FINAL REPORT

Impact Evaluation of PY2022 Custom Electric Installations

Rhode Island Energy

Date: September 13, 2024

DNV

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List of acronyms used in this report

BMS	building management system
CDA	comprehensive design assistance
CHP	combined heat and power
C&I	commercial and industrial
EMS	energy monitoring system
EUL	effective useful life
FCM	forward capacity market
HVAC	heating, ventilating, and air-conditioning
ISP	industry standard practice
LSAF	lifetime saving adjustment factor
M&V	measurement and verification
MBSS	model-based statistical sampling
ML	measure life
non-ops	non-operational parameters
PY	program year
PYR	plan year report
RR	realization rate
SEMP	Strategic Energy Management Plan
SCADA	supervisory control and data acquisition
TMY3	typical meteorological year 3
TA study	technical assistance study
MRD	minimum requirements document
EERMC	Energy Efficiency Resource Management Council

1 INTRODUCTION

1.1 Study purpose and objectives

This document is the final report for DNV's Impact Evaluation of Program Year (PY) 2022 Custom Electric Installations, conducted for RI Energy, carried out from August 2023 to August 2024. The DNV team includes expertise from our partner firm DMI.

The primary objective of the impact evaluation was to provide verification and re-estimation of energy and demand savings for a sample of statistically selected custom electric projects through site-specific verification, monitoring, and analysis. The results of this study, combined with those from previous years, were used to determine the gross realization rates to be used for custom electric energy efficiency projects implemented in 2025 and will be updated annually as subsequent impact evaluations are completed.

The key objectives of this evaluation were as follows:

1. **Evaluate savings impacts of PY2022 custom electric projects** to be pooled with the results of the recently completed PY2020 and PY2021 studies. This study aimed to quantify:
    a. Achieved electric energy savings for custom non-lighting projects, with a targeted combined sampling precision of ±15% at 90% confidence when pooled with the results from the PY2020 and PY2021 studies.
    b. Summer and winter on-peak demand realization rates calculated with a precision target of ±20% at 80% confidence for custom non-lighting when pooled with the results from the PY2020 and PY2021 studies.
    c. Percent on-peak realization rates calculated with a precision target of ±20% at 80% confidence for custom non-lighting for the three-year rolling average.
2. **Evaluate lifetime savings adjustment factors (LSAF)** for PY2022 using the results for the sites included in the study and the sampling weights calculated for Objective 1 above. LSAF was not calculated until PY2020. Therefore, PY2022 (considered Year 3) was the first year that a three-year (rolling/staged; PY2020+PY2021+PY2022) LSAF for program planning purposes was calculated.

1.2 Organization of report

The rest of the report is organized as follows:

- Section 2: Methodology and Approach
- Section 3: Data Sources
- Section 4: Analysis and Results
- Section 5: Conclusions and Recommendations
- Appendices

2 METHODOLOGY AND APPROACH

This section provides details on the PY2022 impact evaluation summarizing both methodology and approach.

This study is the sixth annual C&I custom electric impact evaluation in Rhode Island using the rolling average approach. Like last year's study, this year's study calculated savings and realization rates for non-lighting projects only due to the stability of lighting RRs within the state over the last several years. All 10 sites in the PY2022 sample received on-site full M&V, with loggers installed or data collected on site.

Custom non-lighting projects include HVAC systems and controls, industrial process systems, and other non-lighting energy-using equipment.

2.1 Sample development

The basis for sample development is that the results from this evaluation achieve the targeted statistical estimates mentioned in Section 1.1

2.1.1 Tracking data review

DNV reviewed project parameters found in the raw tracking data files received from RI Energy to uniformly classify measures as lighting or non-lighting projects to prepare the data for the sample design process. The data included a total of 170 non-lighting applications at 102 unique sites. As mentioned earlier, the scope excluded lighting projects in this round of evaluation. More details of this sampling approach are provided below.

Out of the 102 sites, DNV removed another 23 sites that claimed less than 2,000 kWh in energy savings through the program to be cost-effective, leaving DNV with 79 unique accounts. The 23 sites removed constituted 0.2% (<1%) of the total program savings. PY2022 claimed 13.9 million gross annual Energy (kWh) savings, nearly 47% less than the previous year. The reduction in savings is primarily due to two large sites in the PY2021 sample that had a combined total of 15.35 million kWh. Similar sized projects were not completed in PY2022. Table 2-1 details the gross annual energy and peak demand savings for PY2022.

Table 2-1. PY2022 gross annual energy and peak demand savings

Total unique accounts (sampling unit)	Total energy savings (kWh)	Total peak summer savings (kW)	Total peak winter savings (kW)
79	13,916,893	1,864.46	2,090.83

2.1.2 Sampling plan

Model-based statistical sampling (MBSS) techniques were used to develop the sample design. The sample design's general principle is that each year's results would need to achieve ±26% precision at the 90% confidence interval to maintain a three-year pooled result of ±15% precision at 90% confidence for non-lighting gross energy realization rates. The assumed error ratio[1] (ER) for PY2022, presented in Table 2-2, has been carried forward since the PY2020 evaluation, since it has allowed the evaluations to achieve their precision goals with the resulting samples.

Table 2-2. Sampling targets

Annual sampling target	3-year pooled sampling target	Error ratio
±26% on non-lighting energy (kWh) at the 90% confidence interval	±15% on non-lighting energy (kWh) at the 90% confidence interval	PY2022 = 0.45

Table 2-3 presents the sample design for PY2022 and samples from the previous evaluations of PY2020 and PY2021 that are used in support of the three-year combined design. The PY2022 RP was proposed to be 24.0%. The combined

[1] Error ratio is a measure of the population variability between the x (known for population) and y (known only for the sample) variables. The error ratio is defined as the ratio between (a) the sum or average of the residual standard deviations of all customers, and (b) the sum or average of the expected values of y.

precision was proposed to be 10.2% @ 90% confidence for non-lighting: within the goal of ±15 % RP. Table 2-3 details the total savings for each year, their sample size, and RP based on each program year of the three-year pooled result.

Table 2-3. 2022 project sample design and estimated relative precisions

End-use	Program year	Energy savings (kWh)	Sample size	RP @90% CI
Non-lighting	2020	10,676,671	10	±28.2% (actual)
	2021	26,073,183	10	±15.8% (actual)
	2022	13,916,893	10	24.0% (proposed)
Non-lighting (3-year rolling)	**2020+2021+2022**	**50,666,747**	**30**	**10.2% (proposed)**

2.1.2.1 PY2022 site weight calculation

Case weights have been created for each of the 10 sites by determining the total number of observations in the stratum and dividing by the number of evaluated observations.

Table 2-4. Stratification and weighting

Strata#	Population (N)	Sample (n)	Weight
1	63	5	12.60
2	15	4	3.75
3	1	1	1.00

For the PY2022 annual evaluation, each site has a single case weight based on the stratum they were assigned to.

2.1.3 SEMP sub-sample

In the PY2022 primary sample there was one SEMP[2] site (RICE22S011) with 23 separate projects implemented over a three-year period. To be cost-effective and efficient with the customer for this site, a secondary sample of six projects was designed to represent this site as follows:

Table 2-5. SEMP sub-sample

Stratum #	Total projects	Total savings (kWh)	Average savings per project (kWh)	Sampled projects
1	17	190,194	11,188	3
2	5	269,228	53,846	2
3	1	126,623	126,623	1
Overall	23	586,045	25,480	6

These projects were rolled up to the primary sampling unit (site) during analysis so that in aggregate they only counted as a single observation for the purpose of variance calculations.

2.2 Description of methodology

The calculation of the current year (Year 3) and Year 2 realization rates is different from Year 1, as an imputed operational adjustment was not necessary for the two most recent years. Section 4.2.4 explains the process for calculating the current and three-year realization rates, and APPENDIX C explains the process for imputed adjustments.

[2] Strategic Energy Management Plan. The SEMP program allows customers to plan for and implement projects over a 3-4 year period.

DNV has updated the yearly realization rates as part of this custom electric evaluation framework. The evaluation also generated lifetime savings adjustment factors (LSAFs) in this round. See APPENDIX D for more information on LSAF.

2.3 Customer outreach

Project engineers reached out to customer site contacts using an RI Energy-approved communication protocol and the information provided in the project files. During this initial outreach, the engineers discussed the purpose of the outreach, facility operation and usage, the scope of measures installed, the availability of onsite trend/SCADA/production data, any other applicable parameters relevant to the evaluation and confirmed the site's ability and willingness to participate in the evaluation. Efforts were made to minimize pre-recruitment evaluation activities until the customer site contact indicated they would accommodate the evaluation process. A backup site was selected if the site contact was unresponsive or refused to participate in the evaluation.

Two backup sites were selected to replace sites in the primary sample. DNV Site ID RICE22N080 refused to participate and was replaced by RICE22N087. Site ID RICE22N090 had facility electrical problems impacting the installed measure and was replaced by RICE22N042.

With RI Energy's input on the site evaluation plan, the DNV team contacted the customer to schedule an onsite visit at a day and time convenient for the customer site contact.

The DNV team conducted site visits to collect the data listed in the site evaluation plan for each site. In general, each data collection visit consisted of verifying the installed technology, quantities, a discussion with facility personnel regarding installed measure(s) and the baseline conditions that existed before the measure(s) installation.

2.3.1 Onsite M&V

Onsite M&V data collection included physical inspection, an interview with facility personnel, observation of site operating conditions and equipment, metering of equipment usage, and collection of facility-provided data. The physical inspection focused on verifying measure installation and expected operation. In some cases, multiple facility interviews and/or equipment vendor interviews were completed to ensure an accurate understanding of the operating practice.

For all sites, instrumentation and metering equipment such as power recorders, time-of-use (TOU) current loggers, plug load monitors, and temperature loggers were installed to monitor the usage of operating equipment and conditions of the impacted systems. Production data and EMS trends were also collected when available. Each site report includes a full description of the data collected and received, and, where applicable, data from installed meters.

A unique savings analysis was created for each sampled project. When required, a typical meteorological year (TMY3) dataset of ambient temperatures was used for temperature-sensitive calculations. Energy savings were either calculated by the hour in an 8,760-hour spreadsheet or allocated to each hour in the year to estimate on-peak kW and kWh savings impacts. Each analysis provided estimates for annual kWh savings, on-peak kWh savings, and on-peak demand (kW) savings at the times of the winter and summer peaks, as defined by the ISO New England Forward Capacity Market (FCM). All coincident summer and winter peak reductions were calculated using the following FCM definitions:

- *Coincident summer on-peak kW reduction* is the average demand reduction that occurs overall hours between 1 PM and 5 PM on non-holiday weekdays in June, July, and August.
- *Coincident winter on-peak kW reduction* is the average demand reduction that occurs overall hours between 5 PM and 7 PM on non-holiday weekdays in December and January.

Each site report details the specific analysis methods used for each project, including baseline and evaluation algorithms, assumptions, and calibration methods where applicable.

Engineers submitted draft site reports to RI Energy upon completion of each site evaluation. The DNV team responded to the comments received and submitted revised reports for comment. A sample of reports was also submitted to the EERMC consultant team for review. The final site reports are included in APPENDIX E. The body of this report provides an overview of the evaluation methods and findings only.

3 DATA SOURCES

To support the findings of the study, the DNV team used the following data sources:

- PY2022 tracking data provided by RI Energy
- PY2020 and PY2021 impact evaluation results and historical operation adjustment factors
- Project files, which typically include the following: applications, BCR screenings, invoices, technical assistance studies, applicant savings calculations, and post-installation reports.
- Onsite audit observations and data collection, including inspection and verifications of equipment, nameplate data, staff interviews, and vendor interviews
- Customer- or vendor-supplied operational data (metered or trended data).

4 ANALYSIS AND RESULTS

4.1 Introduction

A total of 10 sites were evaluated within the PY2022 population for an onsite visit with full M&V. Full M&V is considered a traditional measurement and verification (M&V) that involves onsite measurements using power, time-of-use meters or validated trend data and measure verification. A summary of sampled projects is listed in APPENDIX A.

Table 4-1 presents the adjustment factors used in the evaluation.

Table 4-1. Adjustment factors for site evaluation

	Adjustment factors						
Ratio name:	Non-operational adjustments					Operational adjustments	
Obtain during:	In-depth file review			First site visit		Logger installation	
Factor:	Baseline	Methodology	Tracking & admin	Technology	Quantity	Operation	HVAC interactive

Calculating the results from this study involved the following steps:

- The evaluated non-operational adjustments and operational adjustments for PY2022 were multiplied together to arrive at an overall realization rate (RR) for PY2022.
- The realization rate for PY2022 was combined with the overall RRs from PY2020 and PY2021 in proportion to their respective first year savings relative to the total savings for the three program years to provide an estimate of the overall three-year RR.

4.2 PY2022 results

4.2.1 PY2022 site-level discrepancies and RR

This section provides an overview of the top five discrepancies from PY2022 that had the biggest difference in site-level tracking and evaluated results. For each of the 10 sites in the PY2022 study, site engineers identified factors that led to differences between the program reported tracking savings and the evaluated savings. The factors are classified into seven categories: baseline, methodology, tracking/administrative, technology, quantity, HVAC interaction, and operational. A more discrete breakdown of possible differences and how they are categorized is presented in **Error! Reference source not found.**.

In PY2022, seven out of 10 sites reported savings below 100% of ex-ante estimates with major discrepancies in operation. The highest RR was 108% at site ID RICE22N087. The lowest RR was 39% at site ID RICE22N095. More details on each site and their discrepancies can be found in the individual site reports in APPENDIX E.

Table 4-2. Discrepancy factors and their mapping to major categories

Major discrepancy category	Discrepancy definition or examples
Baseline	Change in the baseline of the post-retrofit condition
Methodology	Accuracy/appropriateness of analysis methodology Calculation changes Non-metered data input updates
Tracking/Admin	Accuracy of tracking savings Errors during claimed savings input Savings changed but not changed in tracking savings
Technology	Differences in proposed vs. installed technology or measure type
Quantity	Quantity of installed equipment is different Boiler combustion efficiency Difference in equipment hours of operation Different equipment load profile
Operational	Inaccurate pre-project characterization Steam operating pressure difference System optimization or programming not implemented Faulty or improperly installed equipment Operating temperature differences
HVAC Interaction	Interactive effects

The following five sites had the largest discrepancies with respect to savings. Including the sites with a quantity, baseline or HVAC interaction discrepancy, the major source of each site's discrepancy was an operational adjustment.

RICE22N095: Quantity and Operational – the impacted chiller, AHU, and pump loads were less than the tracking estimate, which reduced savings by 60%. Less than 1% of the site discrepancies are attributable to the quantity adjustment. The site had an evaluated energy realization rate of 39%.

RICE22N054: Operational – the hours of operation and cooling load of the RTU was less than the tracking estimate, which reduced savings by 49%. The site had an evaluated energy realization rate of 51%.

RICE22S068: Baseline, HVAC interaction and Operational – Baseline (-2.1%) and HVAC interaction (+0.5%) adjustments made up for a total adjustment of -1.6% with respect to overall site savings. The remaining four discrepancies were all operational, with two being positive and two being negative. The single biggest operational adjustment was negative, coming from the result of a disabled measure (-76%). The total operational adjustments for this site were -47.3%. The site had an evaluated energy realization rate of 51%.

RICE22N093: Operational – the evaluator found the operating load to be less than tracking, which reduced savings but found the evaluated hours of operation to be slightly higher than tracking which increased savings. The total operational adjustment was -32%. The site had an evaluated energy realization rate of 68%.

RICE22N013: Operational – the evaluator found the impacted system to have different condensing temperatures, ton-hours of cooling, operating load, and efficiency, which all reduced savings by -36%. One operational discrepancy increased savings by +10%. The site had an evaluated energy realization rate of 74%.

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4.2.2 Combined program-level results

This section presents rolled-up/program-level realization rates by combining PY2020, PY2021 and PY2022 evaluated sample results.

The site-level evaluation results were aggregated using the final case weights for each respective year. The realization rates for each year were calculated by taking a product of Operational and Non-Operational Adjustment factors and then applied to total tracking savings to determine their total evaluated savings for that year. As discussed above, these one-year RRs were then used to calculate the three-year rolling RR.

Table 4-3 presents the non-lighting realization rates for each year and the combined prospective realization rate for the custom electric program in RI to be used to estimate 2025 savings. The combined RR for non-lighting meets the targeted relative precision (RP) of ±15% at a 90% confidence interval (CI) with a value of ±10.8% for the three-year rolling RR of 81.4%.

Table 4-3. Combined non-lighting realization rates (kWh)

Non-lighting	RI			Combined results
	PY 2020	PY 2021	PY 2022	PY2020-+ PY2021+PY2022
Tracking energy savings (kWh)	10,676,671	26,073,183.00	13,916,893.00	50,666,747
Sample size (n)	10	10	10	30
RR	68.6%	88.4%	78.3%	81.4%
Relative precision @ 90% CI	±28.2%	±15.8%	±12.8%	±10.8%
Operational results	75.4%	86.8%	78.5%	
Non-operational results[3]	91.0%	101.8%	99.8%	

Table 4-4 and Table 4-5 present prospective realization rates for summer and winter peak demand (kW) savings, and Table 4-6 presents prospective realization rates for percent on-peak energy savings. Both summer and winter peak demand (kW) savings RRs decreased from PY2021 but increased compared to PY2020. The three-year rolling/combined results for both summer and winter peak demands met the target precision of ±20% at 80% CI.

Table 4-4. Prospective realization rates from evaluated summer peak demand (kW) savings for non-lighting measures

Non-lighting	RI			Combined results
	PY2020	PY2021	PY2022	PY2020+ PY2021+PY2022
Tracking summer demand (kW)	1,441	3,099	1,864	6,405
Sample size (n)	10	10	10	30
RR	52.7%	83.3%	74.3%	73.7%
Relative precision@ 80% CI	±32.9%	±15.6%	±32.7%	±13.2%
Operational results	56.4%	80.2%	72.0%	
Non-operational results	93.5%	103.9%	103.1%	

[3] Starting in 2020, the non-operational results (non-ops) ratio was calculated using two factors. Non-ops 1 and non-ops 2; non-ops 1 includes discrepancies from Baseline, Methodology and Administrative adjustment factors while non-ops 2 includes Technology and Quantity adjustment factors.

Table 4-5. Prospective realization rates from evaluated winter peak demand (kW) savings for non-lighting measures

Non-lighting	RI PY2020	PY2021	PY2022	Combined results PY2020+ PY2021+PY2022
Tracking winter demand (kW)	1,168	3,685	2,091	6,944
Sample size (n)	10	10	10	30
RR	70.5%	120.6%	75.7%	98.3%
Relative precision@ 80% CI	±26.8%	±23.7%	±22.6%	±14.8%
Operational results	72.3%	115.5%	76.9%	
Non-operational Results[4]	97.5%	104.4%	98.4%	

Table 4-6. Prospective realization rates from evaluated % on-peak energy savings for non-lighting measures

Non-lighting	RI PY 2020	PY 2021	PY 2022	Combined results PY2020+ PY2021+PY2022
% on peak energy	10,676,671	26,073,183	13,916,893	50,666,747
Sample size (n)	10	10	10	30
RR	91.1%	74.7%	100.4%	85.0%
Relative precision@ 80% CI	±8.7%	±7.1%	±14.4%	±5.4%
Operational results	92.5%	70.7%	99.9%	
Non-operational results	98.5%	105.6%	100.5%	

4.2.3 Lifetime savings adjustment factors (LSAFs)

Lifetime savings adjustment factors were developed for the third time in this study using the weighted tracking and evaluated lifetime savings. DNV also calculated the first three-year pooled LSAF in this study. The LSAFs for non-lighting are provided in Table 4-7. As shown below, the lifetime savings RR for PY2022 is 99.0% and the three-year pooled RR is 102.1%. The methodology for these calculations can be found in APPENDIX D.

Table 4-7. Custom non-lighting LSAFs

Non-lighting	RI PY2020	PY2021	PY2022	Combined results PY2020+ PY2021+PY2022
LSAF	97.9%	104.9%	99.0%	102.1%
Relative precision@ 90% CI	±43.0%	±22.2%	±19.1%	±15.8%

[4]Starting in 2020, the non-operational results (non-ops) ratio is calculated using 2 factors. Non-ops 1 and non-ops 2; non-ops 1 includes discrepancies from Baseline, Methodology and Administrative adjustment factors while non-ops 2 includes Technology and Quantity adjustment factors.

Table 4-8. Custom non-lighting lifetime savings (kWh)

Non-lighting	RI PY2020	PY2021	PY2022	Combined results PY2020+ PY2021+PY2022
Lifetime kWh (annual weighted)	10,676,671	26,073,183	13,916,893	50,666,747
Sample size (n)	10	10	10	30
RR	67.2%	92.7%	77.5%	83.2%
Relative precision@ 90% CI	±32%	±16%	±14.2%	±11.6%

Six sites in PY2022 had tracking measure lives that were equal to the evaluated measure life. The remaining four sites had evaluated measure lives that were updated based on site findings.

