITH AVERAGE OUTDOOR TEMPERATURE PROFILE (JULY, SUNDAYS REMOVED)

Figure 15 shows the load shapes for the new equipment for the entire monitoring period. This shows the difference in energy consumption of each day, which reflects the plant's production schedule. The schedule has two shifts each weekday, and a morning shift only on Saturdays as needed, with no shifts on Sundays. This is reflected clearly in the load profiles, which show Saturday having a similar morning usage to weekdays, with low afternoon usage, and Sunday having a low and flat load profile.

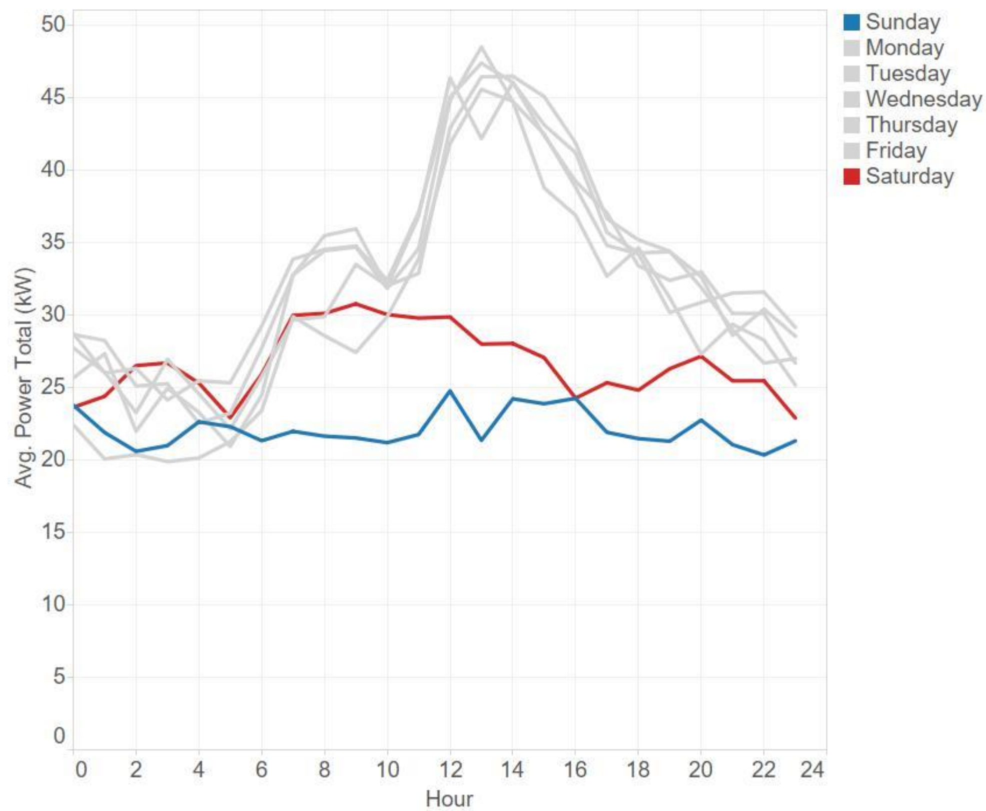

FIGURE 15 LOAD SHAPES BY WEEKDAY FOR FULL TIME PERIOD, WITH NEW EQUIPMENT

Figure 16 shows a single typical summer weekday (Wednesday, July 6) of operation with the new equipment. The outdoor temperature ranged from 65°F to 80°F. The main compressor rack power is shown in blue, the cooling tower in orange, and the CO_2 evaporators and defrost heaters in green. For visual clarity, the power of the other equipment is not shown here. Overnight, the power can be seen turning on and off, as the system is cycling at minimum compressor speed. During these cycles, the compressor turns on, and power is 36 kW, before decreasing to about 29 kW. Right before shutting off, the power briefly increases to about 37 kW. The cooling tower power also turns off when the main compressor rack does. The cooling coil turns off, but briefly cycles on and off during “off” periods to circulate air. Shortly before 10:00 AM the first defrost occurs: a total of about 43 kW of power between the fan coil power and resistance heat power across the two fan coils. In response, the compressor power after the defrost is briefly higher, reaching 48 kW and gradually ramping down over the course of about 40 minutes. Later, in the early afternoon, there is a sustained period of operation where the system appears to run above minimum power for approximately 3 hours. This is followed by another defrost and again a period of higher power consumption. In the evening, the cyclic operation resumes.

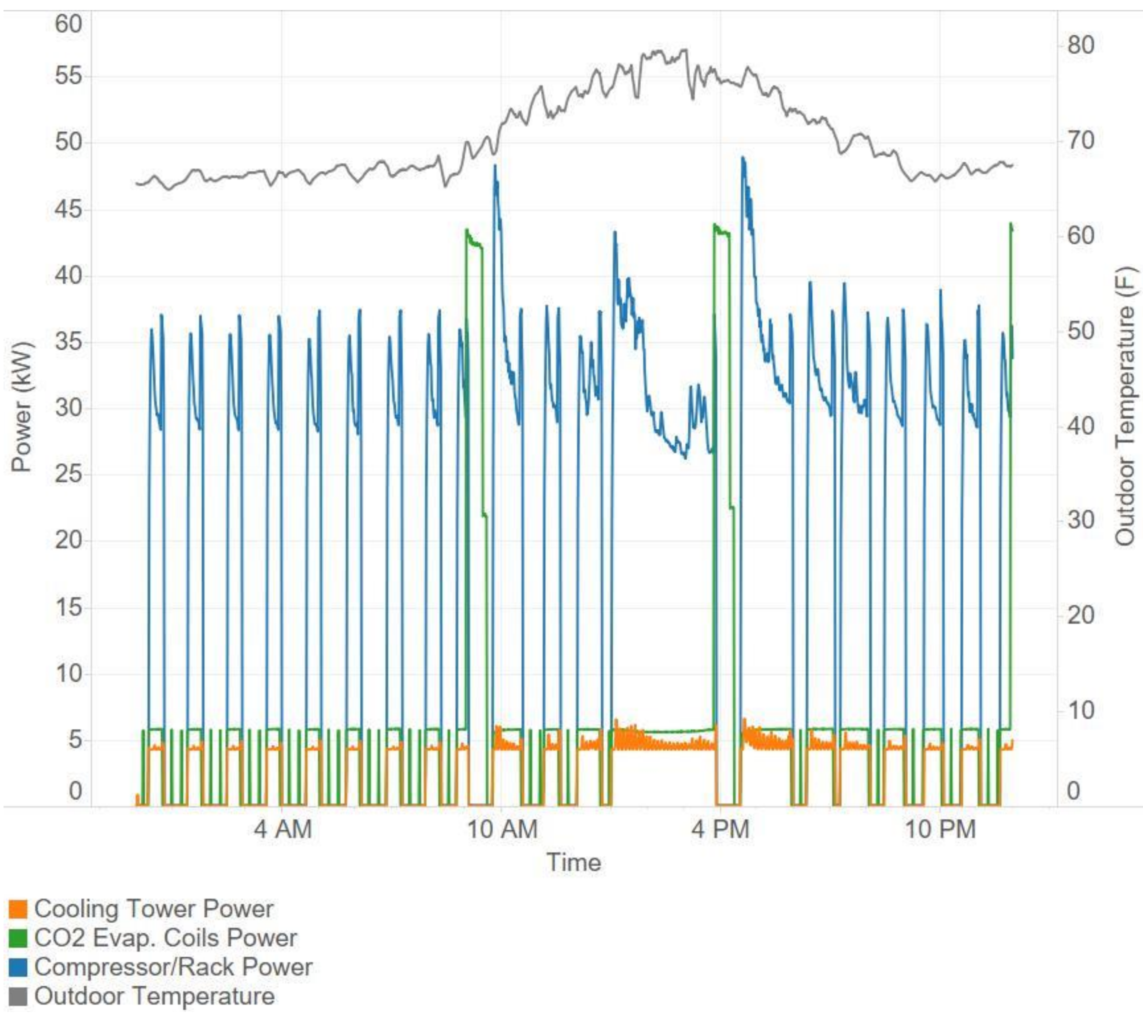

FIGURE 16 POWER OF NH_3/CO_2 EQUIPMENT AND OUTDOOR TEMPERATURE FOR JULY 6, 2016

The compressor RPM for the same time period, as recorded from the Newton system's PLC is shown in Figure 17. This provides some insight into the power profile seen above. The power profile closely follows the compressor RPM; the maximum speed, 4500 RPM, is reached twice (both times, as a rebound after defrost); and the compressor always increases to 4000 RPM briefly prior to shut-down.

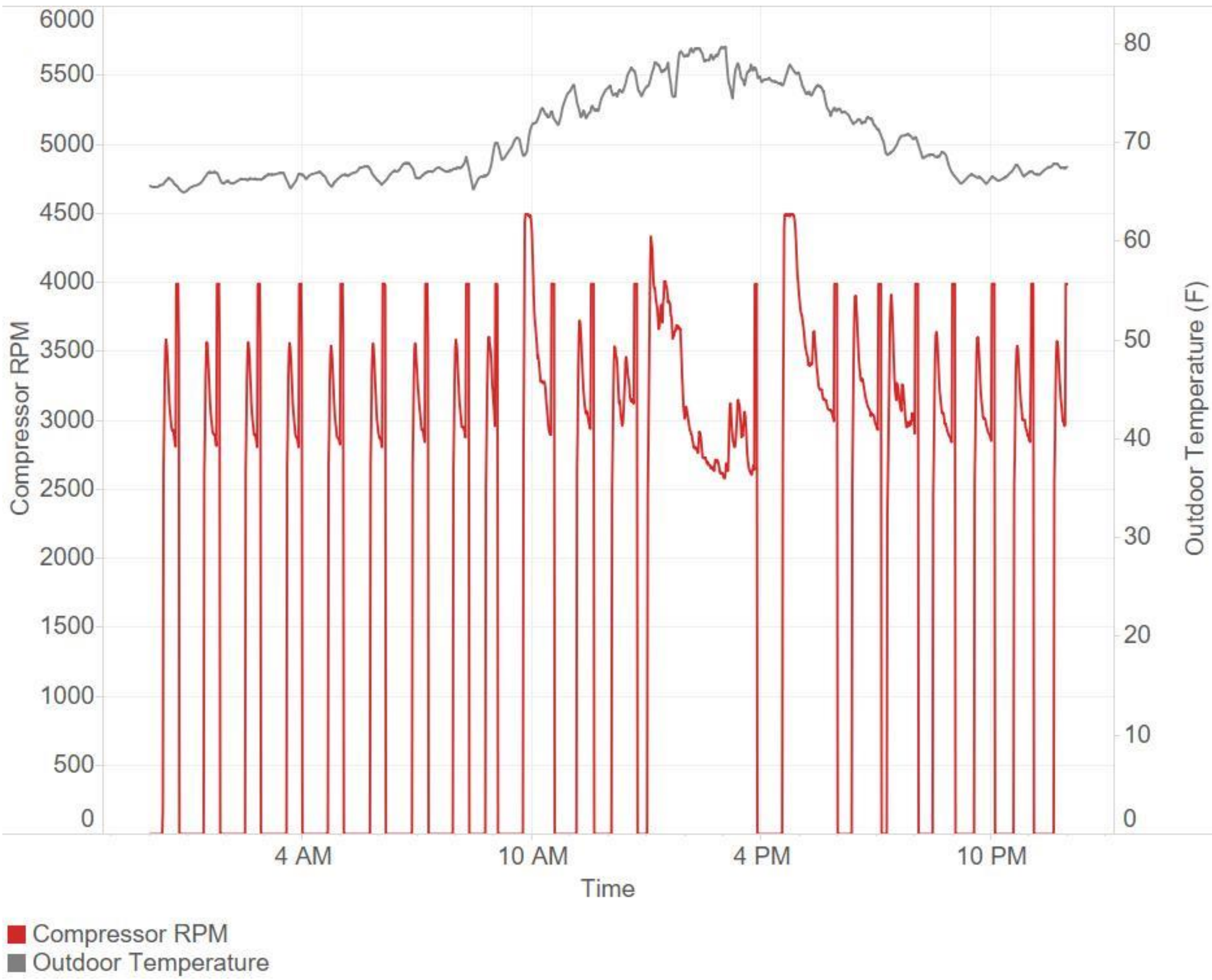

FIGURE 17 COMPRESSOR RPM AND OUTDOOR TEMPERATURE FOR JULY 6, 2016

Figure 18 shows the temperature measured inside the freezer during the period shown above. During defrost, the air temperature increases to approximately 25°F. For the first defrost, the duration from when the temperature begins increasing to when it reaches a low point again (shortly after 10 AM) is one hour, twenty-one minutes. The amount of time it is above 0°F is about 32 minutes. During normal cycling, the temperature range is -15°F to -5°F. The cycling occurs because of the oversized cooling capacity; when the load is higher (between about 1 PM and 4 PM), the temperature is in a tighter range. The time required to pull down the temperature by 10°F during an overnight cycle is about 20 minutes.