4.2.4 PY2022 RR and combined program RR calculation methodology

This section discusses the methodology to calculate combined program level and the PY2022 realization rates. Historical operational adjustment results were used to impute data for PY2020 but not the PY2021 and PY2022 samples. PY2020 used site-specific operational adjustments for 7 out of 10 sampled sites, with 3 out of 10 imputed from the 7 sites. Because they were full M&V, PY2021 and PY2022 each had site-specific operational adjustments for each of their 10 sampled sites. Individual site RRs are shown in APPENDIX B.

Calculation of Combined Program RR:

$RR_{1\text{-}3} = (S_1RR_1 + S_2RR_2 + S_3RR_3)/S_T = q_1RR_1 + q_2RR_2 + q_3RR_3$

That is, the three-program year (PY) RR is the savings-weighted average of the three separately estimated RRs.

Where,

1- represents PY2020, 2 is PY2021, 3 is PY2022 and T is total (2020+2021+2022)

S_y - Population tracked savings of PY-y

S_T- population tracked savings for all three PYs combined ($S_T = S_1 + S_2 + S_3$)

q_T- percentage of three-year population tracked savings represented by each program year

RR_3 = Realization rate calculated for this program year

RR_1 and RR_2 were calculated in previous studies and have not been readjusted as part of this study. Additional details regarding prior year RRs that required imputation of operational adjustments, along with their associated standard error calculations, are provided in APPENDIX C.

Calculation of RR_3:

RR_3 was calculated using a similar, but simplified, methodology as compared to prior years where some sites did not receive full M&V and so needed an imputed operation adjustment (e.g., RR_1). Since the full sample of sites this year received an operational evaluation, no imputation to operational adjustments were needed.

Both the non-operational and operational realization rates (RR_{N3} and RR_{O3}) are calculated from the full sample using the full sample weights and the non-operational and operational adjusted savings respectively for the sample via the usual formulas.

The overall RR is the product of the operational and non-operational RR
$RR_3 = RR_{o3}\ RR_{N3}$

The constituent parts of the overall RRs for PY2022 are shown in Table 4-9. Note that multiplying the separate operational and non-operational results in this table may not produce the provided combined RRs exactly due to rounding, but the result should be within one-tenth of a percent compared to the combined value.

Table 4-9. Non-lighting prospective realization rates PY2022

Statewide results (n=10)	Annual MWh	Summer on-peak kW	Winter on-peak kW	% on-peak energy MWh
Total tracking savings	13,916	1,864	2,091	13,916
Total evaluated savings	10,896	1,385	1,583	13,967
Realization rate	**78.3%**	**74.3%**	**75.7%**	**100.4%**
Confidence interval	90%	80%	80%	80%
Relative precision	±12.8%	±32.7%	±22.6%	±14.4%
Operational results	78.5%	72.0%	76.9%	99.9%
Non-operational results	99.8%	103.1%	98.4%	100.5%

5 CONCLUSIONS, RECOMMENDATIONS, AND CONSIDERATIONS

5.1 Conclusions

This study's scope and approach were like the last round of evaluations (PY2020 and PY2021) in handling operational factors. There were no historical adjustments made for PY2022.

For custom non-lighting, the gross annual energy savings RR saw a net decrease from 88.4% in PY2021 to 78.3% in PY2022, but the RR was greater than the PY2020 RR of 68.7%. Overall, the combined three-year rolling value decreased from 88.2% (PY2019, PY2020, PY2021) to 81.4% (PY2020, PY2021, PY2022). The drop in this RR is attributable to the PY2019 RR of 104.1% dropping from the combined results.

The RRs for summer and winter on-peak demand showed a decrease in non-lighting peak demand RRs from PY2021 to PY2022 due to operational adjustments. The RR for % on-peak energy increased from PY2021 to PY2022.

The decrease in PY2022 energy RRs were due to seven out of 10 sites having energy realization rates less than 100%.

5.2 Recommendations

The RI Energy implementation team and vendors were extremely helpful in addressing questions and concerns during the evaluation. As a result of this assistance, DNV was able to perform a robust evaluation and collect additional research in support of RI Energy.

The DNV team makes the following recommendations based on the data collected, conclusions, results, and process of this impact evaluation. Many of these recommendations could immediately improve operational savings estimates with regards to load, hours of use, etc. As part of these recommendations, DNV suggests that RIE conduct an evaluability assessment specifically for measures in high savings projects that are a result of operational adjustments. This assessment could refine operational assumptions that overstated savings in evaluated projects.

Recommendation 1: This study's RI three year rolling non-lighting (81.4%) realization rate results shall replace the previous realization rates used by RI Energy beginning in PY2025. RI Energy should continue using 95.4% (from the previous evaluation) RR for lighting. The results from this study should be combined with the next round of custom electric impact evaluation, which will evaluate PY2023 applications and is expected to be applied to the PY2026 tracking savings.

Recommendation 2: DNV recommends that advanced control measures that utilize custom express tools are reviewed carefully to utilize site specific strategies.

For example, site RICE22N054 used a prescriptive EMS savings tool to calculate savings that did not accurately account for the strategy of the installed controls.

Recommendation 3: DNV recommends that RI Energy updates the 2014 National Grid Baseline document[5] that was being used to implement projects in PY2022. DNV specifically recommends for large ammonia refrigeration systems updating the minimum condensing temperature from 65°F to 70°F and standardizing VFDs on evaporatively cooled condenser fans.

The site evaluation report and associated memo report for DNV site ID RICE22N013 details the recommended 2014 baseline document updates and can be found in APPENDIX E.

[5] 2014-baseline-document-for-ma-and-ri.pdf (nationalgridus.com)

Recommendation 4: DNV recommends that large retrofit projects reliant on pre-existing conditions include operational data to calibrate the pre-retrofit energy calculations. This can be done with power metering, trend data, or even VFD screen readouts.

For example, site RICE22N095 was a large retrofit project where the applicant modeled baseline air handler power using manufacturer data, which overestimated actual power.

Recommendation 5: DNV recommends that RIE update the savings estimates for cleaning condenser and evaporator coil measures through more rigorous review of ex-ante calculations or pre/post metering. This includes developing typical diversity factors of different grocery store refrigeration systems.

For example, for DNV site ID RICE22N070, the tracking savings estimates were built on a non-empirically backed estimate that cleaning the condenser coils would restore a condenser's cooling capacity from a degraded value (caused by fouling) of 85% back up to 100%. Additional research completed as part of this site evaluation found that cleaning coils may have no significant effect on a unit's cooling capacity.

Recommendation 6: DNV recommends updating the measure life of door gasket measures from five years to one year due to the high replacement rate of door gaskets. This is based on documentation in the Regional Technical Forum (RTF) presentation[6] in the Pacific Northwest that are under the RTF's jurisdiction.

For example, for DNV site ID RICE22N070, the energy savings used for this door gasket measure cite the RTF documentation but doesn't employ the recommended measure life per the RTF documentation. The reason cited by the RTF for the 1-year measure life is based on an ADM report[7] on door gaskets completed for the CPUC.

Recommendation 7: DNV recommends that RIE revise the SEMP review process to include periodic meetings with RIE and the TA vendor to discuss ongoing RCx measures and review savings analysis methodology in more detail. This would improve the quality of savings by providing feedback to the site on an ongoing basis rather than correcting everything at once when the application is submitted.

Recommendation 8: DNV recommends RI Energy continue evaluating lifetime savings and reporting them at the site level in all future custom electric evaluations. A standard three-year rolling was calculated for the first time in this study (Year 3).

5.3 Considerations

The DNV team makes the following considerations based on the data collected, conclusions, results, as part of this impact evaluation.

Consideration 1: RIE should consider reporting lifetime savings in addition to annual savings. Reporting lifetime savings will allow a more detailed understanding of equipment measure lives and potential savings to better support implementation and evaluation efforts. This consideration aligns with MA, as lifetime savings are what the MA program administrators ultimately use for reporting.

Consideration 2: RIE should consider, based on the size of the project, that the implementer performs a post-installation metering period. By conducting a post-installation metering, the implementer can account for the actual post-installation conditions such as hours and load.

[6] 20190618DoorGasketPres.pptx | Powered by Box

[7] Microsoft Word - ComFac Evaluation V3 HIM Appendices _02-18-2010_.doc (calmac.org)

Consideration 3: RIE should consider if coil cleaning projects should continue to be incentivized through the custom program pathway. Based on additional research done as part of the evaluation for DNV site ID RICE22N070, an ASHRAE project found that cleaning coils had no significant effect on a unit's cooling capacity and thus energy savings.

Consideration 4: RIE may consider investigating enhanced grocery store programs around continuous maintenance, O&M, and/or natural refrigerants.[8] Based on customer feedback in evaluating DNV site ID RICE22N070, customers would be interested in programs around refrigeration system continual maintenance or for switching to natural refrigerant systems with a lower global warming potential (GWP). It is worth noting that any program centered on GWP may not significantly impact energy savings but would have a greater impact on greenhouse gas reductions.

[8] SMUD Launches Natural Refrigerant Incentive Program — North American Sustainable Refrigeration Council (nasrc.org)

APPENDIX A. SUMMARY OF SAMPLED PROJECTS

The following table summarizes the tracking and evaluation savings estimates, site weights by site, measure, and evaluation type.

Site ID	App	Tracking kWh	Weight	Measure	Market event
RICE22N050	11818303	45,825	12.6	Blower VFD	Retrofit
RICE22S068	12449900	62,102	12.6	DCV, VFD, refrigeration	Retrofit
RICE22N054	11982673	64,315	12.6	EMS	NC
RICE22N070	13249513, 13475450, 13710972, 13741512	187,197	12.6	Refrigeration, O&M	Retrofit
RICE22N095	11655059, 13321748	177,708	3.75	Controls, VFD	Retrofit
RICE22S011	13839869	586,045	3.75	SEMP	Retrofit
RICE22N013	11216625, 11983237, 11413637, 11983247	1,710,954	3.75	Industrial refrigeration	Retrofit
RICE22N093	9397043, 11977866	2,129,265	1	Industrial process	NC
RICE22N087	13815325	56,248	12.6	Refrigeration	Retrofit
RICE22N042	13623376, 13632501	446,170	3.75	Industrial process	NC

APPENDIX B. SITE SAVINGS SUMMARY

		TRACKING DATA				EVALUATED RESULTS				
Site ID	RI Energy application #	Annual energy savings (kWh)	% on-peak savings	Summer on-peak demand savings (kW)	Winter on-peak demand savings (kW)	Annual energy savings (kWh)	% on-peak savings	Summer on-peak demand savings (kW)	Winter on-peak demand savings (kW)	Energy realization rate
RICE22N050	11818303	45,825	0.47	10.2	10.20	46,629	0.76	7.93	10.74	101.75%
RICE22S068	12449900	62,102	0.46	2.75	1.78	31,758	0.46	4.50	3.53	51.14%
RICE22N054	11982673	64,315	0.0	7.34	7.34	32,718	0.28	0.00	0.00	50.87%
RICE22N070	13249513, 13475450, 13710972, 13741512	187,197	0.48	24.8	25.83	155,306	0.47	17.93	24.73	82.96%
RICE22N095	11655059, 13321748	177,708	0.47	9.93	20.65	68,970	0.20	3.54	0.44	38.81%
RICE22S011	13839869	586,045	0.67	190.62	49.84	620,928	0.72	75.67	24.65	105.95%
RICE22N013	11216625, 11983237, 11413637, 11983247	1,710,954	0.51	151.08	212.40	1,269,904	0.44	113.50	154.80	74.22%
RICE22N093	9397043, 11977866	2,129,265	0.47	241.88	324.30	1,458,260	0.52	347.76	147.87	68.49%
RICE22N087	13815325	56,248	0.40	10.82	13.31	60,954	0.40	8.70	9.40	108.37%
RICE22N042	13623376, 13632501	446,170	0.71	78.87	67.07	376,477	0.68	90.45	97.13	84.38%

APPENDIX C. ADJUSTING GROSS REALIZATION RATE STANDARD ERRORS FOR IMPUTED OPERATING ADJUSTMENT

This appendix explains the process for calculating the current and three-year realization rates. The calculation of the Year 1 realization rate is different from the current year (Year 3), or Year 2, as an imputed operational adjustment was necessary for the first year. This section describes the calculation of the current year realization rate, as well as the operational adjustments used for Year 1, which is included in the three-year rolling result.

Basic structure

We have samples for three successive periods: 1, 2, and 3. In this evaluation these samples are 1) PY2020, 2) PY2021, and 3) PY2022. Samples 2 and 3 are full samples with operational adjustments for all sampled sites. Sample 1 had non-operational results for all sites and operational results for only a subset of sites. The three-year realization rate has imputed operational adjustments for the PY2020 results.

Notation

w_j = full-sample weight for sample site j in the Period 3 sample

S_y = population tracked savings of period y

S_T = population tracked savings for all three periods combined

$= S_1 + S_2 + S_3$

q_y = period-y savings as a fraction of the three-period total

$= S_y/S_T$

SW_y = full sample weighted savings represented by "good" sites, i.e., those with operational data for period y

SW_T = full sample weighted savings represented by "good" sites, i.e., those with operational data for all three periods combined

$= SW_1 + SW_2 + SW_3$

f_{g1} = fraction of Period-1 savings represented by "good" sites, i.e., those with operational data

= (full-sample-weighted savings of Period 1 sample sites with operational data)/(total full-sample weighted savings for Period 1)

S_{Tg} = total savings for population represented by sites with operational data, across all samples

$= f_{g1}S_1 + S_2 + S_3$

RR_{oy} = operational-only realization rate for the period y sample

RR_{Ny} = non-operational-only realization rate for the period y sample

RR_{og1} = operational-only realization rate for the population represented by good sites in the Period 1 sample, those with operational data

RR_{ob1} = imputed operational-only realization rate for the population represented by bad sites in the Period 1 sample, those without operational data

SE(X) = standard error of estimate X

RSE(X) = relative standard error of estimate X

=SE(X)/X

Period 1 operational realization rates: RR_{o1}

- For the portion of the population represented by sampled sites with operational adjustments ("good" sites g), RR_{og1} is directly calculated from the sample, using the full sample weights w_j. That is, RR_{og1} is the weighted sum of verified gross savings, divided by the weighted sum of tracked gross savings for that year.
- For sampled sites without operational adjustment ("bad" sites b), RR_{ob1} is imputed as

  $RR_{ob1} = (f_{g\text{-}2}S_{\text{-}2}RR_{o\text{-}2} + f_{g\text{-}1}S_{\text{-}1}RR_{o\text{-}1} + f_{g1}S_1RR_{og1})/S_{(-2,-1,1)g}$ [9]

  That is, all available sites with operational data from a particular year, along with two earlier years, are used to impute the RR for the uncovered portion of the Period 1 and Period 2 populations, with the RR from different periods weighted by the savings it represented. The specific years used to impute ops adjustments where needed for any particular year in the analysis are shown in Table C-1, with the year of the annual result shown horizontally, and the years used to inform the ops adjustments shown vertically. Years marked as "full sample" indicate that no ops adjustments were imputed for that particular year, while years marked as "partial sample" indicate that ops adjustment imputations were needed for some sites. The imputed ops adjustment for Year 1 (2020) is based on ops adjustments from sites evaluated in 2018, 2019, and those sites with ops adjustments available in 2020.

Table C-1. Ops adjustment imputation sources for each annual result

		Annual RR Results						
		2016	2017*	2018^	2019^	2020	2021	2022
Ops Adjustment Sources	2016	Full Sample		-2) Full Sample	-2) Full Sample			
	2017							
	2018			-1) Partial Sample	-1) Partial Sample	-2) Partial Sample		
	2019			1) Partial Sample	1) Partial Sample	-1) Partial Sample		
	2020					1) Partial Sample		
	2021						Full Sample	
	2022							Full Sample

*No evaluation conducted in 2017.
^The 2018 and 2019 evaluations were completed simultaneously and used the same years for ops adjustment imputation.

- Overall Operational Adjustment for Period 1 is calculated as

  $RR_{o1} = f_{g1}\, RR_{og1} + (1\text{-}f_{g1})RR_{ob1}.$

  That is, the operational adjustment for the directly represented portions of the population and the remainder are combined in proportion to their shares of period 1 and period 2 tracked savings respectively. This formula can be expanded as

  $RR_{o1} = f_{g1}\, RR_{og1} + (1\text{-}f_{g1})\,(f_{g\text{-}2}S_{\text{-}2}RR_{o\text{-}2} + f_{g\text{-}1}S_{\text{-}1}RR_{o\text{-}1} + f_{g1}S_1RR_{og1})/S_{(-2,-1,1)g}$

[9] RR_{-2} and RR_{-1} denote two earlier years prior to the current three-year rolling period, which were used as part of the operational adjustments for RR_1.

$= (1 + (1\text{-}f_{g1})\ S_1/S_{(-2,-1,1g)}f_{g1}RR_{og1} + (1\text{-}f_{g1})(S_{-2}/S_{(-2,-1,1g)})RR_{o-2} + (1\text{-}f_{g1})(S_{-1}/S_{(-2,-1,1g)})RR_{o-1})$
$= a_{og1}\ RR_{og1} + a_{-2}RR_{o-2} + a_{-1}RR_{o-1}$,

Where

$a_{og1} = (1 + (1\text{-}f_{g1})\ S_1/S_{(-2,-1,1)g})f_{g1}$
$a_{-2} = (1\text{-}f_{g1})(S_{-2}/S_{(-2,-1,1)g})$
$a_{-1} = (1\text{-}f_{g1})(S_{-1}/S_{(-2,-1,1)g})$

This expansion expresses the overall Period 1 operational realization rate as a weighted average of three independently estimated terms, the directly observed operational realization rate from each period. The factors multiplying the three realization rates have the property that:

$a_{og1} + a_{-2} + a_{-1} = 1$

- Standard error of Period 1 realization rates: The standard error is calculated from the individual standard errors as

  $SE(RR_{o1}) = sqrt[a_{og1}^2\ SE^2(RR_{og1}) + a_{-2}^2\ SE^2(RR_{o-2}) + a_{-1}^2\ SE^2(RR_{o-1})]$

  This is true because the three RRs at step 3 are from independent samples.

Periods 2 and 3 combined RR

The operational and non-operational realization rates RR_{N2}, RR_{N3} and RR_{O2}, RR_{O3} are calculated from the full sample using the full sample weights and the non-operational and operational adjusted savings for the sample, via the usual formulas.

The Overall RR is the product of the operational and non-operational RRs

$RR_2 = RR_{o2}\ RR_{N2}$

and

$RR_3 = RR_{o3}\ RR_{N3}$

Standard error: First calculate the relative standard errors:

$RSE(RR2) = sqrt[RSE^2(RR_{o2}) + RSE^2(RR_{N2})]$

and

$RSE(RR_3) = sqrt[RSE^2(RR_{o3}) + RSE^2(RR_{N3})]$

This formula is approximately correct, assuming that even though RR_{N2}, RR_{N3} and RR_{O2}, RR_{O3} are from a common sample, they are essentially unrelated so can be treated as independent.

The standard errors are then calculated from the RSEs.

$SE(RR_2) = RR_2\, RSE(RR_2)$

and

$SE(RR_3) = RR_3\, RSE(RR_3)$

Three-year combined RR

RR calculation

The three-year RR is the savings-weighted average of the three separately estimated RRs:

$RR_{1\text{-}3} = (S_1RR_1 + S_2RR_2 + S_3RR_3)/S_T$
$= q_1RR_1 + q_2RR_2 + q_3RR_3$

This calculation produces an overall realization rate for each period, then combines these across periods. This approach is the natural one, combining the historical overall results with the most recent, consistent with our general method for three-year rolling realization rate calculation.

SE calculation

While the first term, RR_1, is determined in part from the operational portions of other years, since RR_1 is not use for any adjustment to RR_2 or RR_3, and since the program year results used to make operational adjustments to RR_1 are no longer included in the three-year rolling period, the three years may be treated as independent estimates to calculate standard errors. This is a change from the prior three program years where the RRs could not be treated as independent, because at least one year's RR contained imputed operational adjustments from at least one other year in the three-year rolling period. This change allows us to use a simplified SE calculation as compared to the prior three reporting cycles.

The standard error for the three-year rolling period is calculated as:

$SE(RR_{N1\text{-}3}) = \mathrm{sqrt}[q_1^2\, SE^2(RR_{N1}) + q_2^2\, SE^2(RR_{N2}) + q_3^2\, SE^2(RR_{N3})]$

Calculating Period 3 and three-period realization rates

The formulas for calculating the Period 3 operational realization rate RR_{o3}, the Period 3 overall realization rate RR_3, and the preferred three-period overall realization rate $RR_{1\text{-}3}$ are applied separately for each reporting category of realization rate. Typically, each reporting category includes sample points from multiple sampling cells.