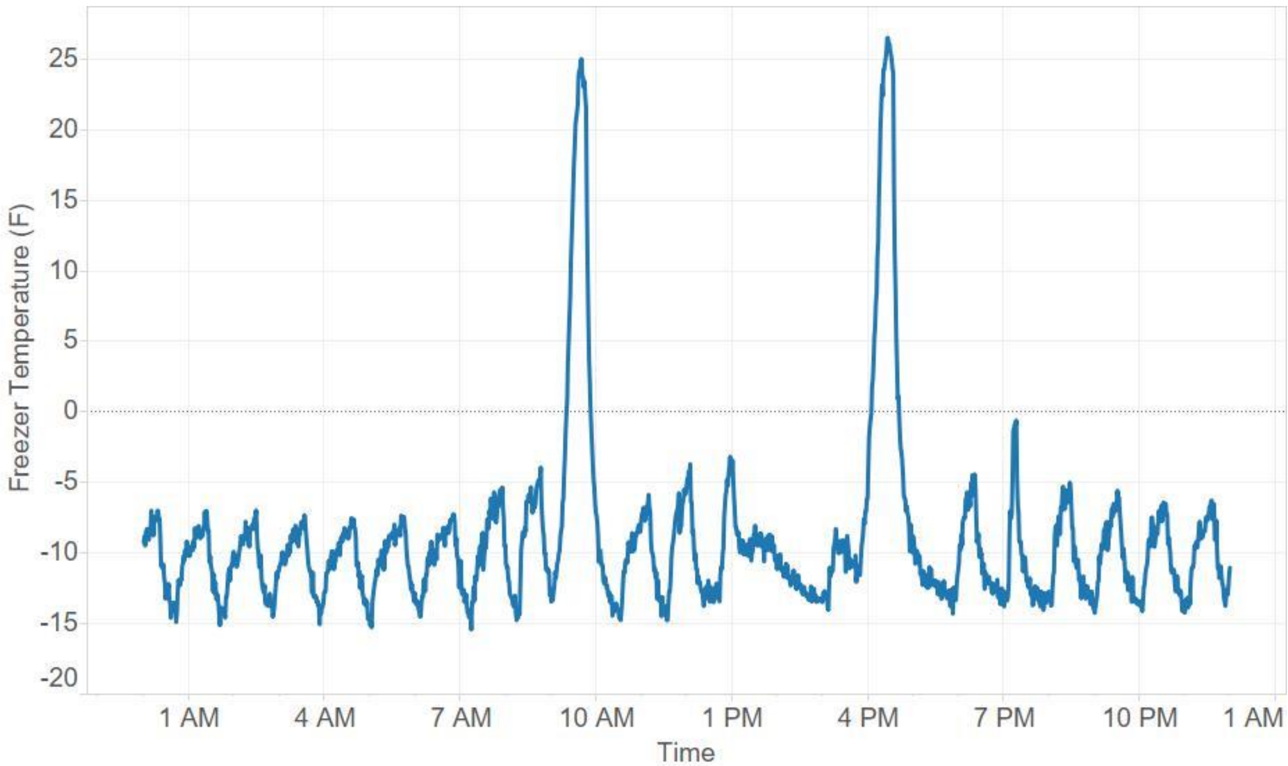

Figure 18 Freezer Air Temperature for July 6, 2016

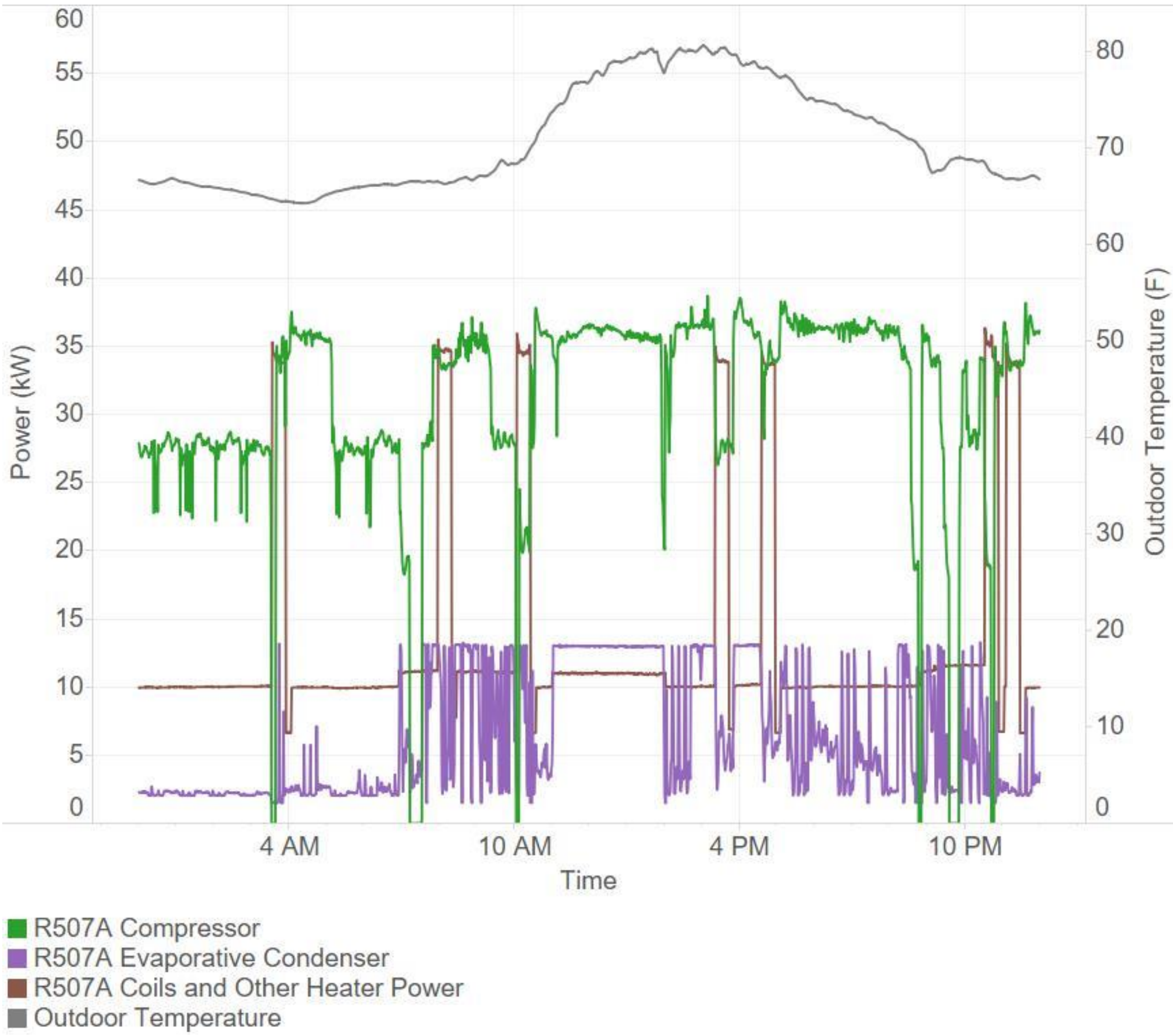

FIGURE 19 BASELINE R507A EQUIPMENT POWER AND OUTDOOR TEMPERATURE FOR JULY 15, 2016

Figure 19 shows a day of baseline operation with a similar outdoor temperature profile. The compressor power is shown in green, the condenser in purple, and the fan coils, defrost heaters, and under-floor heat system in brown. There are a few items to note: first, the defrost happens more frequently and at lower power; a suspected cause is that the defrost for each coil is individual, and therefore out of sync. The compressor power operates between three stages, with power levels of approximately 22 kW, 27 kW, and 36 kW. Since the compressor shares suction and discharge manifolds with other compressors, there is not much variation from those levels with temperature. The system does stay at a higher stage during the hotter part of the day, and a lower stage (even cycling off some in the evening) during evening and early morning hours.

The temperature in the freezer, shown in Figure 20, behaves differently with the baseline equipment. Since the compressor size is well-matched to the load, the temperature oscillates much less than with the new equipment. The more-frequent but lower-power defrosts mean the temperature increases more often, but less severely. The exception is around 4 PM, where both defrost heaters ran in close succession. The air temperature reached approximately 29°F and was over 0°F for approximately 39 minutes.