APPENDIX D. LIFETIME SAVINGS ADJUSTMENT FACTORS (LSAFS) METHODOLOGY

Evaluation lifetime savings findings should be captured in a lifetime savings adjustment factor (LSAF), which is applied to the tracking **measure life** in the BC Tool used to report PA evaluated savings in the Annual Report. The LSAF is intended to account for the following evaluation findings:

1. Incorrect applicant effective useful life (EUL) measure life assumptions
2. Reduced life from equipment removed after a year or more of operation
3. Change in measure application type impacting measure life
4. Change in measure application type impacting dual versus single baseline status[10]
5. Incorrect applicant outyear factor (OYF) assumption

First-year saving realization rate. As a starting point, the annual savings realization rate is calculated as the weighted sample verified annual savings divided by the weighted sample tracked savings.

$$RR\% = \frac{\sum w_i \times FYS_i^{Evaluated}}{\sum w_i \times FYS_i^{Tracking}}$$

where:

$RR\%$ = first-year savings realization rate

w_i = site weight

$FYS_i^{Evaluated}$ = site evaluated first-year savings (kWh)

$FYS_i^{Tracking}$ = site tracking first-year savings (kWh)

Measure-level lifetime savings. For each evaluated measure, the evaluators calculated an evaluated lifetime savings using the following formula:

$$LS_{Savings} = FYS_{Evaluated} \times [RUL_{Evaluated} + \mathrm{OYF} \times (EUL_{Evaluated} - RUL_{Evaluated})]$$

where:

$LS_{Savings}$ = evaluated lifetime savings (kWh)

$FYS_{Evaluated}$ = evaluated first year savings (kWh)

$EUL_{Evaluated}$ = evaluated measure life (years in decimal form) Reflects revisions to measure life due to alignments with eTRM measure lives or other adjustments or to account for equipment removal after one year.

$RUL_{Evaluated}$ = 1/3 of $EUL_{Evaluated}$ (years)

OYF = 100% for single-baseline measures. 90% for non-lighting dual-baseline measures.

10 For non-lighting measures only. The LSAF published for lighting measures does not incorporate the impacts of dual baseline as the PAs at the time did not have the ability in their BCR models to track dual baseline. These dual baseline impacts are covered when applying AMLs published through the LMC study for PAs that have been able to adjust tracking measure lives to use the AMLs, and through the LMC adjustment factor discussed later in this section for PAs that have not been able to make that adjustment, or only partially did.

Program lifetime savings realization rate (LSRR%). The LSRR is calculated in similar fashion to the annual savings RR. To calculate LSRR, the weighted evaluated lifetime savings is divided by the weighted tracked lifetime savings. The team calculated LSRR using the following formula:

$$LSRR\% = \frac{\sum w_i \times LS_i^{Evaluated}}{\sum w_i \times FYS_i^{Tracking} \times EUL_i^{Tracking}}$$

where:

$LSRR\%$	= program lifetime savings realization rate
w_i	= site weight
$LS_i^{Evaluated}$	= site evaluated lifetime savings (kWh)
$FYS_i^{Tracking}$	= site tracking first-year savings (kWh)
$EUL_i^{Tracking}$	= tracking measure life

Program LSAF. The LSAF accounts for differences noted in items 1 to 5 above and the different distribution of savings for both first-year and lifetime savings at sites included in the sample. To avoid double counting the impacts of both the FYS RR and the LS RR, we need to calculate both RRs. The LSAF can now be backed out by calculating the ratio of the lifetime savings RR over the first-year savings RR.

$$LSAF = \frac{LSRR\%}{RR\%}$$

where:

$LSAF$	= lifetime savings adjustment factor
$RR\%$	= program first-year savings realization rate
$LSRR\%$	= program lifetime savings realization rate

The program-level LSAF can be used by PAs for reporting lifetime savings and will incrementally impact the lifetime savings after the annual savings realization rate (RR) is applied. To calculate lifetime adjusted gross savings (LAGI), PAs will use the following formula:

$$LAGI = (Annual\ Gross\ Savings_{Tracking} \times Annual\ RR\%) \times (Measure\ life_{Tracking} \times LSAF)$$

where:

$LAGI$	= lifetime adjusted gross impact savings (kWh)
$Annual\ gross\ savings_{Tracking}$	= tracking annual gross savings (kWh)
$Measure\ life_{Tracking}$	= tracking measure life (years)
$RR\%$	= program realization rate
$LSAF$	= lifetime savings adjustment factor

The BC Model requires as input PA gross annual tracking savings and tracking measure life and does not accept as input tracking lifetime savings. The tracking measure life reflects project level applicant effective useful measure life selections and in the future dual baseline effects. The BC Model specifies evaluation factors that are required to report evaluated savings. Due to the calculation methods employed by the BC Model, the LSAF will be applied to tracking measure life.

APPENDIX E. SITE REPORTS

Final site reports can be found on the following pages.

RHODE ISLAND CUSTOM ELECTRIC SITE-SPECIFIC REPORT
SITE ID: RICE22N013

Report Date: April 16, 2024

Program Administrator	Rhode Island Energy
Application ID(s)	11216625, 11413637, 11983237, 11983247
Project Type	C&I Existing Building Retrofit
Evaluation Type	Ops
Program Year	PY2022
Evaluation Firm	DNV
Evaluation Engineer	Joe St. John
Senior Engineer	Olav Hegland

1 EVALUATED SITE SUMMARY AND RESULTS

This facility completed two projects installed at a new, 200,000 ft^2 meat packing facility. The first project involved installing (2) 177 HP VFD air compressors rather than the less efficient industry standard practice load/no-load air compressors of a similar size. The VFD air compressor project accounts for 22% of the total tracking savings for the site. The second project, accounting for 78% of the claimed tracking savings, involved (3) energy efficiency measures associated with their ~1,570-ton ammonia refrigeration system. Table 1-1 shows the (4) PA ID numbers and the total claimed savings for each measure from the tracking database.

Table 1-1. Measure list

Identifier	Project ID	kWh savings	% of total project savings	Measure description
1	11216625 (Parent) 11983237 (Child)	384,100	22%	New construction frozen meat packing facility. Installed (2) 177 HP VFD Air Compressors and a cycling blower purge dryer instead of (2) load / no-load compressors and a non-cycling heatless desiccant dryer.
2	11413637 (Parent) 11983247 (Child)	954,543	56%	65˚F minimum condensing setpoint, rather than 85˚F minimum condensing setpoint for new ammonia refrigeration system.
3		169,410	10%	VFD control for evaporative condenser fans, rather than two-speed fan control.
4		202,901	12%	Economized high-stage compressors with VFD control for lead compressor, rather than non-economized high stage compressors with slide-valve for lead compressor.
Total		**1,710,954**	**100%**	

During the initial interview with the site contact, evaluators learned the following:

- The site contact is present on-site and agreed to accommodate an on-site evaluation.
- It is safe to visit the facility and inspect the measure.

After reviewing the tracking files and information gathered during the site visit, the evaluator classified this measure as a new construction project with industry standard practice as baseline. The evaluation results are presented in Table 1-2 and have a realization rate of 77.5% while the second set of evaluation results are presented in Table 1-3 and have a realization rate of 39.1%. The second set are based on a mini-ISP done when it was thought the baseline used to calculate savings did not have a clear rationale. The primary evaluation results refer to the industry standard practice baselines referenced in the tracking calculations, which refer to the 2014/15 National Grid Baseline Document[1] and was further documented in an ex-ante review of using the 2014/15 Baseline Document which was older at the time of this project's installation, but no other document existed at the time.

The evaluation results in Table 1-2 are based on the ISP at the time, shall be used in the expansion analysis and final report provided that Rhode Island Energy updates the 2014 National Grid Baseline document for the affected refrigeration measures based on the findings described in Appendix A and refers to those baseline practices for any similar projects going forward. The main findings from Appendix A is that the evaluators found the industry standard practice baseline for measure 2 in Table 1-1 to be a 70˚ F minimum condensing temperature rather than 85˚ that was referenced in the 2014 National Grid Baseline Document, and the evaluators found the industry standard practice baseline for measure 3 to be VFD fan control rather than the two-speed fan control used in the baseline for the tracking calculations.

[1] 2014-baseline-document-for-ma-and-ri.pdf (nationalgridus.com)

Table 1-2. Evaluation results summary

PA application ID	Measure name		Annual electric energy (kWh)	% of energy savings on-peak	Summer on-peak demand (kW)	Winter on-peak demand (kW)
11216625 / 11983237	(2) 177 HP VFD Air Compressors	Tracked	384,100	49.0%	49.10	44.40
		Evaluated	408,211	44.3%	45.7	47.3
		Realization Rate	106.3%	90.3%	93.0%	106.5%
11413637 / 11983247	65˚ minimum condensing setpoint rather than 85˚ minimum condensing setpoint	Tracked	910,249	50.55%	70.0	115.3
		Evaluated	648,193	44.2%	82.1	49.8
		Realization Rate	71.2%	86.6%	117.3%	43.2%
	VFD control for evaporative condenser, rather than two-speed fan control.	Tracked	140,482	50.55%	10.8	17.8
		Evaluated	101,428	44.2%	12.8	7.8
		Realization Rate	72.2%	86.6%	118.9%	43.8%
	Economized high-stage compressors with VFD control for lead ammonia compressor	Tracked	276,124	50.55%	21.2	35.0
		Evaluated	112,071	44.2%	14.2	8.6
		Realization Rate	40.6%	86.6%	66.9%	24.6%
Total		**Tracked**	**1,710,954**	**50.55%**	151.1	212.4
		Evaluated	**1,269,904**	**44.2%**	113.5	154.8
		Realization Rate	**74.2%**	**87.4%**	75.1%	72.9%

Table 1-3. Secondary Evaluation results summary

PA application ID	Measure name		Annual electric energy (kWh)	% of energy savings on-peak	Summer on-peak demand (kW)	Winter on-peak demand (kW)
11216625 / 11983237	(2) 177 HP VFD Air Compressors	Tracked	384,100	49.0%	49.1	44.4
		Evaluated	408,211	44.3%	45.7	47.3
		Realization Rate	106.3%	90.3%	106.5%	93.0%
11413637 / 11983247	65˚ minimum condensing setpoint rather than 85˚ minimum condensing setpoint	Tracked	910,249	50.55%	70.0	115.3
		Evaluated	18,118	44.5%	2.3	1.4
		Realization Rate	2.0%	87.2%	2.0%	2.0%
	VFD control for evaporative condenser, rather than two-speed fan control.	Tracked	140,482	50.55%	10.8	17.8
		Evaluated	-	-	-	-
		Realization Rate	0.0%	0.0%	0.0%	0.0%
	Economized high-stage compressors with VFD control for lead ammonia compressor	Tracked	276,124	50.55%	21.2	35.0
		Evaluated	112,071	44.5%	14.2	8.6
		Realization Rate	40.6%	87.2%	40.6%	40.6%
Total		**Tracked**	**1,710,954**	**50.55%**	151.1	212.4
		Evaluated	**538,400**	**44.3%**	62.1	57.3
		Realization Rate	**31.5%**	**87.7%**	41.1%	27.0%

1.1 Explanation of deviations from tracking

[11216625 / 11983237] (2) 177 HP VFD Air Compressors

The evaluated energy savings are higher than the tracking energy savings because the evaluator found that the lead compressor operated 8,735 hours/year which is 25% more than the 6,972 hours estimated in the tracking analysis. However, the average change in the kW/CFM between the baseline and post-case was found to be 0.021 kW/CFM, compared to 0.044 estimated by the tracking analysis. So, while the increased operating hours resulted in an increase to the tracking energy savings, the effect of the change in the kW/CFM tempered this increase. Just these two factors would suggest the realization rate would be 1.25 x 0.48 = 60% rather than the 106% shown. However, this would be assuming that the baseline and post-case CFM in the tracking and evaluator calculations are the same, when they are not due to the baseline system using about twice as much air in the baseline case compared to the post-case, because the baseline dryer requires 17% of the airflow demand, whereas the post-case dryer requires 1.6% of the airflow demand. The change in the kW/CFM is driven by the amount of time the compressor spends at different loads which fall at different parts of the of compressor efficiency curve.

[11413637 / 11983247] Ammonia Refrigeration Measures

65˚ minimum condensing setpoint, rather than 85˚ minimum condensing setpoint

The realization rate for this measure is 71.2%. This is primarily due to the finding that the average post-case condensing temperature for times when a low condensing temperature would be appropriate was found to be 71˚F, rather than the 65˚used in the tracking calculations. Additionally, the tracking calculations estimated 6,010,000 annual ton-hours, while the evaluator found approximately 5,570,000 annual ton hours. The prospective realization rate was based on the 85˚F baseline condensing temperature which is based on the 2014 National Grid Baseline Document[2].

The secondary realization rate for this measure was found to be 2.0%. This is because the evaluator found that the industry standard practice around the time of the tracking analysis was done, after interviewing seven industrial refrigeration experts, was found to be a minimum condensing temperature of 70˚F, rather than 85˚used in the tracking calculations and referred to in the 2014 National Grid Baseline Document.

This project was installed in 2020, while the evaluators contacted interviewees in 2024 about industry standard practices. For this reason, Interviewees were asked how standard practices have changed over the past 0-5 and 5-10 years, and the responses indicate standard practice has not changed between 2020, when this project was completed, and now, 2024. Further details can be found in Appendix A.

VFD control for evaporative condenser, rather than two-speed fan control

The realization rate for this measure is 72.2%. The this is primarily due to the evaluation finding that the average condenser fan kW reduced from 24 kW to 12 kW with the use of VFDs rather than two-speed fan control, whereas the tracking calculations estimated that the fan kW would reduce from 35.3 kW to 19.3 kW. The evaluator calculations were based on measured post-case condenser fan kW data, whereas the tracking calculations, which were necessarily estimates, over-estimated the condenser load.

The secondary realization rate for this measure is 0%. This is because the evaluator found that the industry standard practice, after interviewing seven industrial refrigeration experts, was to install VFDs on evaporative condenser fans in new construction large ammonia refrigeration systems, whereas the tracking calculations, based upon, and reviewed by the industry professionals involved in developing and reviewing the tracking energy savings, stated that the industry standard practice would be for standard practice to use two-speed condenser fans. Further details can be found in Appendix A.

Economized high-stage compressors with VFD control for lead ammonia compressor

For this measure, there were no difference between the different realization rates because there was no consensus among the 7 industrial refrigeration experts on the industry standard practice for this measure as there was for the previous two measures, and so the evaluator accepted the standard practice baseline used in the tracking calculations.

The realization rate for this measure was 40.6%. This is primarily due to the evaluator finding that the efficiency improvement between the baseline and post-case for this measure went from 0.267 kW/ton for the baseline compressors to 0.258 kW/ton for the efficient compressors, for an improvement of 0.009 kW/ton, whereas the tracking savings estimated that the compressor efficiency would improve from 0.275 kW/ton for the baseline to 0.256 kW/ton for the efficient compressor, an improvement of 0.020 kW/ton. The change in the kW/ton is driven by the amount of time the refrigerant compressors spend at different loads which fall at different parts of the of compressor efficiency curves. Additionally, the tracking calculations estimated 6,010,000 annual ton-hours, while the evaluator found approximately 5,570,000 annual ton hours.

[2] 2014-baseline-document-for-ma-and-ri.pdf (nationalgridus.com)

Further details regarding deviations from the tracked savings are presented in Section 0.

1.2 Recommendations for program designers and implementers

The evaluator recommends that Rhode Island Energy updates the 2014 National Grid Baseline document for the affected refrigeration measures based on the findings described in Appendix A and refers to those baseline practices for any similar projects going forward. Namely, for large industrial ammonia refrigeration systems, the updated industrial standard practice baselines are as follows:

- 70˚F minimum condensing temperature
- VFDs on evaporatively cooled condenser fans

1.3 Customer alert

There is no relevant customer alert.

2 EVALUATED MEASURES

This section presents the evaluation procedure, including the findings from an in-depth review of the supplied applicant calculations and the evaluation methodology determined to be the best fit for the site and the information available.

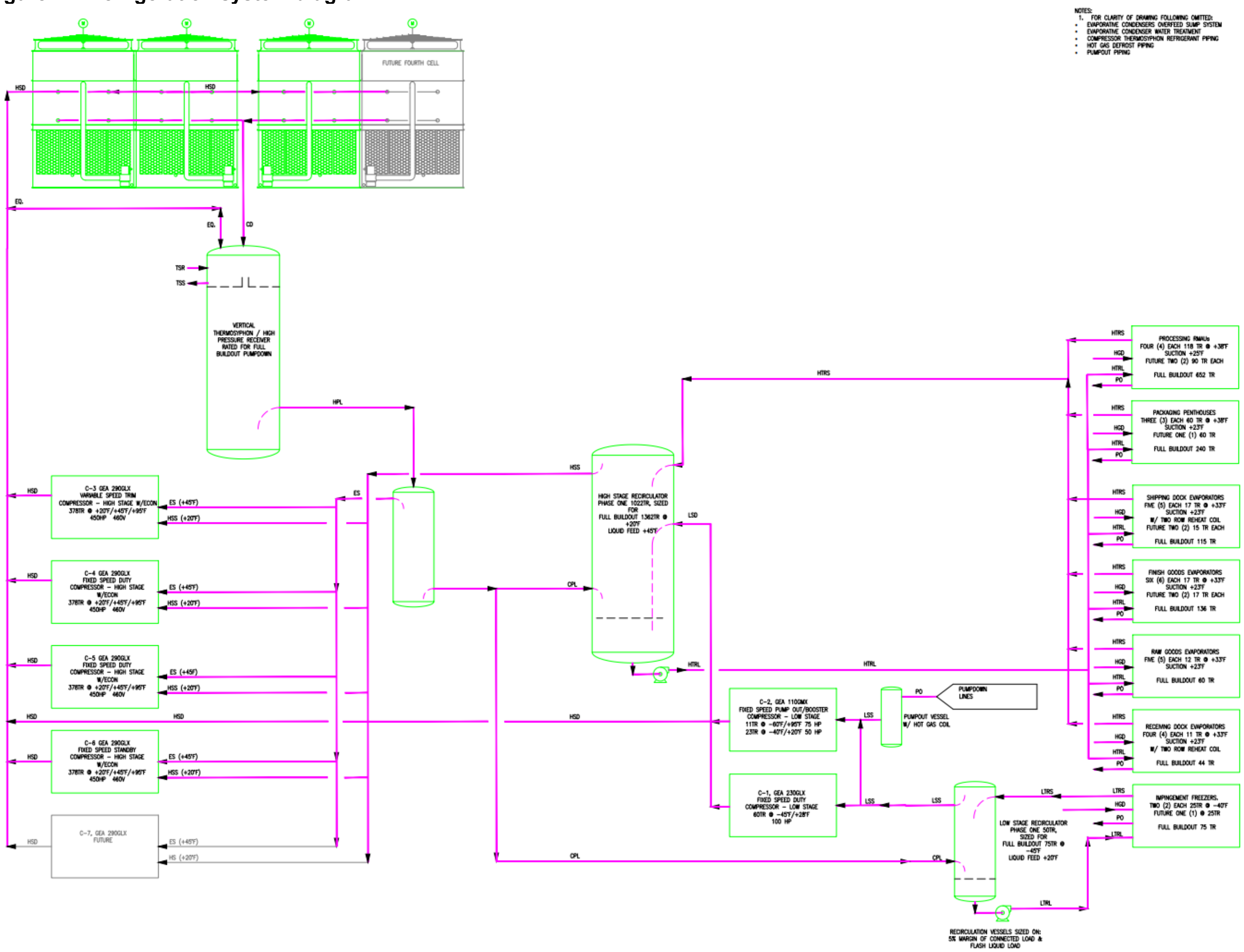
The evaluated measure for this site is summarized in Table 2-1.

Table 2-1. Evaluated measure

Identifier	Project ID	kWh savings	% of total project savings	Measure description
1	11216625 (Parent) 11983237 (Child)	384,100	22%	New construction frozen meat packing facility. Installed (2) 177 HP VFD Air Compressors and a cycling blower purge dryer instead of (2) load / no-load compressors and a non-cycling heatless desiccant dryer.
2	11413637 (Parent) 11983247 (Child)	954,543	56%	65˚ minimum condensing setpoint, rather than 85˚ minimum condensing setpoint for new ammonia refrigeration system.
3		169,410	10%	VFD control for evaporative condenser, rather than two-speed fan control.
4		202,901	12%	Economized high-stage compressors with VFD control for lead compressor, rather than non-economized high stage compressors with slide-valve for lead compressor.
Total		**1,710,954**	**100%**	

A diagram of the refrigeration system is shown in **Figure 2-1**

Figure 2-1.Refrigeration system diagram



2.1 Applicant energy savings algorithm and applicant key parameters

This section describes the applicant's application information, savings methodology, and the evaluation assessment of the savings calculation algorithm used by the applicant.

2.1.1 Applicant energy savings algorithm and applicant key parameters

[11216625 / 11983237] (2) 177 HP VFD Air Compressors

The tracking calculations for the air compressor measure began with metered Amp data that was measured between 12/14/22 and 12/22/22 at 1-minute intervals and then converted to CFM data. The conversion from Amp data to CFM data was not shown in the tracking calculations. This CFM data was then converted to kW data using the following formula:

$$kW_{Baseline} = A \times CFM^3 + B \times CFM^2 + C \times CFM + D$$

Where the baseline constants are:

Variable	Value
A	0.0463
B	-0.5962
C	1.2831
D	0.2686
CFM_{min}	0
CFM_{max}	856

The constants are based on a figure in the Compressed Air Challenge handbook that relates % capacity to % kW for different storage volumes. These correspond to a storage volume of 3 gallons/CFM.

When the measured CFM exceeded the capacity of one compressor, the first compressor was assigned 856 CFM, and the second compressor was assigned the remaining CFM. The coefficients were developed based on the CAGI specification sheet for a 175 HP load/no load screw compressor made by Atlas Copco, the G 132-100. This CAGI sheet was adjusted using the percent capacity vs. percent kW input curves from the Compressed Air Challenge handbook, which provides a different curve depending on the storage of the system (based on gal/cfm). The coefficients come from the curve for 3 gal/cfm, then adjusted to the CAGI sheet for the selected baseline compressor. The calculations show that the operating pressure is 100 psi, so no pressure adjustment was made to the kW vs. CFM curve.

The post-case compressor kW for this measurement period was calculated using the following formula:

$$kW_{Post} = A \times CFM^3 + B \times CFM^2 + C \times CFM + D$$

here the post constants are:

Variable	Value
A	0
B	0
C	0.1269
D	5.7784
CFM_{min}	0
CFM_{max}	998

The above coefficients come from a curve fit of 5 points on a CAGI specification sheet for the installed GA 132 VSD+ screw compressor. The correlation coefficient (R^2) for the curve fit is 1.0. The calculations show that the operating pressure is 100 psi, so no pressure adjustment was made to the kW vs. CFM curve.

From this 1-week period where the baseline and post-case kW was calculated from the measured CFM data, a weekly load profile was developed, as shown in the figure below.

Average Hourly Data (CFM, Base/Existing Demand, Proposed Demand)

Hour	Mon			Tue			Wed			Thu			Fri			Sat			Sun		
	CFM	Base kW	Prop kW	CFM	Base kW	Prop kW	CFM	Base kW	Prop kW	CFM	Base kW	Prop kW	CFM	Base kW	Prop kW	CFM	Base kW	Prop kW	CFM	Base kW	Prop kW
0	274	102.2	58.0	207	92.3	50.8	366	126.7	75.3	217	90.2	50.2	297	117.9	64.6	163	69.0	38.3	415	133.2	83.6
1	117	63.2	31.8	321	113.9	67.8	219	90.4	50.9	187	81.8	44.9	225	103.0	53.5	174	76.6	41.9	289	113.2	64.2
2	144	72.5	36.9	351	119.7	72.9	163	75.4	41.0	299	108.9	64.0	189	81.3	44.5	218	91.8	51.2	186	91.0	47.3
3	162	83.8	41.8	240	100.6	56.0	234	94.9	53.8	219	87.8	49.7	248	98.1	55.8	344	123.0	72.4	323	124.8	70.0
4	348	127.1	73.0	215	91.6	51.0	228	86.3	49.8	157	73.3	39.1	341	117.7	70.4	183	78.9	43.9	200	97.1	49.8
5	197	98.7	50.7	354	127.4	74.1	131	62.5	33.8	158	76.9	40.2	166	77.1	41.9	138	68.2	37.0	189	94.8	47.4
6	243	114.3	59.1	273	109.7	62.4	224	92.5	51.9	329	118.1	70.3	259	104.0	59.1	369	118.8	75.1	255	110.5	59.0
7	458	137.9	89.8	335	126.9	73.3	282	110.3	62.2	419	129.9	83.7	416	133.7	84.0	379	130.2	78.8	352	120.5	71.9
8	309	122.3	67.3	311	120.2	68.6	417	133.5	83.5	266	108.6	60.7	295	113.4	65.3	297	118.9	66.3	213	105.4	52.8
9	356	127.6	74.5	485	144.7	94.7	302	120.4	69.1	329	118.4	70.8	285	110.9	63.8	325	125.1	70.0	219	92.6	50.3
10	426	135.6	84.5	410	135.7	82.3	279	110.4	63.4	413	130.2	83.2	301	119.2	67.5	412	138.5	83.0	300	117.9	65.6
11	452	135.4	88.4	307	119.4	68.4	451	135.5	88.5	280	109.5	62.8	361	123.4	73.6	460	138.7	90.4	177	83.7	43.5
12	318	117.3	68.2	438	130.8	86.1	270	106.3	60.3	291	111.3	64.8	371	124.8	76.4	293	120.3	66.0	200	97.1	50.7
13	362	128.8	77.0	343	128.1	73.0	324	116.6	68.8	343	123.2	71.4	299	112.5	64.8	375	128.2	77.4	334	122.1	71.0
14	443	134.2	87.3	324	122.7	70.7	303	111.7	65.2	504	144.4	97.0	357	121.7	74.3	413	129.6	82.5	270	115.1	62.9
15	290	116.4	65.3	311	119.5	69.0	416	135.0	83.3	340	123.6	73.0	390	127.4	79.9	290	118.7	64.3	166	89.0	44.9
16	300	117.7	66.0	320	118.2	68.8	411	129.4	82.3	235	95.2	53.4	293	110.3	63.4	274	113.9	61.9	301	116.4	64.8
17	249	103.3	57.2	335	118.6	70.3	230	95.6	53.6	385	131.4	78.8	227	96.0	53.0	292	116.9	65.4	158	70.3	37.6
18	359	124.9	74.7	213	93.3	51.2	251	98.1	56.1	219	88.8	50.0	214	88.3	49.5	372	128.4	76.9	118	66.6	32.7
19	265	106.1	60.0	223	92.5	51.3	344	122.3	71.9	207	89.2	49.4	308	112.9	66.3	224	100.9	54.3	135	74.8	36.8
20	195	88.1	48.1	336	120.9	70.3	186	84.2	45.7	224	89.1	50.8	305	109.4	64.8	294	110.8	62.2	244	102.9	55.7
21	326	118.2	68.9	186	80.8	44.5	207	88.5	48.5	361	127.4	75.1	214	89.7	50.4	296	110.5	63.9	291	110.4	62.8
22	225	91.0	51.0	154	69.7	38.1	187	80.9	44.5	327	121.2	69.9	254	99.9	57.0	223	100.9	53.7	161	80.5	41.2
23	178	78.5	43.4	181	82.0	45.2	365	127.2	75.7	164	94.0	45.6	255	92.0	53.8	267	105.0	58.5	149	73.0	37.5
Avg.	**291**	**110.2**	**63.4**	**299**	**111.6**	**65.0**	**283**	**105.6**	**61.6**	**286**	**107.2**	**62.4**	**286**	**107.7**	**62.4**	**295**	**110.9**	**64.0**	**235**	**100.1**	**54.3**

The baseline and post-case kW data was then summed over 50 weeks (assuming 2 weeks for shutdown), to estimate baseline energy use 904,029 kWh, post energy use of 519,929, and annual energy savings of 384,100 kWh.

[11413637 / 11983247] Ammonia Refrigeration Measures

65˚ minimum condensing setpoint, rather than 85˚ minimum condensing setpoint

The baseline energy for this measure was calculated using the following formula, which is the sum of kW for the booster compressors, and the high-stage (HS) compressors 1-4, as well as the condenser fan and pump kW. The kWh for each of these components is calculated using a bin analysis, using the local weather station, with outdoor air temperature bins ranging from 2.5˚ F to 97.5˚ F, at 5˚ temperature bins.

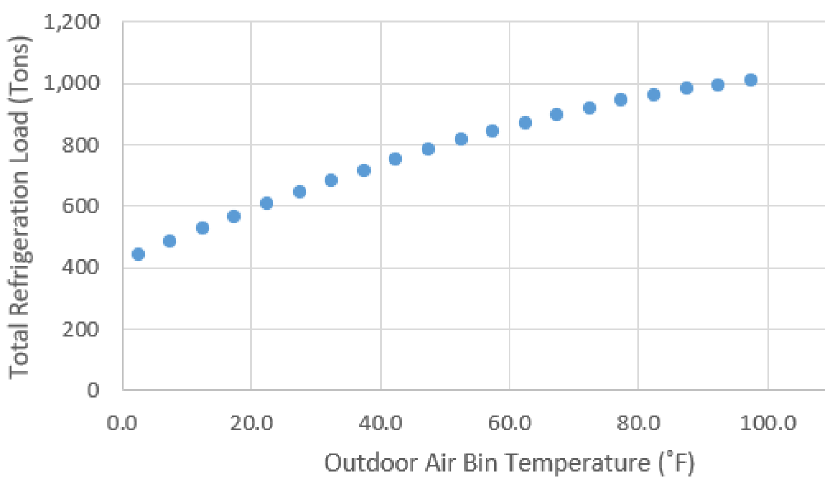
$$\begin{aligned} Baseline\ kW &= Booster\ Compressor\ kW + HS\ Compr\ 1\ kW + HS\ Comp\ 2\ kW + HS\ Comp\ 3\ kW + HS\ Comp\ 4\ kW \\ &\quad + Cond\ Fan\&Pump\ kW \end{aligned}$$

The spreadsheet which calculates the final savings shows the following compressor loading profile for the (6) ammonia compressors at the facility, based on outdoor air temperature. The spreadsheet does not show how the tonnage values were calculated, but instead, the tonnage values are hard coded. The total tonnage ranges from 1,008 tons at the hottest outdoor air temperature and goes down to 441 tons at the lowest outdoor air temperature.

OADB	OAWB	Condensing Temp	C-1 Booster Compressor Tons	C-2 Pump Down Compressor Tons	C-3 High Stage Compressor #1 with Slide Valve	C-4 High Stage Compressor #2 with Slide Valve	C-5 High Stage Compressor #3 with Slide Valve	C-6 High Stage Compressor #4 with Slide Valve	Total Tons
97.5	71.0	88.0	119	0	371	345	173	0	1,008

OADB	OAWB	Condensing Temp	C-1 Booster Compressor Tons	C-2 Pump Down Compressor Tons	C-3 High Stage Compressor #1 with Slide Valve	C-4 High Stage Compressor #2 with Slide Valve	C-5 High Stage Compressor #3 with Slide Valve	C-6 High Stage Compressor #4 with Slide Valve	Total Tons
92.5	74.1	91.1	109	0	373	342	171	0	995
87.5	72.7	89.7	99	0	366	344	172	0	980
82.5	69.6	86.6	89	0	353	347	173	0	962
77.5	68.0	85.0	80	0	339	349	174	0	942
72.5	65.8	85.0	72	0	325	349	174	0	920
67.5	62.6	85.0	65	0	308	349	174	0	896
62.5	57.2	85.0	58	0	289	349	174	0	871
57.5	52.1	85.0	52	0	268	349	174	0	843
52.5	47.6	85.0	47	0	244	349	174	0	814
47.5	43.2	85.0	43	0	349	349	44	0	783
42.5	38.8	85.0	39	0	349	349	15	0	751
37.5	33.9	85.0	35	0	349	333	0	0	717
32.5	29.2	85.0	33	0	349	300	0	0	682
27.5	24.3	85.0	31	0	349	265	0	0	645
22.5	19.7	85.0	30	0	349	228	0	0	607
17.5	15.1	85.0	29	0	364	174	0	0	567
12.5	10.2	85.0	30	0	323	174	0	0	526
7.5	5.8	85.0	30	0	280	174	0	0	484
2.5	2.1	85.0	32	0	234	174	0	0	441

The figure below shows the total tonnage for all the compressors, for all the weather bins from the table above:



The documentation states that the tonnage was calculated from measured post-case amp data on each of the installed compressors, along with measured discharge pressure, and suction pressure data. The Amp data was first converted to kW data using an assumed power factor, and the discharge pressure and suction pressure data were converted to saturated discharge temperature and saturated condensing temperature data using R-717 properties. These temperatures were in Rankine and used in the following equations which were developed from manufacturer's data, to get the tonnage for each compressor during the metering period which occurred from July 1, 2022, to December 31, 2022. However, the calculations for converting this trend data to load (tonnage) data was not provided.

For the high-side compressors with slide valve unloading

tons= c0 + c1 * kW+c2*kW*SST+c3*kW*SCT+c4*SST/SCT

c0: -118.637618
c1: -131.488669
c2: -0.25980623
c3: 0.220078065
c4: 156.8735862
R^2: 0.998262086

For the high-side compressor with VFD control

tons= c0 + c1 * kW+c2*kW*SST+c3*kW*SCT+c4*SST/SCT

c0: -4.3295E+00
c1: -1.0794E+02
c2: -2.1083E-01
c3: 1.8028E-01
c4: 1.2752E+02
R^2: 0.998115473

For the booster compressor with slide valve unloading

tons= c0 + c1 * kW+c2*kW*SST+c3*kW*SCT+c4*SST/SCT

c0: -3.5459E+01
c1: -8.5965E+01
c2: -1.9184E-01
c3: 1.6529E-01
c4: 1.0134E+02
R^2: 0.997917235

With the tonnage data calculated, the baseline kW data for the compressors was calculated as follows:

1.) The full load bhp/ton values were calculated with the following equation, which was developed based on manufacturer's data:

$$\frac{BHP}{ton} = c_0 + c_1 \times T_{evap(R)} + c_2 \times T_{cond(R)} + c_3 \times T_{evap(R)} \times T_{cond(R)}$$

Where,

For the booster compressor (C-1):

c0:	-56.02421875
c1:	0.115867187
c2:	0.15384375
c3:	-0.000323437

Suction temperature = -45˚F
Condensing temperature= 20˚F

For the high-stage compressors with slide valve control (C-3, C-4, C-5, and C-6):

c0:	-33.17333333

c1:	0.054
c2:	0.079
c3:	-0.000133333

Suction temperature = 20˚ F
Condensing temperature = IF(OAWB+TD_setpoint>85˚F,OAWB+TD_setpoint,85˚F)
TD_setpoint = 17˚ F.

The equation for the condensing temperature for the high-stage compressors says that the condensing temperature is the minimum of 85˚ F, and the outdoor air wet-bulb temperature plus the temperature delta (TD) setpoint of 17˚ F.

2.) The % capacity is calculated for each of the compressors, by dividing the tonnage (as shown in the table above) by the rated nameplate tonnage.

3.) The adjustment to the full load BHP/ton, to account for the compressors being more efficient at reduced load, called %BHP/ton, is calculated using the following formula:

$$\% \frac{BHP}{ton} = c_0 + c_1 \times (1 - \%Capacity)$$

Where,

For the booster compressor (C-1):

c0:	1
c1:	0.628753211
c2	0
c3	0

For the high-stage compressors with slide valve control (C-3, C-4, C-5, and C-6):

c0:	1
c1:	0.391831332
c2	0
c3	0

The kW for each temperature bin is then calculated using the following equation:

$$kW = Tons \times \frac{BHP}{Ton} \times \% \frac{BHP}{Ton} \times 0.746 \frac{kW}{HP} \times \frac{1}{Motor\ Efficiency}$$

Where the motor efficiency is 93.6% for C-1, the booster compressor, and is 95.8% for the high stage compressors with slide valve control (C-3, C-4, C-5, and C-6).

The kW is calculated for each compressor, for each weather bin, and multiplied by the number of hours in each weather bin, and then summed to determine the total baseline kWh for the compressors. The refrigeration system runs 24/7/365.

The energy for the baseline evaporative condenser, made up of (3) 50 HP 2-speed fans, and (3) 7.5 pumps that move the water for the evaporative condenser, is calculated using the following steps:

1.) The available capacity (in Btu/h) of the condenser at each weather bin is calculated using the following equation:

$$Available\ Capacity = \frac{Full\ Load\ Rated\ Capacity}{(Condensing\ Temp - Wet\ Bulb)} \times (Condensing\ Temperature - Outdoor\ Air\ Wetbulb\ Temperature)$$

The full load rated capacity is 969,992 Btu/hr/˚F, which means that the capacity is 969,992 Btu/hr for each ˚F temperature difference between the condensing temperature and the outdoor air wet-bulb temperature. The condensing temperature is the minimum of 85˚ F, and the outdoor air wet-bulb temperature plus the temperature delta (TD) setpoint of 17˚ F.

2.) The heat rejection load on the evaporative condenser (in Btu/hr) at each bin is calculated using the following equation:

$$\begin{aligned} &Heat\ Rejection\ Load \\ &\quad = Total\ Tonnage\ On\ the\ Refrigeration\ Compressors\ \times 12{,}000 \frac{Btuh}{ton \cdot hr} \\ &\quad + Total\ BHP\ of\ the\ Compressors\ \times 2{,}545 \frac{Btuh}{BHP \cdot hr} \end{aligned}$$

3.) The percent utilized capacity of the condenser at each weather bin is calculated using the following equation:

$$\%\ Utilitized\ Capacity = \frac{Heat\ Rejection\ Load\ (Btu/hr)}{Available\ Capacity\ (Btu/hr)}$$

4.) The evaporative condenser fan and pump kW for each weather bin is calculated using the following equation:

$$kW_{cond} = e_0 + e_1 \times \%Utilized\ Capacity + e_2 \times (\%Utililized\ Capacity)^2$$

Where,

e0	25.5
e1	-72.1
e2	175.7

The coefficients appear to come from plotting data on the condenser fans and pumps, but the source of this data is not clear.

The condenser fan and pump kW (kW_cond) for each weather bin is then multiplied by the number of hours in that weather bin to determine the total baseline kWh for the condenser system. This is summed with the total baseline compressor kWh, to obtain the total baseline system kWh.

The post-case for the first refrigeration measure, 65˚ F minimum condensing setpoint, rather than 85˚, is calculated using all the same steps as above, except in the calculation of the condensing temperature, which in the baseline was:

Condensing temperature = IF(OAWB+TD_setpoint>85˚F,OAWB+TD_setpoint,85˚F)
TD_setpoint = 17˚ F.

Is changed to:

Condensing temperature = IF(OAWB+TD_setpoint>65˚F,OAWB+TD_setpoint,65˚F)
TD_setpoint = 17˚ F.

VFD control for evaporative condenser, rather than two-speed fan control

For this measure, the baseline energy calculated in the measure for the first refrigeration measure, *65˚ F minimum condensing setpoint, rather than 85˚*, is set as the baseline.

The post-case is calculated the same way as the post-case for the *65˚ F minimum condensing setpoint, rather than 85˚ post-case* with the only difference in the calculation of the evaporative condenser fan and pump kW.

The new equation is:

$$kW_{cond} = e_0 + e_1 \times \%Utilized\ Capacity + e_2 \times (\%Utililized\ Capacity)^2$$

Where,

e0	6.855021272
e1	-21.77334348
e2	91.66943689

The coefficients appear to come from plotting data on the condenser fans, but the source of this data is not clear.

Economized high-stage compressors with VFD control for lead ammonia compressor

For this measure, the baseline energy is the same as the post-case energy for the VFD control for evaporative condenser measure.

The post-case is calculated the same as the VFD control for the evaporative condenser measure, but with the following changes:

The load profile on each of the compressors is updated. The new load profile is shown in the table below. Also note that C-3 now has a VFD, rather than a slide-valve, which means the coefficients used in the equations for tons, bhp/ton and %bhp/ton as a function of SST and SCT are all updated. These coefficients come from a regression of data from the compressor manufacturer. Additionally, the coefficients for the C-4, C-5, and C-6 are also updated. This may be because the equations for the (4) high-stage compressors now use an SST of 22˚F rather than 20˚F, though it is not clear why/how the suction temperature would be reduced. Also note the lower minimum condensing temperatures (65˚F, rather than 85˚F).