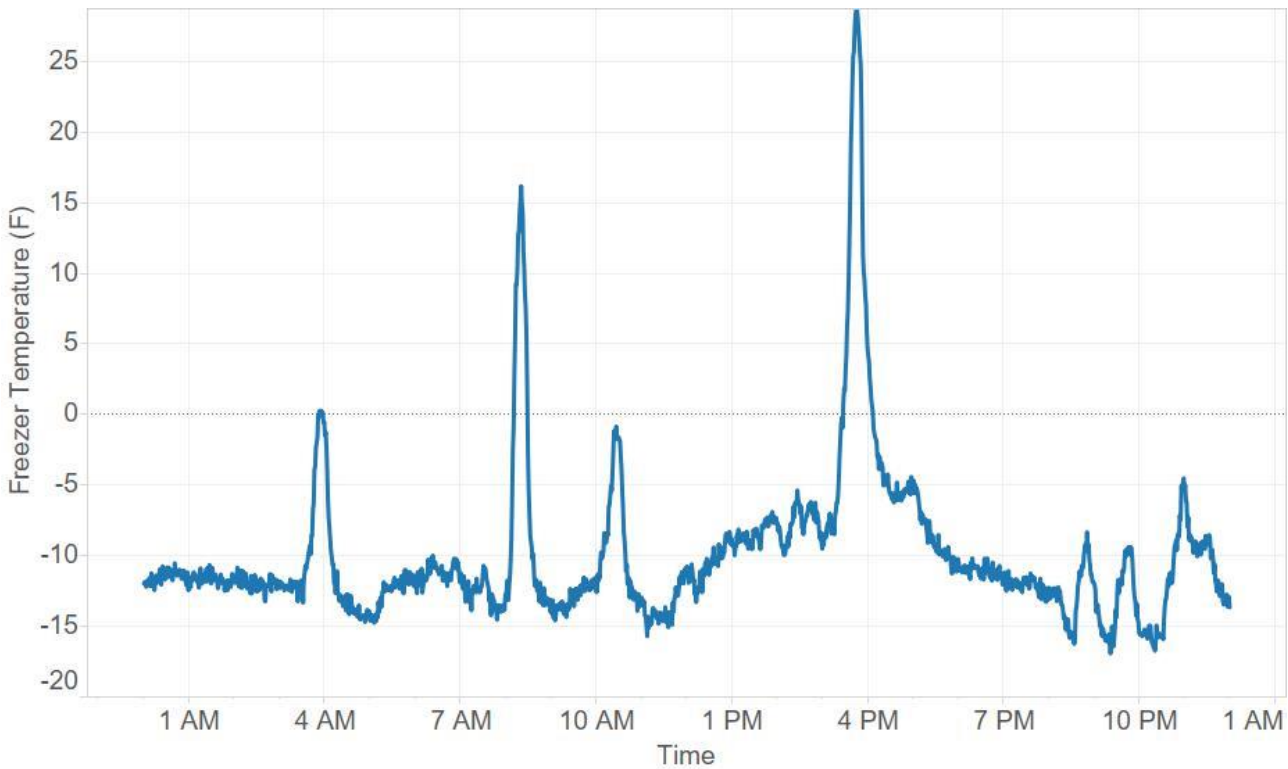

FIGURE 20 FREEZER AIR TEMPERATURE FOR JULY 15, 2016

Considering freezer temperatures more generally over the course of the year, Figure 21 shows a box-and-whisker plot of the hourly average temperatures inside the freezer, for each month and separately for baseline and new equipment if both ran in a given month. The box shows the first quartile, second quartile (the median temperature) and the third quartile. The whiskers extend to 1.5 times the inter-quartile range. The individual dots show each hourly measurement. The median temperature was mostly similar between baseline and new equipment, until October when the median temperature of the baseline equipment was lower by about 2°F. The cause of this is most likely the reduced R507A suction pressure. As mentioned above, the manufacturer reduced the suction pressure for their ice cream freezer; this freezer shares a suction manifold with the freezer under test, and it is suspected that this change led to the lower average temperature. Also, the extents of the temperature range for the new equipment, in blue, are considerably wider. For a typical month, about 13-15% of the hourly readings for the new equipment and about 7-10% extend above the inter-quartile range. A likely reason for this difference is the different sequencing of defrost heaters, as was addressed above: with the NH_3/CO_2 system, the two coils engage defrost at the same time, and the temperature deviates by a large amount for every defrost. For the baseline system, deviations are more frequent but smaller in magnitude because the two heaters do not usually defrost at the same time. Depending on the constraints of the facility, in some cases it may be useful to change the control of the defrost heaters to operate separately, particularly if large temperature variations are problematic.

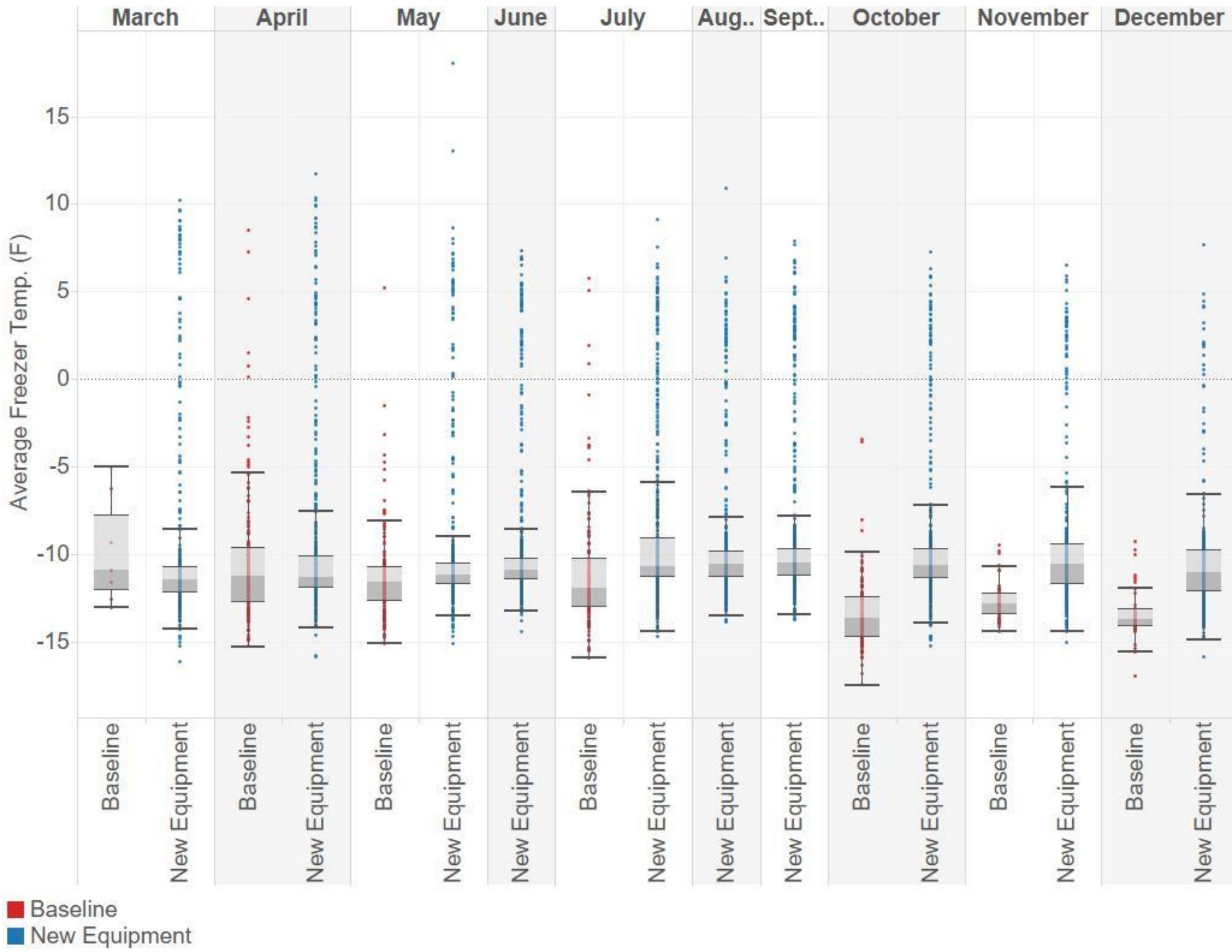

FIGURE 21 BOX AND WHISKER (MEDIAN – QUARTILES – 1.5X INTER-QUARTILE RANGE

DEMAND RESPONSE

This section details the results of demand response simulations run with the new equipment. Tests were performed using simple set-point adjustments within the freezer. The nominal freezer set-point is -15°F. For a pre-cool event, the set-point could be set lower, then allowed to float up to normal during the event. For a utility-requested shed event, the set point could be set higher. Early testing revealed that to provide cooling lower than about -20°F, the settings of the CO_2 pressure on the system would have to be lowered. While this is possible via remote control from the manufacturer's Japan location (and could be programmed into future controllers for local control), an option for performing such a test was not available in the timeframe of this effort. Therefore, all adjustments were simple set-point adjustments without further refrigeration cycle adjustments.

The first test was spread over two days, Wednesday, August 17 and Thursday, August 18. On the first day, a simple "pre-cool" was performed, where the set-point was set to -20°F and held inside the freezer. The results are shown in Figure 22. This test was planned from 10:45 AM to 1:45 PM, but ran long, until approximately 3:00 PM. The purpose of this initial test was to set and hold a new, low set-point and observe the response in the room.