OADB (˚F)	Condensing Temp (˚F)	C-1 Booster Compressor Tons	C-2 Pump Down Compressor Tons	C-3 High Stage Compressor #1 with VFD Tons	C-4 High Stage Compressor #2 with Slide Valve Tons	C-5 High Stage Compressor #3 with Slide Valve Tons	C-6 High Stage Compressor #4 with Slide Valve Tons	Total Tons
97.5	88.0	119	0	385	385	0	143	889
92.5	91.1	109	0	383	383	140	0	1,014
87.5	89.7	99	0	384	384	130	0	995
82.5	86.6	89	0	385	385	115	0	975
77.5	85.0	80	0	386	386	100	0	953
72.5	82.8	72	0	387	387	81	0	929
67.5	79.6	65	0	389	389	60	0	903
62.5	74.2	58	0	392	392	28	0	871
57.5	69.1	52	0	395	395	1	0	843
52.5	65.0	47	0	397	370	0	0	814
47.5	65.0	43	0	397	343	0	0	783
42.5	65.0	39	0	397	315	0	0	751
37.5	65.0	35	0	397	284	0	0	717
32.5	65.0	33	0	397	252	0	0	682
27.5	65.0	31	0	415	199	0	0	645
22.5	65.0	30	0	378	199	0	0	607
17.5	65.0	29	0	339	199	0	0	567
12.5	65.0	30	0	298	199	0	0	526
7.5	65.0	30	0	255	199	0	0	484
2.5	65.0	32	0	210	199	0	0	441

The equations to calculate tons, bhp/ton as a function of SST and SCT for the high stage compressors (C-3, C-4, C-5, and C-6, (the equations for the booster compressor does not change) are below. Note that only the equation for %bhp/ton are different for the high stage compressor with a VFD (C-3), and the high-stage compressors with slide-valve control (C-4, C-5, and C6). The equations for the %bhp/ton are presented subsequently. The coefficients for the compressor operation comes from regressions on performance data from the refrigeration compressor manufacturer.

$$tons = c_0 + c_1 \times T_{evap(R)} + c_2 \times T_{cond(R)} + c_3 \times T_{evap(R)} \times T_{cond(R)}$$

Where,

c0:	- 39183.89974
c1:	82.71576334
c2:	69.74929213
c3:	- 0.145845747

Suction temperature, T_evap = 22˚F

Condensing temperature= IF(OAWB+TD_setpoint>65˚F,OAWB+TD_setpoint,65˚F)

TD_setpoint = 17˚ F.

$$\frac{BHP}{ton} = c_0 + c \times T_{evap(R)} + c_2 \times T_{cond(R)} + c_3 \times T_{evap(R)} \times T_{cond(R)}$$

Where,

The equations to calculate the kW for C-3 High Stage Compressor #1 with VFD are updated as follows:

$$\frac{BHP}{ton} = c_0 + c_1 \times T_{evap(R)} + c \times T_{cond(R)} + c_3 \times T_{evap(R)} \times T_{cond(R)}$$

Where,

c0:	0
c1:	- 0.014692532
c2:	0.013019913
c3:	3.23404E-06

Suction temperature, T_evap = 22˚F

Condensing temperature= IF(OAWB+TD_setpoint>65˚F,OAWB+TD_setpoint,65˚F)

TD_setpoing = 17˚ F

For the high-stage compressor with the VFD (C-3), the equation for the %BHP/ton is:

$$\frac{\%BHP}{ton} = c + c_1 \times (1 - \%Capacity)$$

Where,

c0:	1
c1:	0.015310546

The equation for calculating compressor kW for the compressor with the VFD on it is:

$$kW = Tons \times \frac{BHP}{Ton} \times \% \frac{BHP}{Ton} \times 0.746 \frac{kW}{HP} \times \frac{1}{Motor\ Efficiency} \times \frac{1}{VFD\ Efficiency}$$

Where the motor efficiency is set at a static value of 95.8%, and the VFD efficiency is set at a static value of 97%.

For the high-stage compressors with slide-valve control, the equation for the %BHP/ton is the same, but the constants are:

c0:	1
c1:	0.406546133

The equation to calculate the kW for the compressors without VFDs are the same as above, but without the VFD efficiency term, and using the following coefficients:

c0:	1
c1:	0.0153105458304512

2.1.2 Evaluation assessment of applicant methodology

The evaluator determined that the applicant's calculation methodologies for each of the measures is reasonable. However, the evaluation team recommended further investigation into the industry standard practice baselines that were used for the industrial refrigeration measures should be investigated, and the results of that investigation resulted in recommended updates to the industry standard practice baselines for the minimum condensing temperature setpoint measure, as well as the VFD on evaporatively cooled condenser measure.

2.2 On-site inspection and metering

The site contact indicated that it was safe to visit the site and accommodated an on-site visit for the installation of loggers, as well as facilitated the collection of trend data from their refrigeration control system. The evaluator conducted the site visit on October 31, 2023, and was assisted by the facility manager. Table 2-2 summarizes the list of data that was incorporated into the evaluator analysis.

Table 2-2. Data Collection

PA Project Number	Measure name	Equipment Name	Measurement	Start Date - End Date	Interval
11216625 / 11983237	(2) 177 HP VFD Air Compressors	Air Compressor 1 (with VFD)	Post kW	10/31/23 – 2/10/24	5-minute
11216625 / 11983237	(2) 177 HP VFD Air Compressors	Air Compressor 2 (no VFD)	Post Amps	10/31/23 – 2/10/24	5-minute
11413637 / 11983247	Refrigeration Measures	N/A	Outdoor Air Temperature, Humidity, Wet-Bulb from trend system	7/1/23 – 12/31/23 and 10/17/23 – 1/1/24	Hourly
11413637 / 11983247	Refrigeration Measures	Refrigerant Compressor 1, 2, 3, 4, 5, 6	Discharge Pressure, Suction Pressure, Percent Loading, Amps from system	7/1/23 – 12/31/23 and 10/17/23 – 1/1/24	Hourly
11413637 / 11983247	Refrigeration Measures	C-1 100 HP Booster Compressor	Post kW	10/31/23 – 2/10/24	5-minute
11413637 / 11983247	Refrigeration Measures	C-2 75 HP Pump Down Compressor	Post Amps	10/31/23 – 2/10/24	5-minute
11413637 / 11983247	Refrigeration Measures	C-3 450 HP High Stage	Post kW	10/31/23 – 2/10/24	5-minute

PA Project Number	Measure name	Equipment Name	Measurement	Start Date - End Date	Interval
		Compressor w/ VFD			
11413637 / 11983247	Refrigeration Measures	C-4 450 HP High Stage HP Compressor with Slide Valve	Post kW	10/31/23 – 2/10/24	5-minute
11413637 / 11983247	Refrigeration Measures	C-5 450 HP High Stage HP Compressor with Slide Valve	Post kW	10/31/23 – 2/10/24	5-minute
11413637 / 11983247	Refrigeration Measures	C-6 450 HP High Stage HP Compressor with Slide Valve	Post Amps	10/31/23 – 2/10/24	5-minute
11413637 / 11983247	Refrigeration Measures	30 HP Condenser Fan 1	Post kW	10/31/23 – 2/10/24	5-minute
11413637 / 11983247	Refrigeration Measures	30 HP Condenser Fan 2	Post Amps	10/31/23 – 2/10/24	5-minute
11413637 / 11983247	Refrigeration Measures	30 HP Condenser Fan 2	Post Amps	Data logger malfunction	N/A

Figure 2-2 shows the kW data collected on air compressor 1, and the Amp data collected on air compressor 2. The data in Figure 2-2 shows that air compressor 1 was on for 99.7% of the time, and compressor 2 was on for 0.24% of the time. During the site visit, the compressors were observed to be producing air at 115 psi.

Figure 2-2. kW data collected on air compressor 1 and Amp data collected on air compressor 2

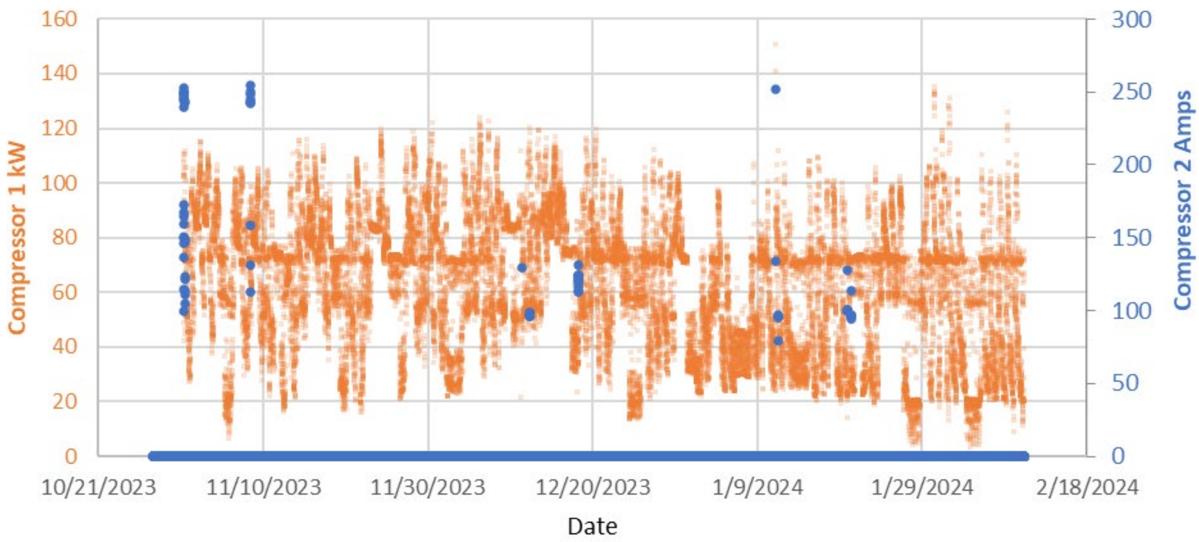


Figure 2-3 shows a heat map of the 103-day evaluation monitoring period of compressor 1 kW use. Note that there is no difference in use between weekdays, weekends, and holidays, and no down time.

Figure 2-3. Evaluation monitoring period heat map of kW compressor 1 kW use

Week #	Weekday	Date	Hour																							
			-	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
1	Holiday	1/1/2024	43	45	30	31	51	35	35	33	48	31	34	41	28	35	29	28	27	26	24	36	23	33	23	25
1	Tue	1/2/2024	35	23	54	24	37	61	70	45	55	54	29	55	62	52	50	47	47	42	43	40	43	37	34	30
1	Wed	1/3/2024	51	56	50	43	41	58	55	34	66	57	51	68	72	57	69	58	49	56	58	50	72	60	56	54
1	Thu	1/4/2024	58	56	72	69	72	89	95	73	94	95	68	70	73	64	51	39	26	33	34	26	28	31	25	28
1	Fri	1/5/2024	33	47	40	36	45	77	72	55	72	57	55	66	46	48	48	45	34	31	32	41	42	31	24	25
1	Sat	1/6/2024	30	42	43	38	31	24	24	30	38	46	45	40	50	30	41	42	44	40	38	36	30	30	30	30
2	Sun	1/7/2024	30	30	30	30	31	35	31	36	42	45	45	45	30	30	30	30	30	30	30	36	43	43	45	44
2	Mon	1/8/2024	49	47	39	33	46	58	70	57	73	72	44	72	107	51	36	39	47	41	37	45	33	47	44	30
2	Tue	1/9/2024	35	43	33	41	47	76	77	54	80	76	59	72	81	53	41	46	29	72	76	70	74	73	71	69
2	Wed	1/10/2024	71	76	66	70	86	91	71	50	70	60	42	71	70	40	47	35	42	51	41	43	31	27	25	27
2	Thu	1/11/2024	51	46	41	37	141	73	87	66	86	82	74	78	72	29	46	76	31	34	48	35	75	72	26	70
2	Fri	1/12/2024	33	36	51	25	67	68	73	92	74	71	56	57	58	34	32	42	73	43	43	32	64	50	44	69
2	Sat	1/13/2024	45	35	36	70	37	39	25	62	37	39	41	67	30	33	37	70	35	36	30	70	38	37	38	71
3	Sun	1/14/2024	33	28	27	68	33	38	47	37	26	32	36	72	38	33	71	28	33	30	27	72	29	25	63	36
3	Mon	1/15/2024	30	29	75	31	38	67	104	51	72	73	89	72	75	75	48	47	32	48	74	26	35	65	25	22
3	Tue	1/16/2024	25	71	25	39	26	90	72	51	54	74	84	73	70	78	45	36	28	78	44	28	71	41	35	40
3	Wed	1/17/2024	29	77	35	28	75	48	54	44	62	89	45	57	97	26	38	42	67	73	72	65	72	72	56	56
3	Thu	1/18/2024	73	71	73	73	72	90	96	74	86	93	76	63	56	70	30	31	31	70	39	25	26	73	23	23
3	Fri	1/19/2024	30	73	25	24	26	94	72	39	58	83	29	48	50	76	35	37	33	76	39	25	44	30	24	22
3	Sat	1/20/2024	30	73	58	72	66	0	38	69	35	26	25	24	72	26	27	72	29	24	25	24	73	73	73	72
4	Sun	1/21/2024	67	73	73	73	73	72	71	67	61	73	74	60	73	60	74	74	74	31	23	73	29	27	25	28
4	Mon	1/22/2024	73	27	34	33	72	53	55	35	95	49	38	67	61	40	32	45	72	38	37	63	39	38	24	26
4	Tue	1/23/2024	28	72	39	26	72	55	66	48	63	103	32	42	79	22	23	24	73	73	74	67	74	74	62	56
4	Wed	1/24/2024	73	69	74	65	73	83	89	72	92	84	74	89	82	73	80	74	72	78	76	70	76	59	68	67
4	Thu	1/25/2024	72	73	72	72	73	88	92	87	96	98	74	86	81	59	73	67	73	70	55	69	61	73	68	72
4	Fri	1/26/2024	70	72	72	59	72	89	101	86	96	93	77	89	92	69	72	73	56	60	23	31	32	25	64	26
4	Sat	1/27/2024	27	25	29	21	19	56	22	19	8	12	51	9	7	17	57	20	20	14	53	8	4	9	40	20
5	Sun	1/28/2024	16	20	13	56	20	20	19	59	20	20	13	18	56	5	14	8	19	56	72	72	71	72	67	60
5	Mon	1/29/2024	67	59	68	57	61	85	81	43	60	79	30	51	89	70	101	71	71	75	73	71	106	73	27	22
5	Tue	1/30/2024	72	31	33	68	27	57	73	47	96	72	85	82	133	69	79	72	102	73	72	101	72	55	65	23
5	Wed	1/31/2024	71	39	41	69	20	55	71	83	57	92	71	115	76	58	72	102	63	72	89	61	73	58	24	73
5	Thu	2/1/2024	31	23	72	21	52	92	128	80	58	58	81	75	53	37	36	35	72	43	42	71	29	29	35	21
5	Fri	2/2/2024	70	30	29	50	25	71	69	46	96	71	34	82	60	42	38	79	31	43	30	72	40	32	28	75
5	Sat	2/3/2024	43	37	71	24	24	29	34	56	21	17	19	58	19	20	19	56	21	21	19	56	19	20	20	19
6	Sun	2/4/2024	67	20	13	18	56	19	9	19	68	15	15	21	10	68	19	20	17	57	21	21	18	56	9	20
6	Mon	2/5/2024	27	27	47	21	26	72	102	36	71	72	78	69	64	71	33	32	37	70	30	31	35	73	23	28
6	Tue	2/6/2024	27	67	28	37	22	90	73	41	76	101	38	38	31	67	42	39	25	70	26	26	21	50	27	44
6	Wed	2/7/2024	32	70	29	24	74	56	55	53	60	94	42	55	94	35	40	40	40	75	33	25	71	29	21	22
6	Thu	2/8/2024	40	73	25	22	72	55	54	29	55	92	81	89	83	69	32	34	32	71	29	30	27	63	23	25
6	Fri	2/9/2024	57	71	42	25	20	92	55	37	70	101	50	59	92	38	42	40	38	72	31	25	71	26	24	26
6	Sat	2/10/2024	66	27	34	20	55	21	21	20	66															
44	Tue	10/31/2023										42	76	72	71	97	68	56	88	52	52	52	63	60	34	45
44	Wed	11/1/2023	74	30	47	44	71	95	96	86	69	101	65	65	55	48	79	91	91	88	89	84	89	83	84	92
44	Thu	11/2/2023	94	85	89	84	88	105	101	96	111	110	104	96	70	73	74	103	72	73	64	55	89	86	86	93
44	Fri	11/3/2023	100	88	91	94	84	103	107	105	99	75	77	102	72	74	57	71	56	44	79	38	42	47	56	83
44	Sat	11/4/2023	86	79	77	78	76	90	94	89	81	97	93	87	94	86	87	76	56	59	59	73	59	73	72	74
45	Sun	11/5/2023	72	58	58	72	73	60	58	60	15	23	29	73	21	20	55	21	18	56	14	21	55	11	20	62
45	Mon	11/6/2023	30	22	28	62	23	37	90	51	71	54	87	89	82	96	100	98	95	84	80	77	79	78	76	80
45	Tue	11/7/2023	87	76	78	83	79	93	105	93	86	71	71	69	103	51	69	39	49	82	53	76	49	46	44	47
45	Wed	11/8/2023	86	47	52	80	49	73	72	0	52	65	99	46	62	64	65	87	84	84	83	78	72	78	77	80
45	Thu	11/9/2023	82	76	81	83	80	90	98	88	95	102	96	89	73	54	72	86	62	43	59	74	50	41	76	41
45	Fri	11/10/2023	46	52	86	40	44	91	59	52	72	73	95	72	71	96	62	54	51	47	69	36	73	41	54	47
45	Sat	11/11/2023	48	81	52	58	75	34	38	31	77	76	74	75	70	72	73	73	73	67	74	72	74	74	73	74
46	Sun	11/12/2023	74	75	74	73	74	38	26	30	72	24	52	25	35	66	18	24	59	60	61	54	70	63	63	63
46	Mon	11/13/2023	73	68	64	67	66	89	72	33	90	65	37	54	61	71	46	47	73	25	34	27	73	49	38	61
46	Tue	11/14/2023	50	50	51	51	76	99	107	55	103	78	65	75	73	91	59	56	90	62	53	55	88	66	65	83
46	Wed	11/15/2023	55	65	55	79	51	80	109	73	80	78	71	100	102	94	88	86	81	91	87	80	83	78	76	76
46	Thu	11/16/2023	87	79	83	81	69	91	96	90	101	99	76	95	97	74	76	75	73	72	72	60	77	77	85	78
46	Fri	11/17/2023	94	83	87	92	77	109	108	53	80	57	85	55	66	51	53	58	86	73	57	83	57	49	44	35
46	Sat	11/18/2023	79	94	87	84	79	80	79	78	55	73	59	72	68	60	58	68	66	67	72	73	58	72	73	57
47	Sun	11/19/2023	72	61	72	72	58	58	72	72	25	25	29	36	72	30	55	23	33	59	23	18	58	24	22	72
47	Mon	11/20/2023	44	27	24	67	72	89	93	51	88	71	44	70	64	84	80	92	47	52	82	49	46	47	48	84
47	Tue	11/21/2023	41	39	74	52	21	61	96	83	105	104	80	102	100	93	90	55	50	54	73	20	66	24	29	65
47	Wed	11/22/2023	47	38	36	71	28	56	91	41	71	60	57	107	72	52	81	74	55	64	91	62	58	89	53	69
47	Holiday	11/23/2023	54	80	84	85	84	85	84	85	84	83	83	84	84	84	83	84	84	84	83	82	84	84	83	84
47	Fri	11/24/2023	94	89	90	88	85	110	118	94	115	108	83	89	89	76	75	74	73	79	80	77	78	77	76	77
47	Sat	11/25/2023	87	85	78	79	77	79	78	71	67	72	73	58	73	74	73	58	73	72	68	73	58	73	58	73
48	Sun	11/26/2023	57	73	69	73	64	73	65	73	72	73	72	58	66	73	26	37	23	24	72	27	30	71	32	26
48	Mon	11/27/2023	42	70	35	41	37	97	81	59	108	72	48	75	70	90	89	91	88	93	86	77	87	78	73	71
48	Tue	11/28/2023	81	83	84	79	79	111	113	94	116	79	63	71	109	61	49	48	50	82	49	49	91	56	53	39
48	Wed	11/29/2023	84	56	54	89	57	71	72	62	83	73	79	56	56	49	50	94	73	58	95	56	73	57	90	90
48	Thu	11/30/2023	103	96	92	98	94	105	79	57	116	73	62	76	83	89	53	48	80	43	51	39	83	84	80	83
48	Fri	12/1/2023	97	93	86	80	84	100	105	55	73	102	44	54	56	43	78	47	46	82	38	32	48	93	55	48
48	Sat	12/2/2023	98	61	65	55	44	85	39	39	71	36	27	25	63	27	28	68	26	34	27	24	72	32	36	73
49	Sun	12/3/2023	34	26	34	73	71	72	68	71	72	72	67	71	38	23	71	30	26	35	33	26	67	36	38	72
49	Mon	12/4/2023	34	44	49	74	36	70	111	50	74	78	59	100	98	93	54	54	87	52	52	49	92	53	55	85
49	Tue	12/5/2023	75	71	54	59	98	78	83	98	70	75	70	102	72	61	93	56	53	56	72	89	60	58	86	53
49	Wed	12/6/2023	66	72	98	98	89	109	122	89	106	114	89	75	68	65	36	39	35	47	79	33	42	89	53	52
49	Thu	12/7/2023	58	107	73	64	86	74	86	69	75	117	63	71	97	62	58	56	89	56	53	86	55	51	48	52
49	Fri	12/8/2023	93	59	53	83	49	76	82	85	103	112	96	94	93	77	54	56	90	56	55	57	96	73	57	85
49	Sat	12/9/2023	68	54	72	53	83	52	54	86	51	55	54	89	90	87	88	87	86	88	87	84	84	84	83	83
50	Sun	12/10/2023	84	84	83	84	84	83	45	51	52	83	50	56	44	83	68	46	83	50	51	51	62	83	85	85
50	Mon	12/11/2023	86	86	91				0	0	51	50	76	68	74	59	84	60	53	89	68	52	53	57	88	61
50	Tue	12/12/2023	71	91	67	53	57	103	82	66	115	75	57	56	78	95	87	56	87	57	54	55	55	84	54	53
50	Wed	12/13/2023	85	58	63	54	87	68	83	88	81	76	69	97	106	68	54	86	59	54	55	85	90	88	87	85
50	Thu	12/14/2023	88	96	92	85	90	101	110	67	84	103	58	79	82	81	93	88	90	92	88	87	91	82	87	84
50	Fri	12/15/2023	95	95	93	84	86	105	102	90	112	106	96	106	113	83	96	95	79	90	95	78	90	98	90	85
50	Sat	12/16/2023	91	105	96	91	83	88	94	74	96	94	72	94	85	73	75	74	75	76	82	82	75	74	74	74
51	Sun	12/17/2023	74	75	74	74	74	74	75	74	74	75	33	46	35	76	58	38	38	39	76	35	31	42	75	34
51	Mon	12/18/2023	36	39	72	42	0	38	38	34	72	70	53	67	69	65	51	54	44	91	72	53	69	56	86	54
51	Tue	12/19/2023	51	91	54	43	38	111	81	55	109	74	73	76	74	70	59	83	51	60	73	47	90	92	82	82
51	Wed	12/20/2023	79	82	74	77	79	105	106	40	71	120	61	74	82	58	80	48	29	76	35	35	29	73	73	60
51	Thu	12/21/2023	62	71	62	71	73	89	101	41	61	90	35	73	59	72	43	47	73	46	40	31	37	76	44	43
51	Fri	12/22/2023	73	51	47	47	75	86	56	85	70	71	49	61	84	69	74	80	73	77	78	72	75	75	70	72
51	Sat	12/23/2023	81	83	82	77	75	80	92	83	95	97	76	80	75	59	73	59	62	58	73	76	71	68	59	73
52	Sun	12/24/2023	61	54	66	60	74	52	72	58	55	59	24	14	24	45	20	15	60	21	15	66	18	18	54	21
52	Holiday	12/25/2023	25	70	16	19	56	58	59	51	58	63	20	58	28	19	59	24	18	58	23	18	55	50	58	64
52	Tue	12/26/2023	59	64	73	56	73	91	72	55	101	72	56	58	94	69	70	40	39	73	32	24	72	42	31	72
52	Wed	12/27/2023	35	57	41	31	42	50	58	82	77	74	53	103	69	40	73	41	47	60	71	44	51	40	73	34
52	Thu	12/28/2023	44	33	35	74	48	59	100	51	72	71	92	53	71	87	48	45	45	54	83	45	49	89	35	35
52	Fri	12/29/2023	44	39	39	31	72	97	72	87	71	75	52	63	100	75	78	44	37	77	56	51	52	52	49	78
52	Sat	12/30/2023	84	88	78	79	82	87	95	90	97	99	84	93	93	74	75	75	75	73	73	79	77	72	71	73
53	Sun	12/31/2023	74	73	72	71	73	72	72	72	73	72	72	50	32	32	45	44	37	33	45	42	32	31	41	35

Figure 2-4. CFM vs kW data from CAGI sheet for post case VFD air compressor Atlas Copco G132VSD-145 at 116 psig and 102 psig

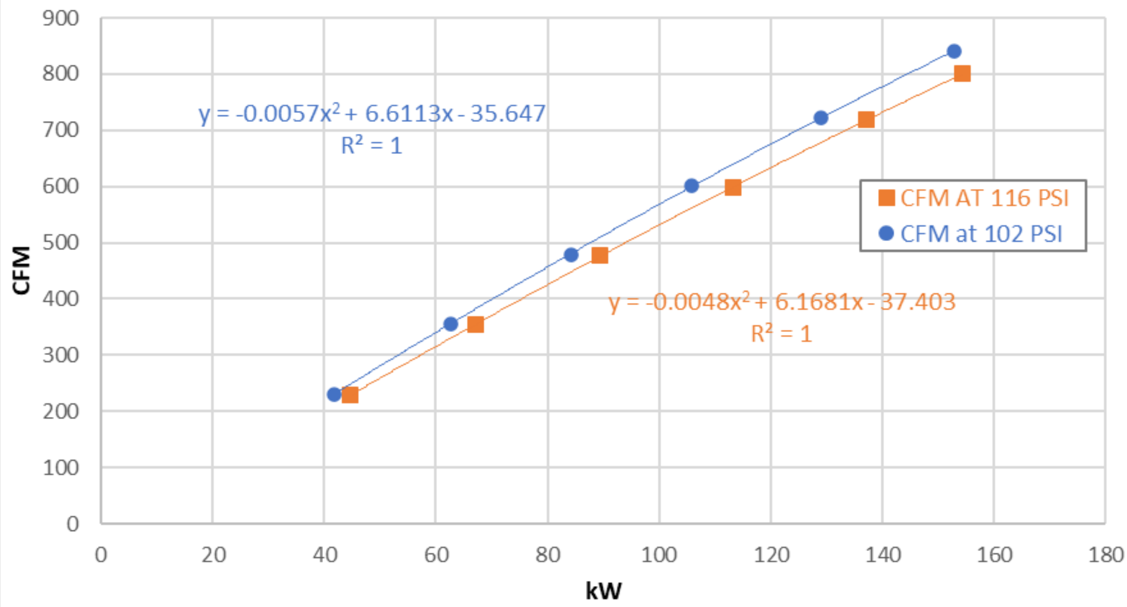
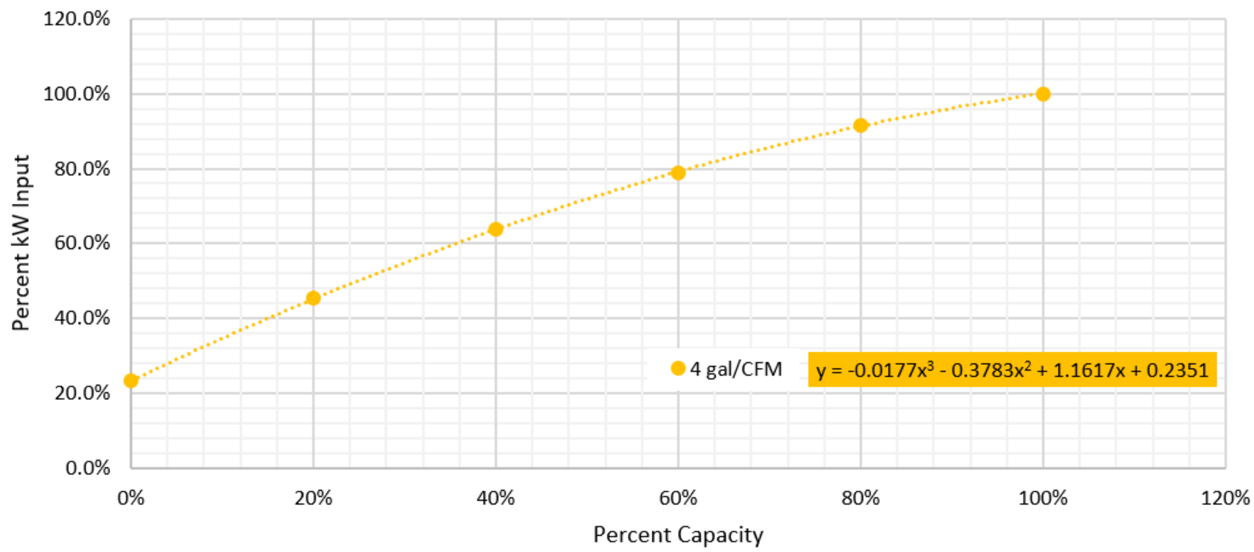


Figure 2- shows the % kW vs % CFM capacity for the industry standard practice load/no-load compressor operating at 115 psi, with 4 gal/cfm storage that was used in the evaluator calculations.

Figure 2-4 % kW vs. % CFM capacity for baseline air compressor operating at 115 psi, with 4 gal/CFM of storage, based on Compressed Air Challenge Data



Refrigeration Measures

Figure 2-5 shows the saturated condensing temperature for refrigeration compressors 3, 4, 5, and 6 vs outdoor air temperature. This data was based on converting the discharge pressure data from the facility's trend data system to saturated condensing pressure data using the properties of saturated ammonia tables. The data in Figure 2-5 shows that when the outdoor air temperature is below about 57˚ F, the average saturated condensing temperature is on average 71.2˚F. This is greater than the 65˚F minimum saturated condensing temperature that was indicated in the measure description and is the primary reason for the lower realization rate found in the evaluation findings for the "*65˚ minimum condensing setpoint rather than 85˚ minimum condensing setpoint"* measure.

Figure 2-5. Refrigeration compressors 3,4,5 and 6 saturated condensing temperature vs. outdoor air temperature

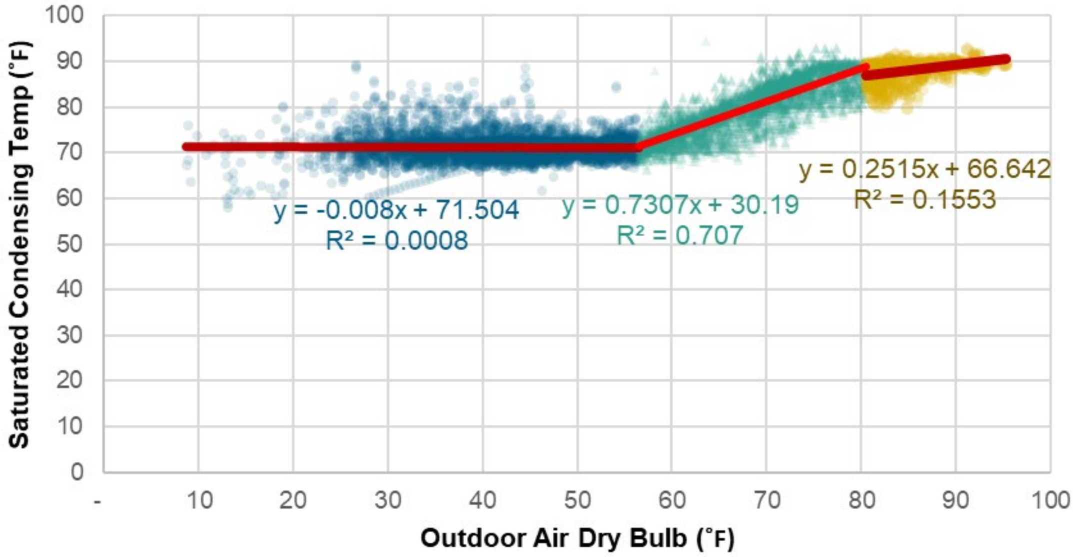


Figure 2-6 shows the suction temperature vs. outdoor air temperature for refrigeration compressors 3, 4, 5, and 6. This is based on suction pressure data from the facility's trend data system, converted to temperature using the properties of ammonia. The data in Figure 2-6 shows that the average suction temperature is 21.8˚F.

Figure 2-6. Refrigeration compressors 3,4,5 and 6 suction temperature vs. outdoor air temperature

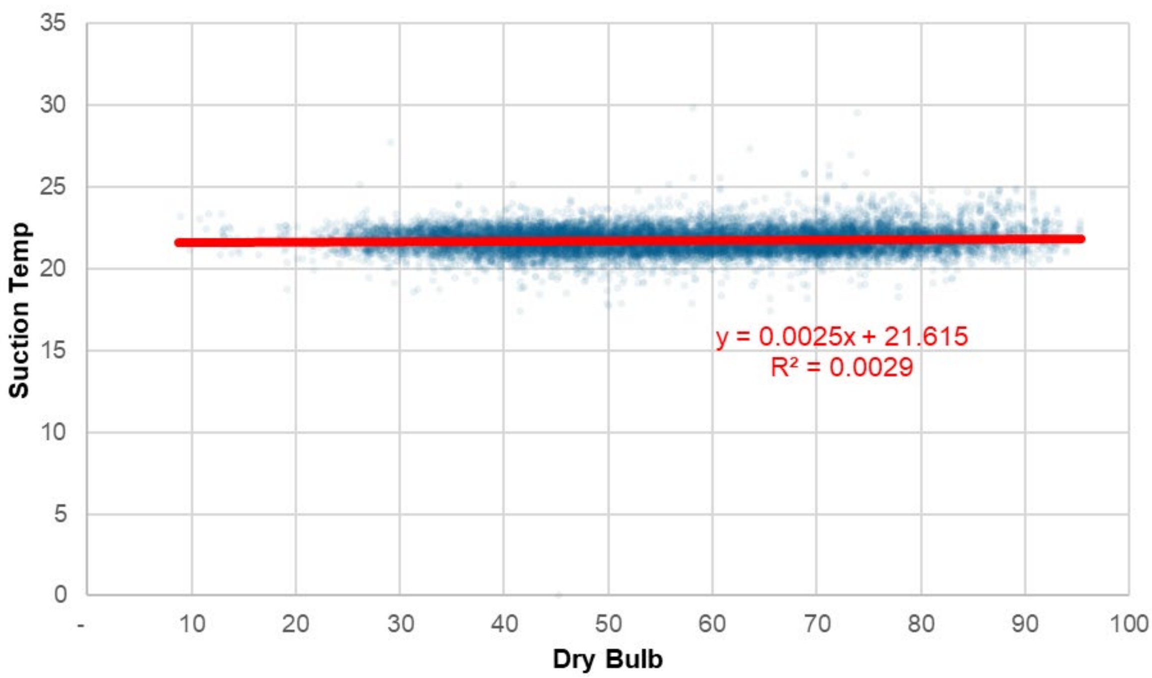


Figure 2-7 shows the % on data for Ammonia compressors 1-6 based on the hour of the day, and day of the week. This is based on the approximately 9 months of Amperage trend data provided by the facility on each of these compressors. Table 2-3 shows a higher level picture of the data shown in Figure 2-7, by showing the total % on for each of the Ammonia compressors.

Table 2-3. % On data for Ammonia compressors 1-6 for entire 9-month evaluation monitoring period

Ammonia Compressor	% On
C1	55.1%
C2	35.6%
C3 (w VFD)	98.7%
C4 (slide valve)	88.5%
C5(slide valve)	12.6%
C6 (slide valve)	1.9%

Figure 2-7. %On data for Ammonia compressors 1-6 based on hour of day and day of week

Ammonia Compressor 1							
	Sun	Mon	Tue	Wed	Thu	Fri	Sat
0	0%	2%	6%	5%	14%	11%	5%
1	0%	0%	0%	0%	0%	0%	3%
2	0%	0%	0%	0%	0%	0%	0%
3	0%	0%	0%	3%	0%	0%	0%
4	0%	23%	17%	22%	8%	0%	0%
5	0%	74%	78%	86%	54%	32%	24%
6	11%	81%	100%	100%	95%	97%	76%
7	14%	81%	100%	100%	95%	100%	76%
8	14%	81%	100%	100%	95%	100%	76%
9	14%	83%	100%	100%	95%	100%	76%
10	11%	85%	100%	100%	95%	100%	79%
11	11%	85%	100%	100%	95%	100%	79%
12	14%	85%	100%	100%	95%	100%	76%
13	11%	85%	100%	100%	95%	100%	76%
14	14%	85%	97%	100%	95%	100%	76%
15	14%	85%	100%	100%	95%	100%	74%
16	16%	85%	100%	100%	95%	100%	74%
17	16%	85%	100%	100%	95%	100%	66%
18	11%	85%	100%	97%	70%	63%	16%
19	11%	81%	100%	86%	65%	58%	3%
20	5%	69%	92%	81%	59%	55%	0%
21	3%	63%	76%	70%	57%	45%	0%
22	3%	60%	49%	70%	51%	45%	0%
23	0%	46%	43%	46%	30%	34%	0%

Ammonia Compressor 2							
	Sun	Mon	Tue	Wed	Thu	Fri	Sat
0	0%	23%	6%	11%	5%	5%	3%
1	0%	28%	11%	16%	5%	0%	0%
2	0%	32%	28%	30%	8%	3%	3%
3	3%	36%	28%	27%	11%	3%	3%
4	0%	67%	64%	57%	38%	26%	24%
5	8%	77%	94%	97%	92%	95%	71%
6	11%	67%	72%	65%	65%	74%	58%
7	11%	53%	56%	57%	51%	51%	42%
8	11%	54%	57%	57%	51%	53%	42%
9	8%	52%	57%	54%	51%	54%	42%
10	8%	55%	57%	54%	51%	57%	42%
11	8%	54%	57%	54%	50%	58%	42%
12	8%	54%	57%	56%	51%	55%	42%
13	5%	54%	57%	54%	51%	58%	42%
14	3%	54%	56%	54%	51%	53%	42%
15	3%	54%	57%	54%	51%	55%	41%
16	3%	54%	57%	54%	46%	47%	32%
17	0%	56%	57%	43%	32%	34%	11%
18	0%	52%	57%	43%	30%	32%	8%
19	3%	50%	49%	41%	30%	32%	3%
20	3%	45%	43%	38%	30%	32%	0%
21	0%	46%	32%	32%	30%	26%	0%
22	3%	42%	27%	30%	24%	24%	0%
23	3%	33%	8%	19%	16%	21%	0%

Ammonia Compressor 3 (with VFD)							
	Sun	Mon	Tue	Wed	Thu	Fri	Sat
0	97%	98%	100%	100%	97%	100%	97%
1	97%	98%	97%	100%	97%	100%	97%
2	97%	98%	100%	100%	97%	100%	97%
3	97%	100%	100%	100%	97%	100%	95%
4	97%	96%	100%	100%	100%	100%	97%
5	97%	100%	100%	100%	100%	100%	97%
6	97%	100%	100%	100%	100%	100%	97%
7	100%	98%	100%	100%	100%	97%	97%
8	100%	98%	100%	97%	97%	97%	97%
9	100%	100%	100%	95%	100%	97%	97%
10	100%	100%	100%	97%	100%	95%	97%
11	100%	100%	100%	97%	100%	97%	97%
12	100%	100%	100%	97%	100%	97%	97%
13	100%	100%	100%	97%	100%	95%	97%
14	100%	100%	100%	97%	100%	97%	97%
15	100%	100%	100%	97%	100%	97%	97%
16	100%	100%	100%	97%	100%	97%	97%
17	100%	100%	100%	100%	100%	97%	97%
18	97%	100%	100%	100%	100%	97%	97%
19	97%	100%	100%	100%	100%	97%	97%
20	97%	98%	100%	100%	100%	97%	97%
21	97%	100%	100%	97%	100%	97%	97%
22	97%	100%	100%	97%	100%	97%	97%
23	97%	100%	100%	97%	97%	97%	97%

Ammonia Compressor 4 (with slide valve)							
	Sun	Mon	Tue	Wed	Thu	Fri	Sat
0	84%	70%	92%	81%	84%	87%	81%
1	85%	70%	91%	89%	86%	92%	89%
2	70%	72%	94%	89%	86%	92%	86%
3	68%	67%	91%	86%	84%	92%	86%
4	65%	77%	94%	92%	81%	92%	82%
5	76%	88%	97%	92%	86%	95%	84%
6	81%	87%	97%	92%	89%	95%	89%
7	78%	83%	97%	92%	95%	95%	89%
8	73%	90%	97%	95%	95%	95%	89%
9	70%	90%	94%	92%	97%	95%	89%
10	73%	90%	95%	95%	97%	92%	89%
11	76%	92%	97%	92%	97%	94%	89%
12	84%	94%	97%	92%	97%	95%	89%
13	86%	94%	97%	92%	97%	95%	89%
14	81%	92%	94%	92%	97%	95%	89%
15	81%	92%	95%	92%	97%	95%	89%
16	76%	90%	89%	89%	97%	95%	89%
17	81%	92%	92%	89%	97%	95%	89%
18	84%	92%	89%	89%	97%	95%	84%
19	86%	92%	89%	89%	97%	92%	76%
20	81%	88%	89%	89%	97%	89%	76%
21	81%	88%	89%	89%	97%	89%	82%
22	84%	87%	92%	89%	95%	89%	82%
23	84%	87%	92%	86%	89%	71%	87%

Ammonia Compressor 5 (with slide valve)							
	Sun	Mon	Tue	Wed	Thu	Fri	Sat
0	8%	6%	6%	11%	14%	5%	5%
1	8%	8%	9%	11%	14%	5%	5%
2	8%	8%	9%	11%	14%	5%	5%
3	8%	8%	9%	8%	14%	5%	8%
4	8%	17%	20%	32%	16%	5%	11%
5	30%	27%	46%	49%	46%	31%	43%
6	32%	21%	29%	43%	38%	45%	39%
7	14%	19%	17%	30%	32%	29%	34%
8	8%	8%	11%	27%	22%	21%	24%
9	5%	6%	8%	19%	14%	19%	13%
10	3%	8%	14%	14%	11%	17%	13%
11	3%	6%	14%	16%	14%	17%	13%
12	3%	4%	8%	19%	14%	18%	13%
13	3%	2%	11%	16%	11%	13%	11%
14	3%	4%	11%	16%	11%	13%	11%
15	3%	2%	8%	11%	11%	11%	8%
16	3%	2%	19%	14%	11%	11%	11%
17	8%	2%	16%	14%	8%	11%	8%
18	8%	2%	14%	14%	5%	8%	5%
19	8%	2%	11%	11%	3%	8%	5%
20	8%	4%	11%	11%	3%	8%	5%
21	8%	4%	11%	14%	3%	8%	5%
22	8%	4%	8%	14%	5%	8%	3%
23	8%	4%	8%	14%	8%	8%	5%

Ammonia Compressor 6 (with slide valve)							
	Sun	Mon	Tue	Wed	Thu	Fri	Sat
0	3%	0%	0%	0%	0%	0%	5%
1	3%	0%	0%	0%	0%	0%	3%
2	3%	0%	0%	0%	0%	0%	3%
3	0%	0%	0%	0%	0%	0%	3%
4	0%	4%	6%	5%	3%	0%	3%
5	8%	4%	6%	5%	11%	8%	22%
6	3%	6%	6%	5%	11%	0%	3%
7	0%	4%	6%	0%	3%	3%	3%
8	0%	2%	0%	3%	3%	3%	3%
9	0%	2%	3%	5%	0%	3%	3%
10	0%	0%	6%	3%	0%	6%	3%
11	0%	0%	3%	3%	0%	3%	3%
12	0%	0%	3%	3%	0%	3%	3%
13	0%	0%	0%	3%	0%	5%	3%
14	0%	0%	0%	3%	0%	3%	3%
15	0%	0%	0%	3%	0%	3%	3%
16	0%	0%	0%	5%	0%	5%	3%
17	0%	0%	0%	3%	0%	3%	3%
18	0%	0%	0%	3%	0%	3%	3%
19	0%	0%	3%	3%	0%	3%	3%
20	0%	0%	0%	0%	0%	5%	0%
21	0%	0%	0%	0%	0%	5%	0%
22	0%	0%	0%	0%	0%	5%	3%
23	0%	0%	0%	0%	0%	3%	3%

Figure 2-8 through Figure 2-13 shows the trended Amp data vs. outdoor air temperature for Ammonia compressors 1-6.