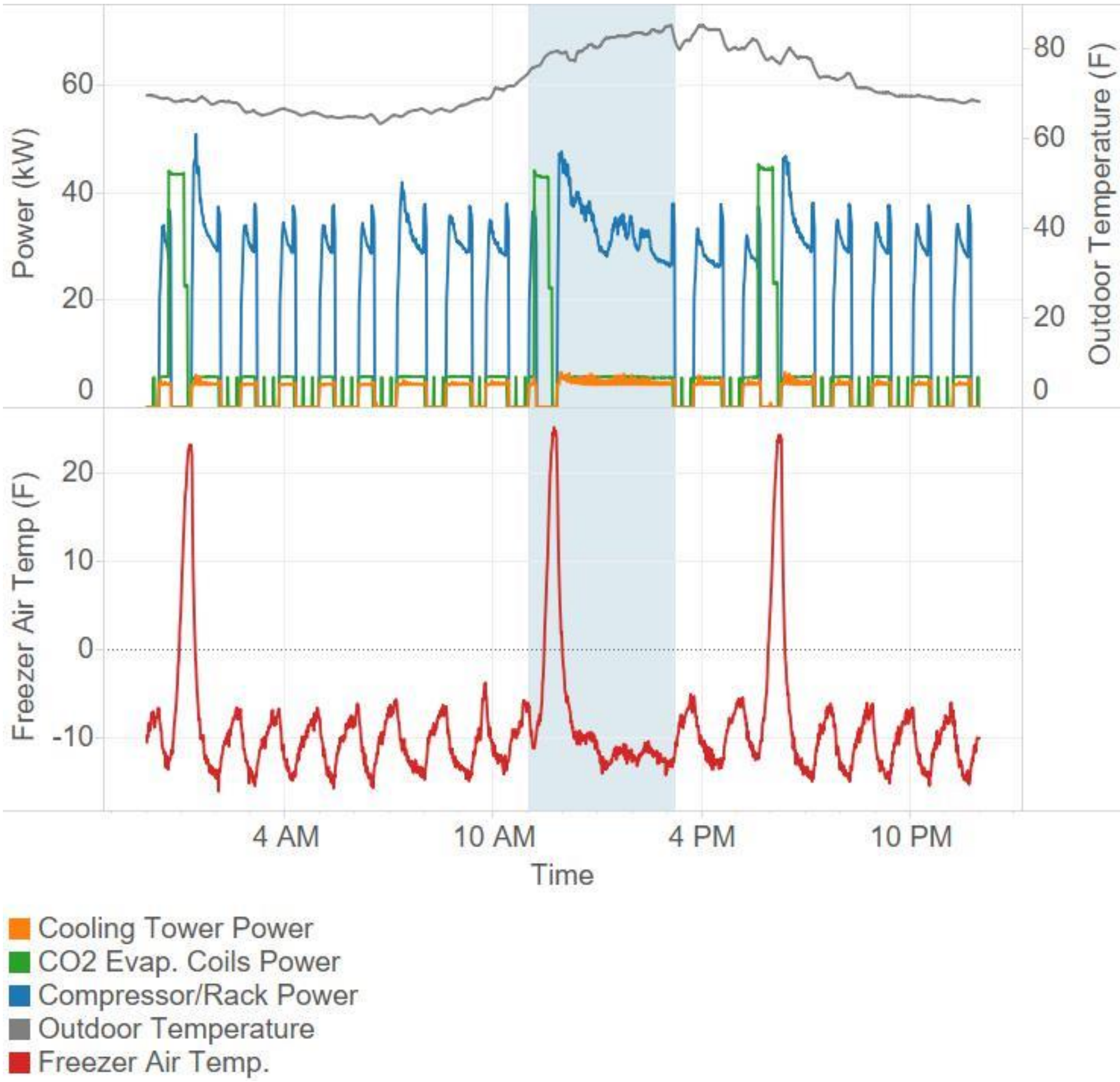

Figure 22 Pre-Cool Simulated DR Event, August 17

Shortly after the test began, a defrost initiated. After the defrost, a long period of compressor operation can be observed, and the temperature in the freezer (in this case measured near the middle of the space) stayed on the low end of its normal cycling range. The temperatures of air entering and leaving the CO_2 evaporator coils are shown, along with rack power for reference, in Figure 23. During the pre-cool period, the temperatures entering and leaving the coils do not change much, and from some slight increases (more pronounced in the measurements at Coil 1), it appears there is some changing load in the freezer during this time. After the pre-cool period, the system resumes operation as normal, and there does not appear to be a noticeable difference in behavior.

FIGURE 23 COIL AIR INLET AND OUTLET TEMPERATURES FOR CO_2 COILS ON DAY OF PRE-COOL DR SIMULATION

These results show that the ability to drop temperature below set-point may be limited by the temperature of the CO_2 being supplied. This is supported by the method of operation: the CO_2 reservoir is held as a liquid at a fixed pressure, which is maintained by the ammonia system. The controls of the Newton serve to maintain the condition of the CO_2, meaning that the supply of

CO_2 is at a roughly fixed temperature and pressure. The thermostat of the freezer only turns the CO_2 coils on and off. So, the temperature of air that can be reached with the coils is limited by the CO_2 temperature, and achieving much lower temperature in a reasonable timeframe would require adjusting the CO_2 conditions. This would be possible, and it would be possible to see more aggressive demand response measures, with integration of advanced controls to the system. However, the pre-cool as evaluated still serves a purpose: it ensures that the freezer is at the low end of the temperature range, and by holding the temperature lower, the temperature of the product in the freezer is likely lowered, which will in effect store some thermal energy. This effect was not directly measured, as product simulators were not installed.