Figure 2-8. Ammonia compressor 1 trended Amps vs. outdoor air temperature when on

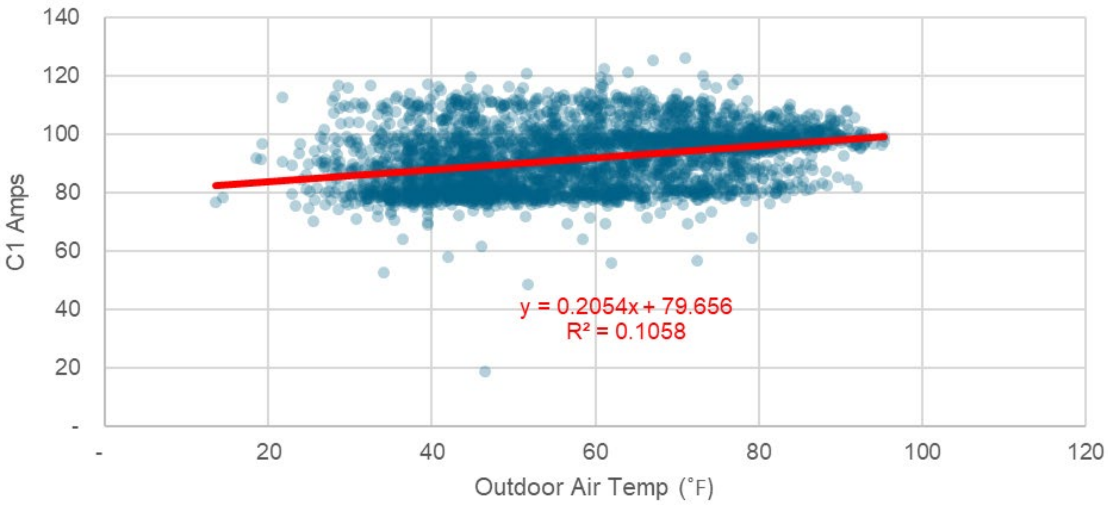


Figure 2-9. Ammonia compressor 2 trended Amps vs. outdoor air temperature when on

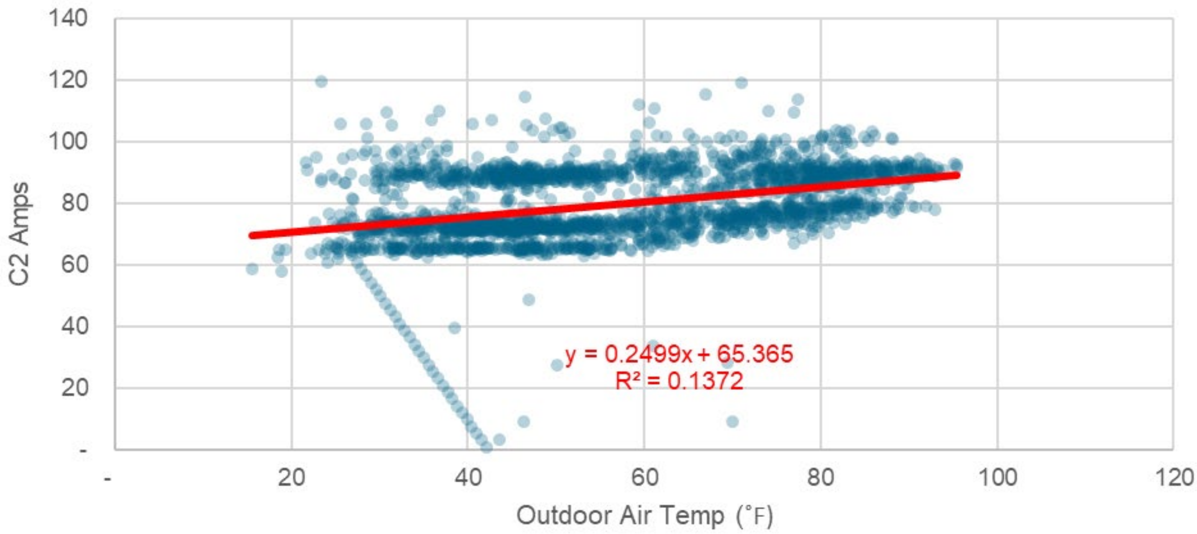


Figure 2-10. Ammonia compressor 3 trended Amps vs. outdoor air temperature when on

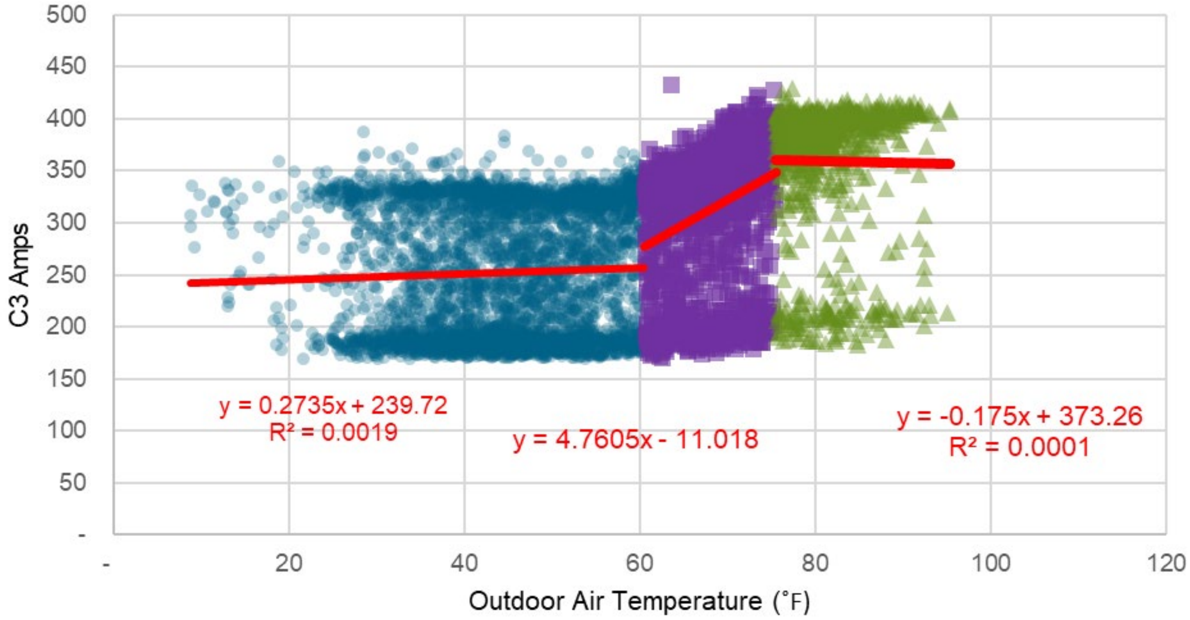


Figure 2-11. Ammonia compressor 4 trended Amps vs. outdoor air temperature when on

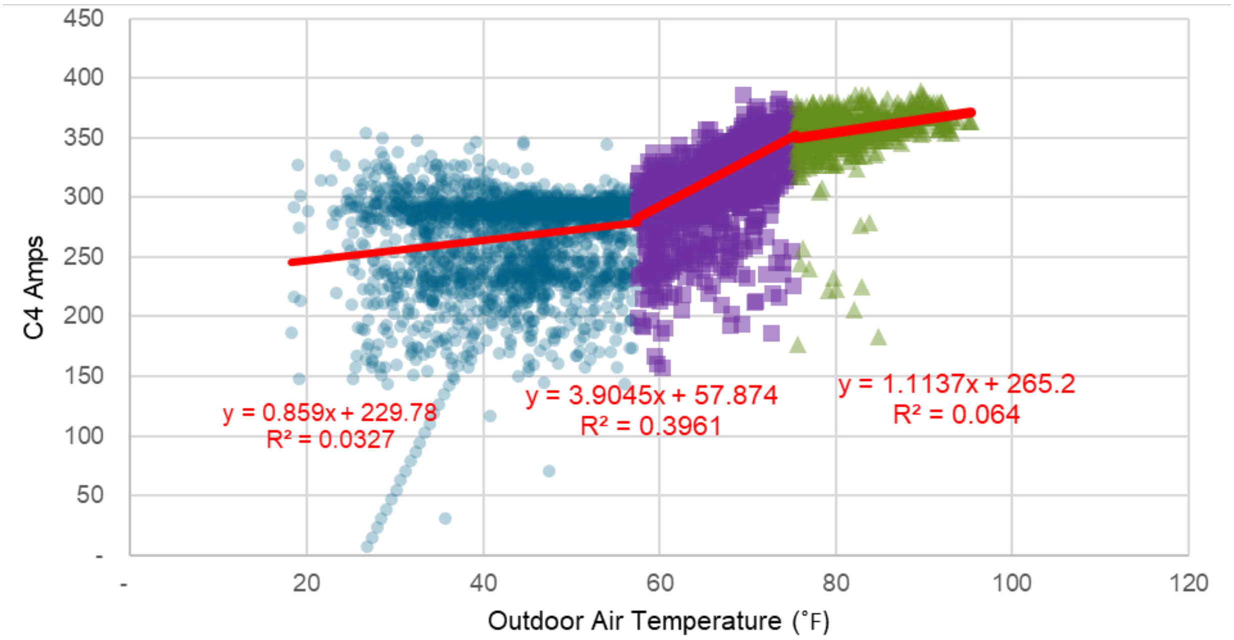


Figure 2-12. Ammonia compressor 5 trended Amps vs. outdoor air temperature when on

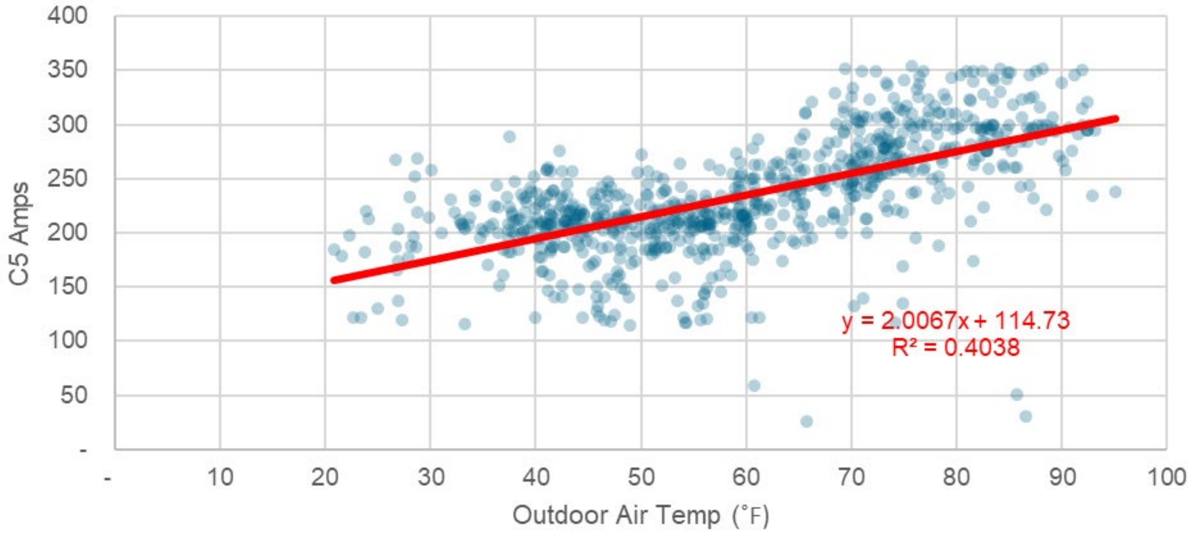


Figure 2-13. Ammonia compressor 6 trended Amps vs. outdoor air temperature when on

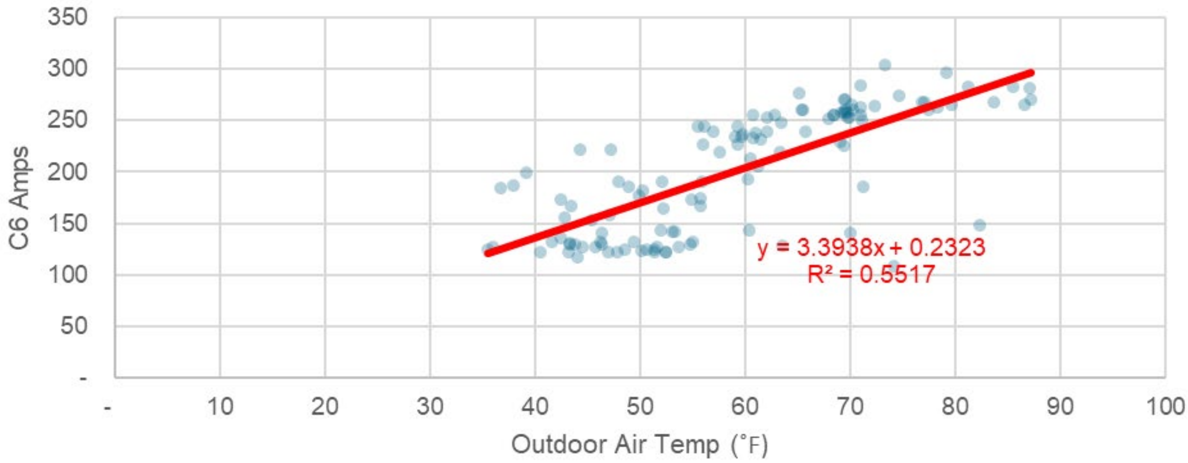
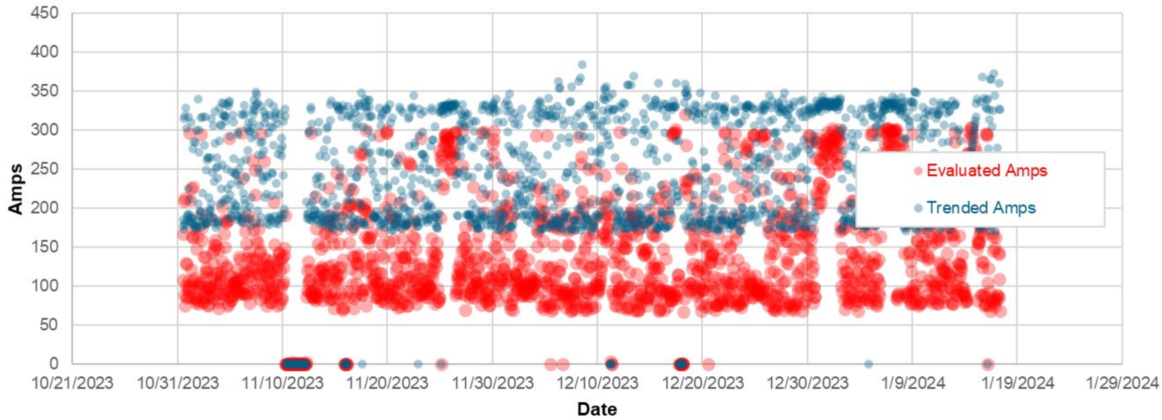


Figure 2-14 shows the Amps provided by the facility's trending system plotted alongside the Amps measured using the evaluator's revenue grade kW loggers. As Figure 2-14 shows, the Amps from the facility's trending system are significantly higher than the Amps measured with the evaluator's kW meters, even though both loggers line up in terms of when each says the compressor turns off and turns on. The evaluators believe this is due to the evaluator's use of true root mean square loggers, while the facility's Amp values may be using analog Amp measurements. This can lead to differences of as much as 40%[3].

[3] Why is True-RMS So Important? | Rockwell Automation (archive.org) https://web.archive.org/web/20230315092408/https:/www.rockwellautomation.com/en-us/company/news/magazines/why-is-true-rms-so-important-.html

Figure 2-14. Ammonia compressor 3 Amps from facility trending system and Amps from evaluator loggers



The evaluators adjusted the Amp data from the facility's trending system by recognizing that the maximum observed trended values should correspond to the maximum observed evaluated values, and similarly for the minimum (non-zero) values. The points in between the minimum and maximum were scaled linearly. This scaling was necessary because the facility provided 9 months of Amp data, whereas the evaluator data was only available for 3 months. To use the full 9 months of data, a transformation had to be made to the trend data from the facility. Figure 2-15 shows the updated trend data, called "modeled evaluated Amps", that used the scaling transformation. To transform the data, the evaluator recognized that the minimum trended value of 183.8 Amps corresponds to 86.1 evaluated amps, and likewise, the 325.3 maximum trended amps corresponds to 296 evaluated Amps. A linear equation can be built from these 4 points in the form of:

$$C3\ Modeled\ Evaluated\ Amps\ = 1.48\ \times C3\ Trended\ Amps - 186.7$$

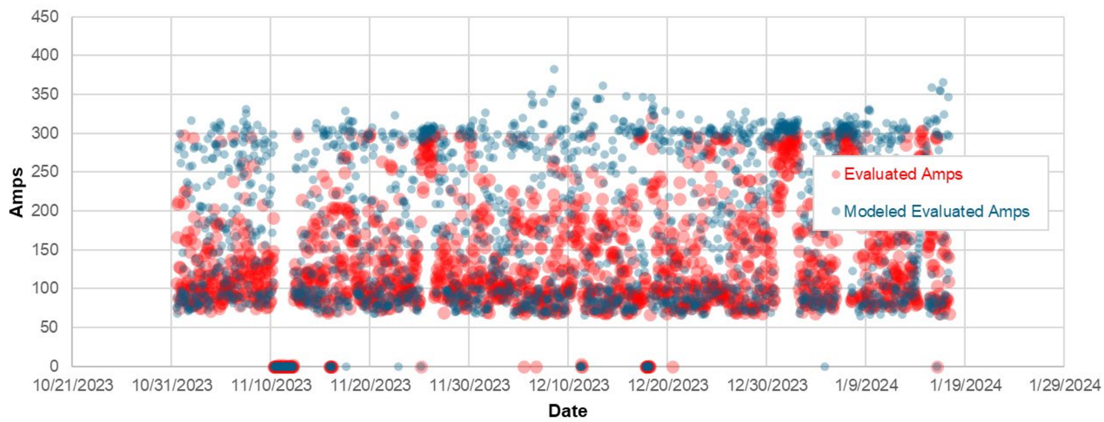
The transformations for the other equations are below:

$$C4\ Modeled\ Evaluated\ Amps\ = IF(C4\ Trended\ Amps < 10, C4\ Trended\ Amps, C4\ Trended\ Amps + 14.99)$$

$$C5\ Modeled\ Evaluated\ Amps\ = 0.814\ \times C5\ Trended\ Amps - 32.1956$$

Since the evaluator did not collect kW data on compressor 6, no transformation was performed. Since Compressor 1 and Compressor 2 did not contribute any savings to the tracking calculations, no transformation was performed on these compressors either.

Figure 2-15. Ammonia compressor 3 evaluated Amps, and trended Amps that have been transformed to "modeled evaluated Amps"



The same issue identified above for Compressor 3 was also an issue for the other Ammonia compressors. Figure 2-16 through Figure 2-19 show the original data, as well as the transformation made to the trend data for compressors 4, 5, and 6. Since no savings were claimed for Ammonia compressors 1 and 2 in the tracking analysis, this transformation was not performed on these ammonia compressors. Since kW was not measured on Ammonia compressor 6, the no transformation function was used.