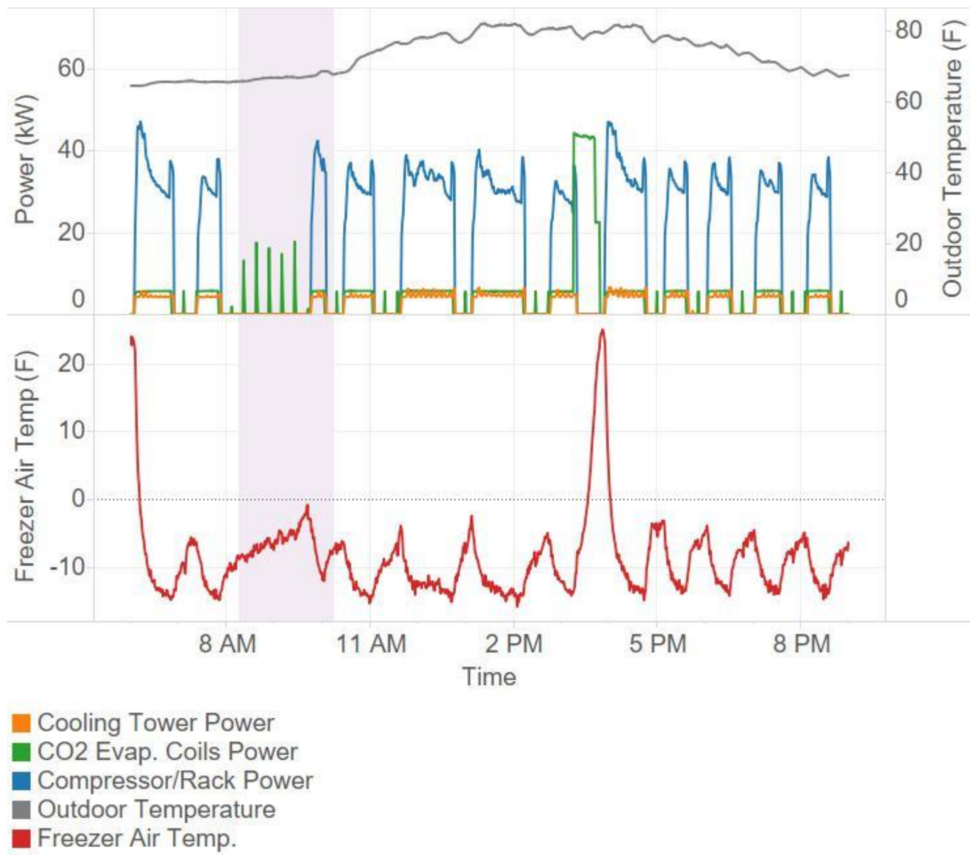

FIGURE 24 LOAD SHED SIMULATED DR EVENT, AUGUST 18

The following day, August 18, a test load shed event was run in the morning. The schedule for this event was approximately 8:15 AM to 10:15 AM. At the beginning of the event, the system was off as part of normal cycling. During the event, the temperature measured in the freezer increased from -15°F to 0°F between 7:51 AM and 9:41 AM, during which time the system did not run. As can be seen in the CO_2 evaporator coil power measurement, there is a short power usage approximately every 15 minutes. This is most likely the defrost heater briefly engaging. The data resolution is one minute, so power measured for under one minute cannot be observed in detail. The reason this occurred is not clear, and in subsequent shed tests, it did not occur. The effect of these spikes is small: the average power of all NH_3/CO_2 equipment during the period between cycles is 0.42 kW. Once the temperature in the freezer reached approximately 0°F, the system turned on and cooled until the temperature reached about -12°F (the normal shut-off is about -15°F). The temperature again begins to increase during the "off" interval. At approximately 10:30, the system turns on again, and normal operation resumes. The operation after the shed is not noticeably different than normal.

Several other demand response tests were run, with the pre-cool and shed in succession. A test on September 20 is shown in Figure 25. The pre-cool period was initiated at approximately 6:50 AM, and the load shed at approximately 10:00 AM. A defrost occurred during the middle of the pre-cool period. The shed took place from 10:00 AM to 1:00 PM, during which time the system was off for one hour, thirty-five minutes, and then there were two cooling cycles, approximately twenty minutes each. The shed was followed by normal operation with longer run-times to satisfy the -15°F set-point.

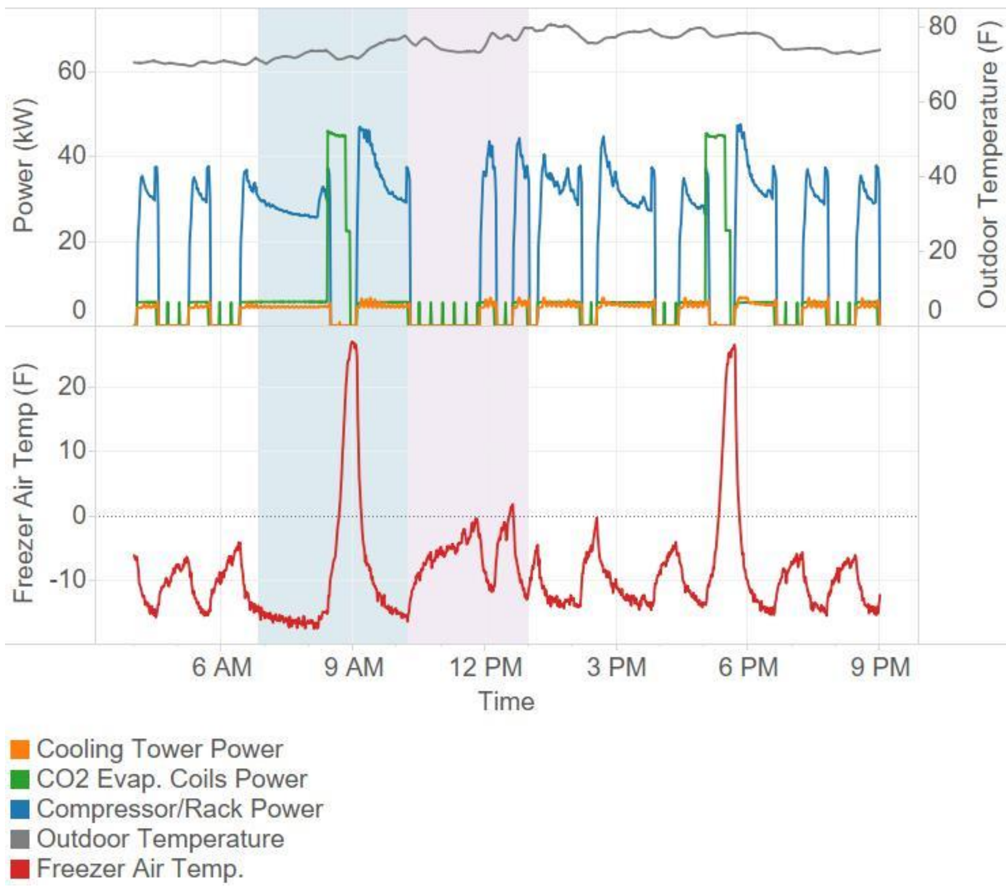

FIGURE 25 SIMULATED DR PRE-COOL AND SHED, SEPTEMBER 20

A similar event was run on September 21, with a targeted pre-cool period of 10:00 AM to 12:00 PM and shed from 12:00 PM to 2:00 PM. The results are shown in Figure 26. The behavior is similar to the test above, though the uninterrupted "off" time is shorter: a 38-minute off period is interrupted by a 20-minute on period. The system turns on again after 32 minutes, shortly after which the set-point is set back to normal, so the system remains on.

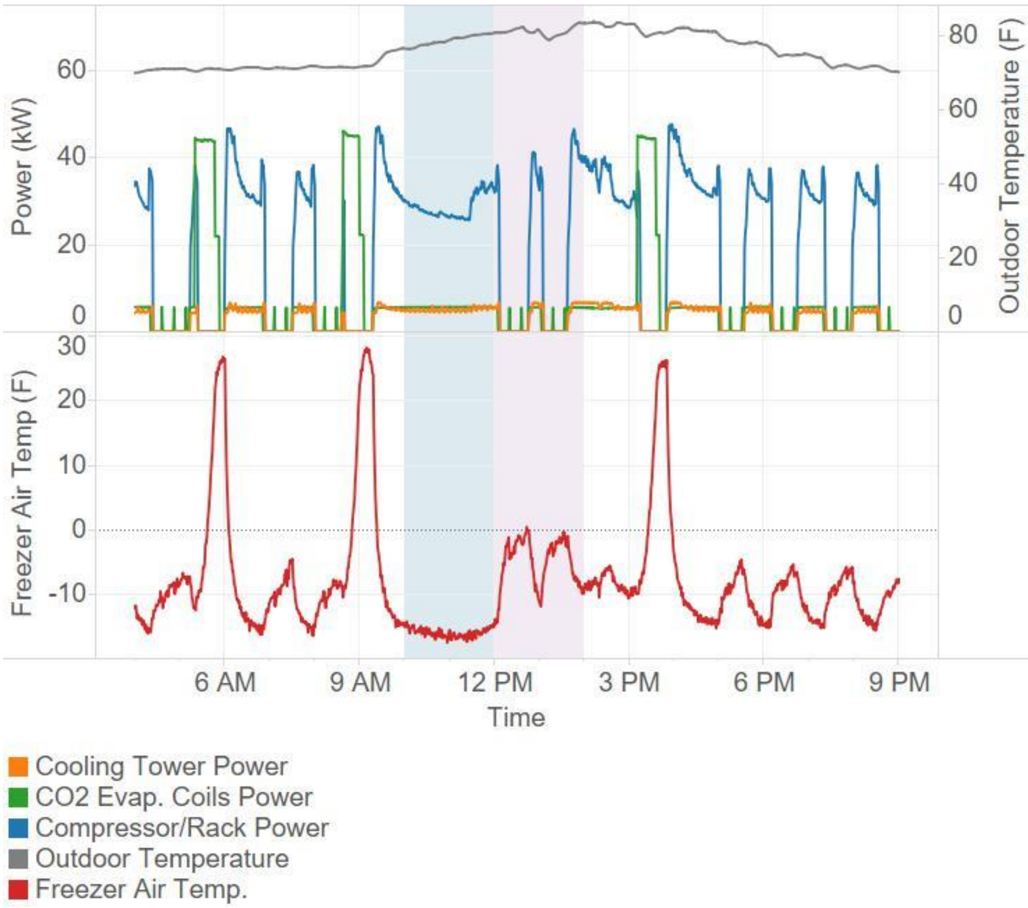

FIGURE 26 SIMULATED DR PRE-COOL AND SHED, SEPTEMBER 21

An additional test was run on September 30, with pre-cool beginning at 1:00 PM and shed from 5:00 PM to approximately 6:30 PM. The results are shown in Figure 27. During the shed period, there was one, 23-minute on-cycle, and the temperature in the freezer was held between -12°F and +2°F. The effect of the pre-cool is not pronounced like previous tests; there may have been a period of higher activity in the freezer during this time.

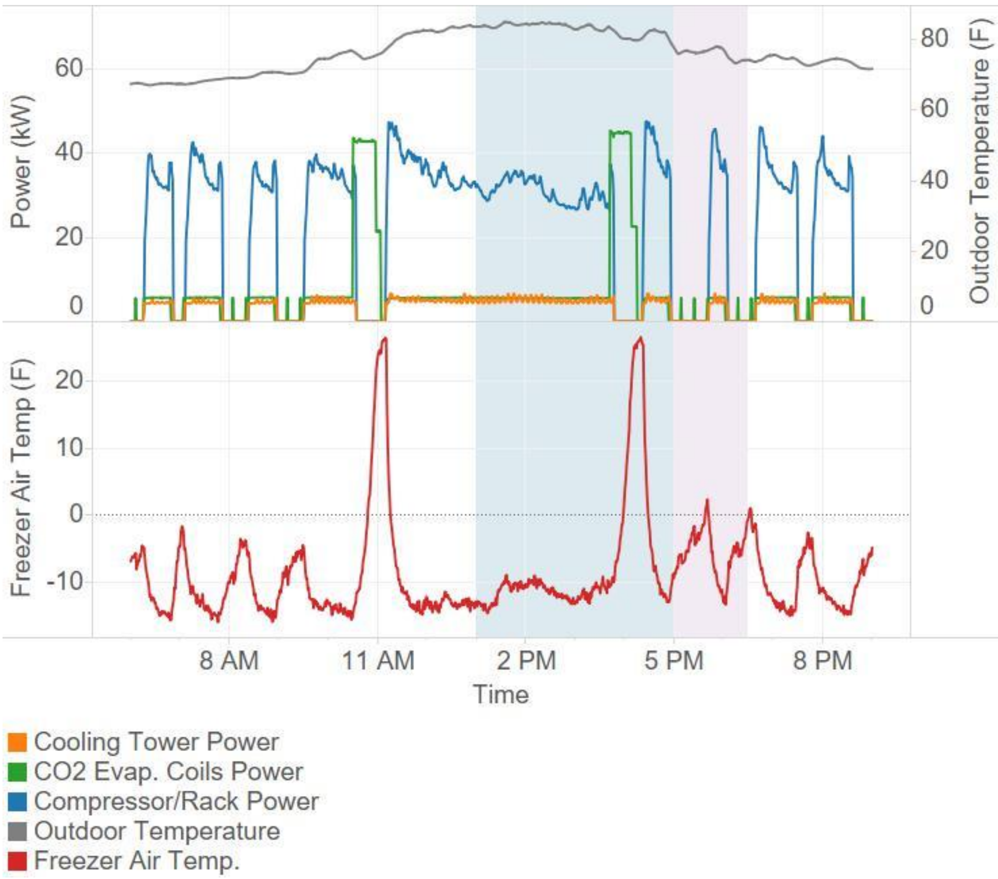

FIGURE 27 SIMULATED DR PRE-COOL AND SHED, SEPTEMBER 30

The change in load due to the DR events can be compared against the typical for similar weather. In this case, the hourly-average power for each day is compared with the average of all other (non-DR) weekdays in September in the following figures. The figures show the power of all NH_3/CO_2 equipment, as well as the R507A coil circuit because it includes heaters which affect the NH_3/CO_2 system. The DR pre-cool and shed time periods are approximated and drawn on each graph, in blue for pre-cool and red for shed. The data is hourly averages, with each hour indicated as, "hour beginning". For example, hour 13 indicates 1:00 PM through 1:59 PM. Since each simulated DR event represents only one day, some significant deviations from the monthly average are to be expected. Also, since the start and stop times were manually determined by personnel on-site and reported back through email, the start and finish do not necessarily align exactly with each hour. However, the resulting load shapes give a strong indication of the pre-cool and shed power profiles.

Figure 28 shows the DR test of September 20 compared with the average September weekday load shape. The hourly average power was 8-23 kW higher during the pre-cool period than average for the month; the power during the shed period was 12 kW and 20 kW lower than average during the first two hours of shed, and 8 kW lower than average during the third. After the shed, the power was a slightly lower than average for the next several hours.

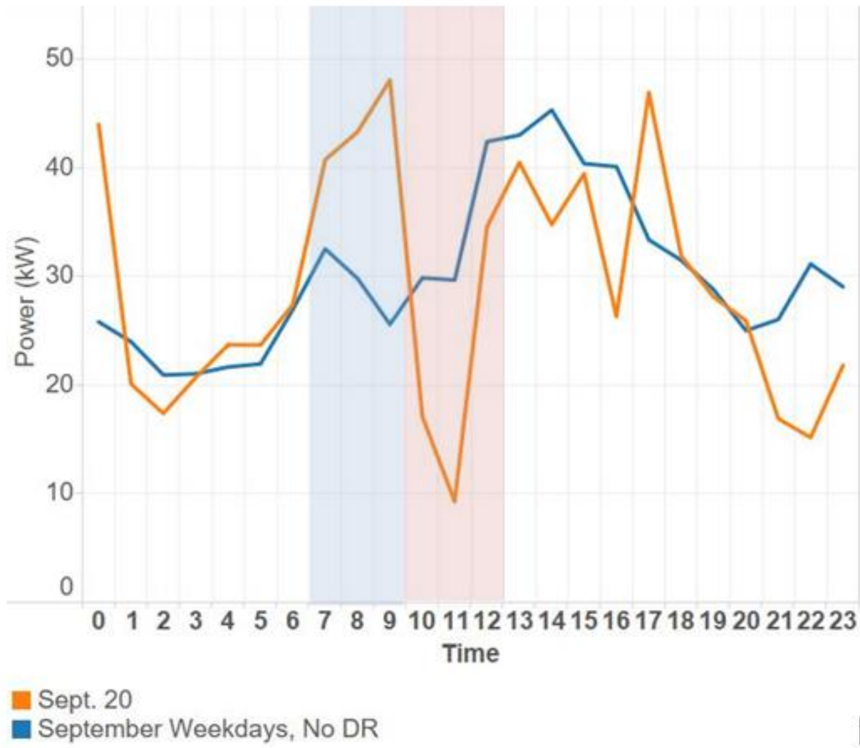

FIGURE 28 SEPTEMBER 20 DR EVENT COMPARED WITH SEPTEMBER NON-DR WEEKDAYS

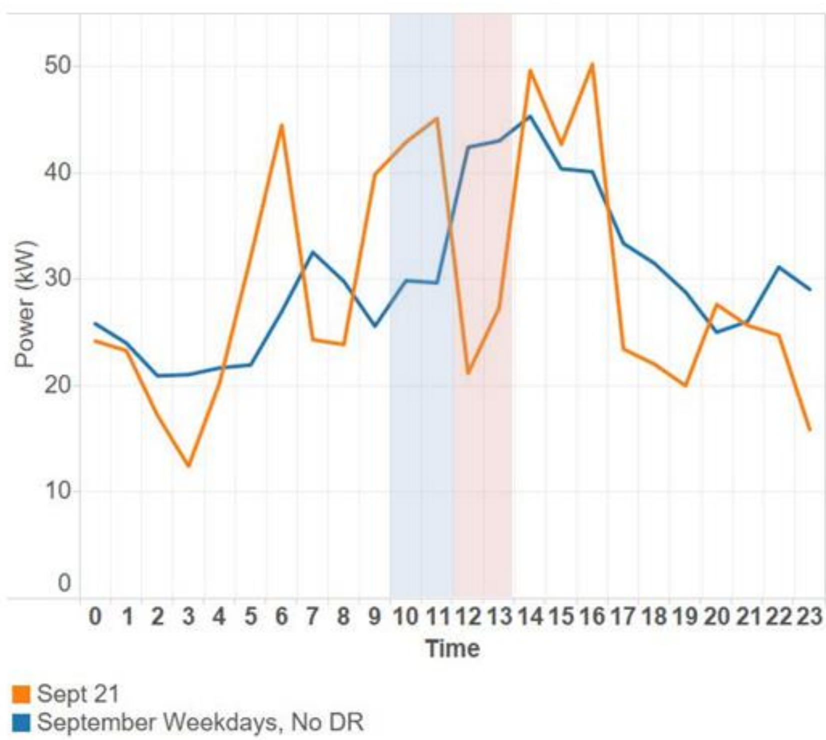

FIGURE 29 SEPTEMBER 21 DR EVENT COMPARED WITH SEPTEMBER NON-DR WEEKDAYS

Figure 29 shows the simulated DR event of September 21. The hourly average power was 13-15 kW higher than average during the pre-cool period, and 16-21 kW lower during the shed period. After the shed, the power was slightly higher for the next several hours.

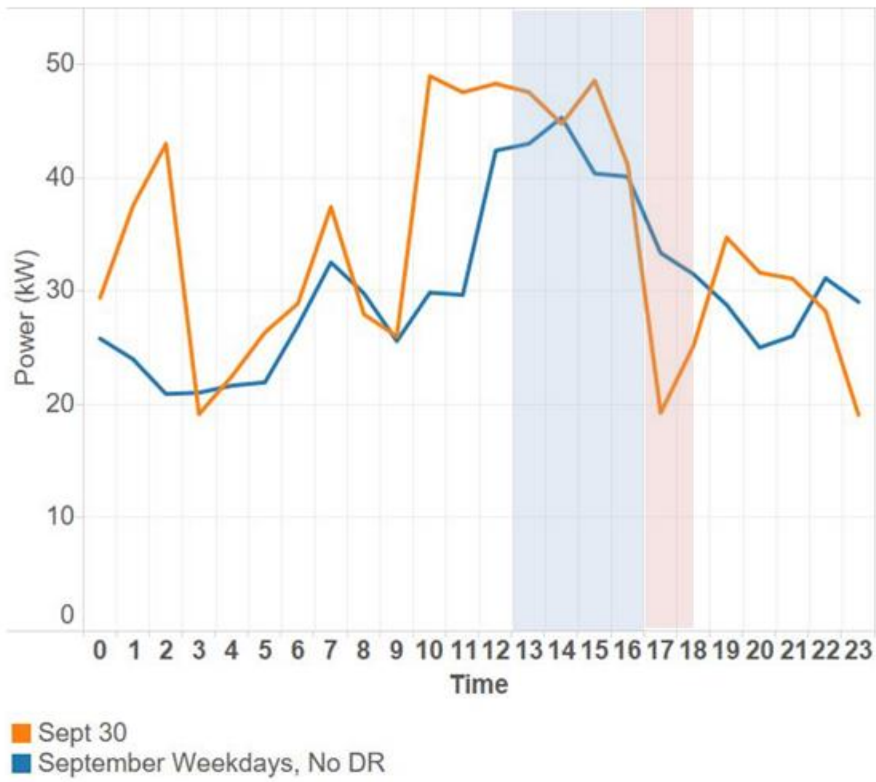

FIGURE 30: SEPTEMBER 21 DR EVENT COMPARED WITH SEPTEMBER NON-DR WEEKDAYS

Figure 30 shows the September 30 simulated DR event. For this day, the power was considerably higher than average before the simulated event started, probably due to higher-than-average traffic in the freezer. During the designated pre-cool period, the power was between approximately the same and 8 kW higher than average. During the shed the average power was 14 kW lower during the full hour of shed (5:00 to 6:00 PM) and 6 kW lower for the following hour, during the first 30 minutes of which the system was set to shed. The average power was 5 kW to 6 kW higher for the next few hours.

The simulated demand response testing performed here shows that a pre-cool and shed can be performed with this equipment. Due to the system's control functionality, where CO_2 is maintained at a fixed condition and pumped on-demand to the freezer, the ability to over-cool is limited. This could be extended by allowing adjustment to the CO_2 conditions prior to or as part of the pre-cool period.

The ability to provide simple "on/off" load shedding is as much a function of the load itself as the equipment. The system under test was mostly able to fully power off during the shed for extended periods, with limited cycling to keep temperatures in the adjusted set-point range. This allowed reductions of average hourly power up to 21 kW, without requiring a large rebound or drastically altering the freezer condition. These shedding patterns could have significant financial impact, depending on the utility rate structure and incentives.

CONCLUSIONS

In this field study, a new NH_3/CO_2 refrigeration system was installed at a food processing facility in Irvine, California. The facility, which manufactures snack foods, has a walk-in freezer which previously used a R507A refrigeration system. The old system was left in place, enabling direct "Baseline/Treatment" testing to compare with the new equipment. Instrumentation was installed to monitor all power circuits relevant to the freezer refrigeration equipment, as well as temperatures in the freezer and outdoors, and other measurements.

The energy consumption was significantly lower with the new equipment, in the range of approximately 200-300 kWh per day lower in typical conditions. Each system also maintained a similar average freezer temperature in the early part of the study; after July baseline testing, a change was made to the R507A circuit to improve performance elsewhere in the facility, and baseline temperatures were slightly lower after that change. The freezer temperature deviated higher during defrost with the new equipment, for which both defrost heaters ran simultaneously, than for the baseline equipment which defrosted typically one coil at a time, leading to more frequent temperature changes but of smaller magnitude. The new system was noted to be considerably oversized for the existing load, so although it has a variable-speed compressor, it often ran in cyclic operation, particularly during low-load hours. The host expects to double the freezer capacity by adding an additional, near-identical freezer, and anticipates only CO_2 coils connected to the new system will be used for the new freezer.

Demand response simulations were also performed. Since the NH_3/CO_2 system's operation calls for maintaining CO_2 at a fixed condition, and pumping liquid to the freezer, it was determined that pre-cool capability was limited since the instantaneous capacity at the evaporator coil is essentially fixed; to modify this, the CO_2 set-point conditions could be changed, but this could not be implemented in time to include in this evaluation. However, testing was performed by adjusting the freezer temperature set-point. The testing showed an ability to increase load during the pre-cool period, and to shed load by increasing the temperature set-point. The shed, which was executed with a simple 5°F increase in set-point temperature, resulted in average hourly power in the range of 14-21 kW for the first hour of load shed. Generally, the system did cycle on and run later in the shed event, but run times were shorter than in typical cycling. Also, defrost operation interrupted the demand response events (particularly the pre-cool) several times. The defrost heaters were not controlled as part of the demand response study. The overall performance of demand response would be improved with increasing control capability: the ability to shift defrost forward or back in time, and the ability to adjust the CO2 conditions to facilitate a deeper pre-cool, would both potentially offer a greater resource.