Figure 2-16. Ammonia compressor 4 Amps from facility trending system and Amps from evaluator loggers

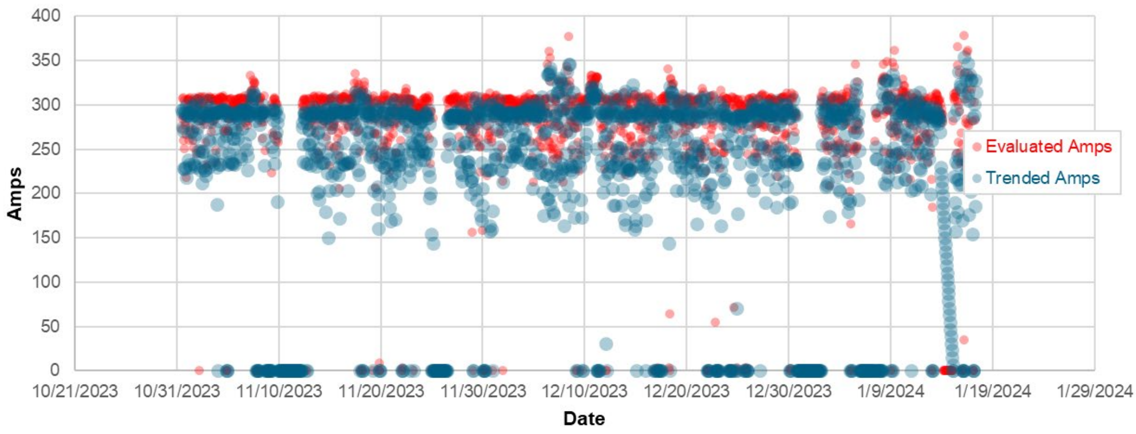


Figure 2-17. Ammonia compressor 4 evaluated Amps, and trended Amps that have been transformed to "modeled evaluated Amps"

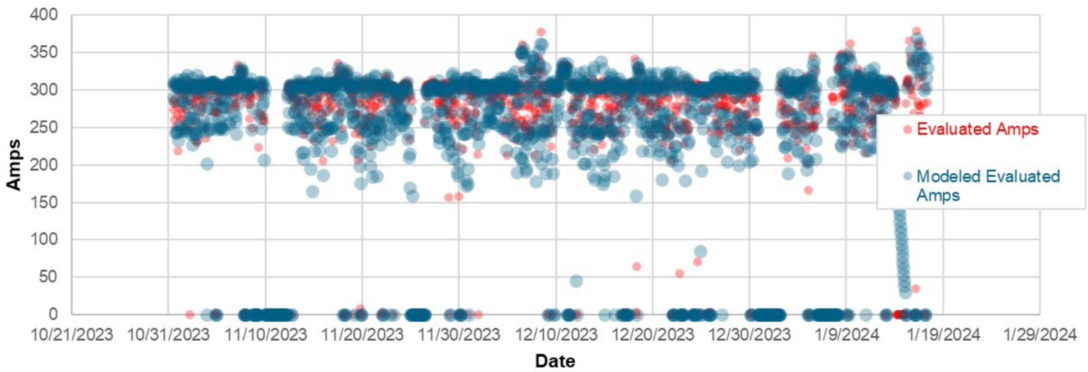


Figure 2-18. Ammonia compressor 5 Amps from facility trending system and Amps from evaluator loggers

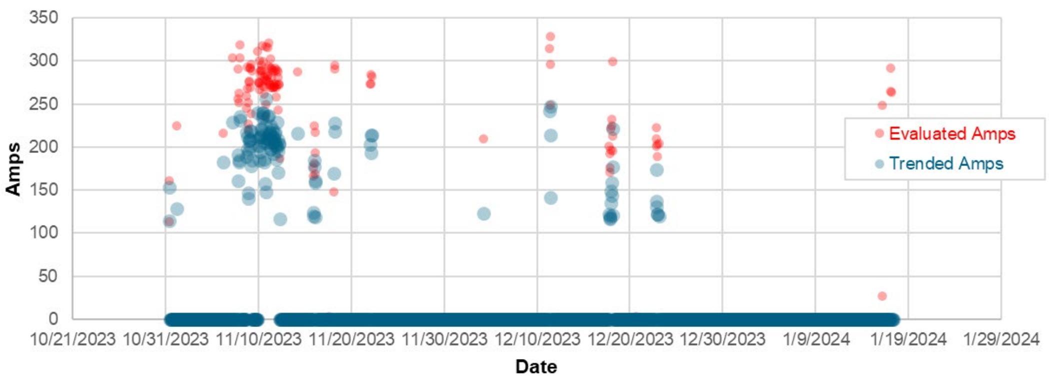


Figure 2-19. Ammonia compressor 5 evaluated Amps, and trended Amps that have been transformed to "modeled evaluated Amps"

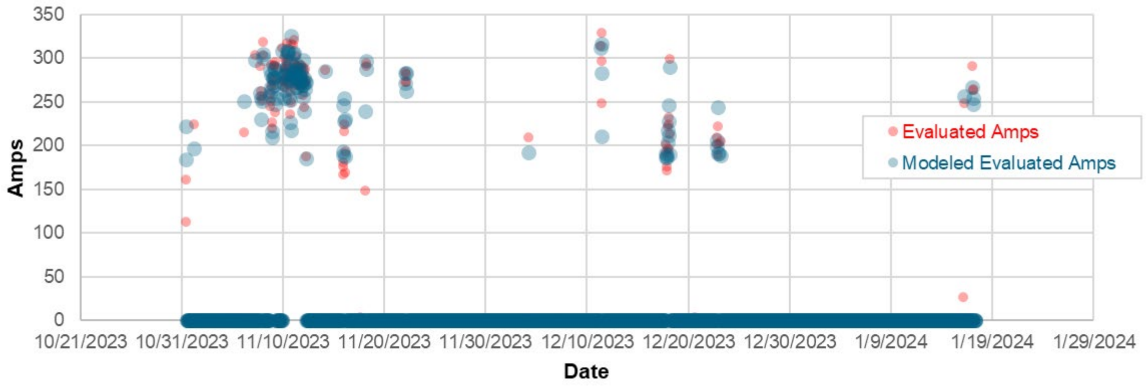
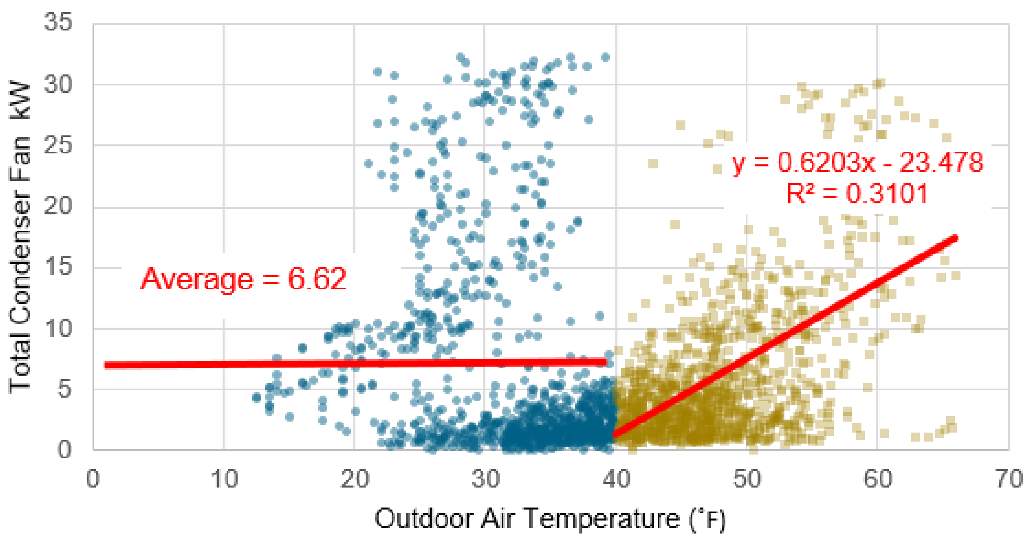


Figure 2-20 shows the total evaluator condenser fan kW vs. outdoor air temperature. Note that the data for condenser fan 3 was missing due to a logger malfunction, so the data from condenser fan 2 was duplicated to simulate condenser fan 3 kW. The average of the two was not taken because CF-1 was lower than CF-2, and the evaluator thought it was possible that CF-1 was not operating correctly. During the evaluation monitoring period, the average kW of condenser fan 1 was 1.79 kW, while the average of the condenser fan 2 was 2.32 kW. The average of all three fans is 6.42 kW.

Figure 2-20. Total evaluator measured condenser kW vs. outdoor air temperature



2.3 Evaluation methods and findings

This section describes the evaluator methods and findings.

2.3.1 Evaluation description of baseline

[11216625 / 11983237] (2) 177 HP VFD Air Compressors

The evaluator agrees with the ISP baseline of using load/no-load compressors for this project, which is consistent with the MA ISP document commonly referred to in RI for compressed air systems[4] . This reference recommends the baseline should be load/no-load with 4 gal/cfm of storage. The storage used in the tracking calculations assume 3 gal/cfm of storage in the baseline kW vs CFM performance curve, so the evaluators updated the baseline performance curve to follow the 4 gal/cfm curve.

[11413637 / 11983247] Ammonia Refrigeration Measures

65˚ minimum condensing setpoint, rather than 85˚ minimum condensing setpoint

The tracking calculations referred to the 2014 National Grid MA and RI baseline document for using the 85˚F minimum condensing temperature setpoint. The evaluators investigated this ISP baseline by interviewing 7 industrial refrigeration experts, and the conclusion from that conversation was that when this project was completed in 2020, the industry standard practice would have likely been to use a 70˚F minimum condensing temperature setpoint.

The average typical minimum condensing temperature setpoint in new construction industrial refrigeration projects as reported by the 7 subject matter experts is 71˚F. Two respondents said that the major original equipment manufacturers (OEMs) have 70˚F as the default minimum temperature setpoint in their packaged control software. Based on these responses, DNV recommends that the baseline minimum condensing temperature setpoint be 70˚F. Since the project used the baseline data approved at the time and conducted extra review of that baseline, the evaluated results use the 85˚ minimum condensing setpoint. Those results are shown in Table 1-2. These results shall be used in the expansion analysis and final report provided that Rhode Island Energy updates the 2014 National Grid Baseline document for the affected refrigeration measures based on the findings described in Appendix A and refers to those baseline practices for any similar projects going forward.

The evaluators used the 70˚F minimum condensing temperature to calculate secondary evaluation results using the updated baseline results as shown in Table 1-3.

VFD control for evaporative condenser, rather than two-speed fan control

The tracking calculations referred to the 2014 National Grid MA and RI baseline document for using two-speed fan control for the baseline for this measure. The evaluators questioned that baseline and investigated it by interviewing 7 industrial refrigeration experts, and the conclusion from that conversation was that when this project was completed in 2020, the industry standard practice would have been to use VFDs for evaporatively cooled condensers, not two-speed controls as was cited in the 2014 National Grid MA and RI baseline document.

Again, since the two speed fans was the approved baseline at the time and was further reviewed in 2020, the evaluators used two-speed fan control to calculate the evaluation results shown in Table 1-2. These results shall be used in the expansion analysis and final report provided that Rhode Island Energy updates the 2014 National Grid Baseline document for the affected refrigeration measures based on the findings described in Appendix A and refers to those baseline practices for any similar projects going forward.

The evaluators used the VFD fan control baseline to calculate the secondary evaluation results shown in Table 1.3, which show 0 savings for this measure, since baseline and installed case are identical.

Economized high-stage compressors with VFD control for lead ammonia compressor

The evaluators agreed with the baseline for economized high-stage compressors with VFD control for lead ammonia compressors. The evaluators investigated the baseline for this industrial refrigeration measure by speaking with 7 industrial refrigeration experts. The evaluators concluded that the baseline assumption used in the tracking calculations was reasonable after speaking with these 7 industrial refrigeration experts. Further details on this can be found in Appendix A.

[4] https://ma-eeac.org/wp-content/uploads/AirCompressors_ISP_Memo_final.pdf

2.3.2 Evaluation calculation method

[11216625 / 11983237] (2) 177 HP VFD Air Compressors

The evaluator calculations for the compressed air measure followed the following steps:

1.) The post-case air compressor kW data which was collected over the 103 days of the evaluation monitoring period was applied to an 8760-hour analysis by repeating the data from this hourly 103-day period approximately 3.54 times over the course of a full year and matching the weekday and hour of the evaluation monitoring data to the correct weekday and hour of the simulated 8,760 file.

The post-case kW data in the 8760 file was converted to post CFM data using the compressor's CAGI sheets, adjusted to 115 psi, which is the psi observed during the site visit. The kW vs. CFM curve is shown in Figure 2-3. Evaluation monitoring period heat map of kW compressor 1 kW use

Week #	Weekday	Date	Hour																							
			-	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
1	Holiday	1/1/2024	43	45	30	31	51	35	35	33	48	31	34	41	28	35	29	28	27	26	24	36	23	33	23	25
1	Tue	1/2/2024	35	23	54	24	37	61	70	45	55	54	29	55	62	52	50	47	47	42	43	40	43	37	34	30
1	Wed	1/3/2024	51	56	50	43	41	58	55	34	66	57	51	68	72	57	69	58	49	56	58	50	72	60	56	54
1	Thu	1/4/2024	58	56	72	69	72	89	95	73	94	95	68	70	73	64	51	39	26	33	34	26	28	31	25	28
1	Fri	1/5/2024	33	47	40	36	45	77	72	55	72	57	55	66	46	48	48	45	34	31	32	41	42	31	24	25
1	Sat	1/6/2024	30	42	43	38	31	24	24	30	38	46	45	40	50	30	41	42	44	40	38	36	30	30	30	30
2	Sun	1/7/2024	30	30	30	30	31	35	31	36	42	45	45	45	30	30	30	30	30	30	30	36	43	43	45	44
2	Mon	1/8/2024	49	47	39	33	46	58	70	57	73	72	44	72	107	51	36	39	47	41	37	45	33	47	44	30
2	Tue	1/9/2024	35	43	33	41	47	76	77	54	80	76	59	72	81	53	41	46	29	72	76	70	74	73	71	69
2	Wed	1/10/2024	71	76	66	70	86	91	71	50	70	60	42	71	70	40	47	35	42	51	41	43	31	27	25	27
2	Thu	1/11/2024	51	46	41	37	141	73	87	66	86	82	74	78	72	29	46	76	31	34	48	35	75	72	26	70
2	Fri	1/12/2024	33	36	51	25	67	68	73	92	74	71	56	57	58	34	32	42	73	43	43	32	64	50	44	69
2	Sat	1/13/2024	45	35	36	70	37	39	25	62	37	39	41	67	30	33	37	70	35	36	30	70	38	37	38	71
3	Sun	1/14/2024	33	28	27	68	33	38	47	37	26	32	36	72	38	33	71	28	33	30	27	72	29	25	63	36
3	Mon	1/15/2024	30	29	75	31	38	67	104	51	72	73	89	72	75	75	48	47	32	48	74	26	35	65	25	22
3	Tue	1/16/2024	25	71	25	39	26	90	72	51	54	74	84	73	70	78	45	36	28	78	44	28	71	41	35	40
3	Wed	1/17/2024	29	77	35	28	75	48	54	44	62	89	45	57	97	26	38	42	67	73	72	65	72	72	56	56
3	Thu	1/18/2024	73	71	73	73	72	90	96	74	86	93	76	63	56	70	30	31	31	70	39	25	26	73	23	23
3	Fri	1/19/2024	30	73	25	24	26	94	72	39	58	83	29	48	50	76	35	37	33	76	39	25	44	30	24	22
3	Sat	1/20/2024	30	73	58	72	66	0	38	69	35	26	25	24	72	26	27	72	29	24	25	24	73	73	73	72
4	Sun	1/21/2024	67	73	73	73	73	72	71	67	61	73	74	60	73	60	74	74	74	31	23	73	29	27	25	28
4	Mon	1/22/2024	73	27	34	33	72	53	55	35	95	49	38	67	61	40	32	45	72	38	37	63	39	38	24	26
4	Tue	1/23/2024	28	72	39	26	72	55	66	48	63	103	32	42	79	22	23	24	73	73	74	67	74	74	62	56
4	Wed	1/24/2024	73	69	74	65	73	83	89	72	92	84	74	89	82	73	80	74	72	78	76	70	76	59	68	67
4	Thu	1/25/2024	72	73	72	72	73	88	92	87	96	98	74	86	81	59	73	67	73	70	55	69	61	73	68	72
4	Fri	1/26/2024	70	72	72	59	72	89	101	86	96	93	77	89	92	69	72	73	56	60	23	31	32	25	64	26
4	Sat	1/27/2024	27	25	29	21	19	56	22	19	8	12	51	9	7	17	57	20	20	14	53	8	4	9	40	20
5	Sun	1/28/2024	16	20	13	56	20	20	19	59	20	20	13	18	56	5	14	8	19	56	72	72	71	72	67	60
5	Mon	1/29/2024	67	59	68	57	61	85	81	43	60	79	30	51	89	70	101	71	71	75	73	71	106	73	27	22
5	Tue	1/30/2024	72	31	33	68	27	57	73	47	96	72	85	82	133	69	79	72	102	73	72	101	72	55	65	23
5	Wed	1/31/2024	71	39	41	69	20	55	71	83	57	92	71	115	76	58	72	102	63	72	89	61	73	58	24	73
5	Thu	2/1/2024	31	23	72	21	52	92	128	80	58	58	81	75	53	37	36	35	72	43	42	71	29	29	35	21
5	Fri	2/2/2024	70	30	29	50	25	71	69	46	96	71	34	82	60	42	38	79	31	43	30	72	40	32	28	75
5	Sat	2/3/2024	43	37	71	24	24	29	34	56	21	17	19	58	19	20	19	56	21	21	19	56	19	20	20	19
6	Sun	2/4/2024	67	20	13	18	56	19	9	19	68	15	15	21	10	68	19	20	17	57	21	21	18	56	9	20
6	Mon	2/5/2024	27	27	47	21	26	72	102	36	71	72	78	69	64	71	33	32	37	70	30	31	35	73	23	28
6	Tue	2/6/2024	27	67	28	37	22	90	73	41	76	101	38	38	31	67	42	39	25	70	26	26	21	50	27	44
6	Wed	2/7/2024	32	70	29	24	74	56	55	53	60	94	42	55	94	35	40	40	40	75	33	25	71	29	21	22
6	Thu	2/8/2024	40	73	25	22	72	55	54	29	55	92	81	89	83	69	32	34	32	71	29	30	27	63	23	25
6	Fri	2/9/2024	57	71	42	25	20	92	55	37	70	101	50	59	92	38	42	40	38	72	31	25	71	26	24	26
6	Sat	2/10/2024	66	27	34	20	55	21	21	20	66															
44	Tue	10/31/2023										42	76	72	71	97	68	56	88	52	52	52	63	60	34	45
44	Wed	11/1/2023	74	30	47	44	71	95	96	86	69	101	65	65	55	48	79	91	91	88	89	84	89	83	84	92
44	Thu	11/2/2023	94	85	89	84	88	105	101	96	111	110	104	96	70	73	74	103	72	73	64	55	89	86	86	93
44	Fri	11/3/2023	100	88	91	94	84	103	107	105	99	75	77	102	72	74	57	71	56	44	79	38	42	47	56	83
44	Sat	11/4/2023	86	79	77	78	76	90	94	89	81	97	93	87	94	86	87	76	56	59	59	73	59	73	72	74
45	Sun	11/5/2023	72	58	58	72	73	60	58	60	15	23	29	73	21	20	55	21	18	56	14	21	55	11	20	62
45	Mon	11/6/2023	30	22	28	62	23	37	90	51	71	54	87	89	82	96	100	98	95	84	80	77	79	78	76	80
45	Tue	11/7/2023	87	76	78	83	79	93	105	93	86	71	71	69	103	51	69	39	49	82	53	76	49	46	44	47
45	Wed	11/8/2023	86	47	52	80	49	73	72	0	52	65	99	46	62	64	65	87	84	84	83	78	72	78	77	80
45	Thu	11/9/2023	82	76	81	83	80	90	98	88	95	102	96	89	73	54	72	86	62	43	59	74	50	41	76	41
45	Fri	11/10/2023	46	52	86	40	44	91	59	52	72	73	95	72	71	96	62	54	51	47	69	36	73	41	54	47
45	Sat	11/11/2023	48	81	52	58	75	34	38	31	77	76	74	75	70	72	73	73	73	67	74	72	74	74	73	74
46	Sun	11/12/2023	74	75	74	73	74	38	26	30	72	24	52	25	35	66	18	24	59	60	61	54	70	63	63	63
46	Mon	11/13/2023	73	68	64	67	66	89	72	33	90	65	37	54	61	71	46	47	73	25	34	27	73	49	38	61
46	Tue	11/14/2023	50	50	51	51	76	99	107	55	103	78	65	75	73	91	59	56	90	62	53	55	88	66	65	83
46	Wed	11/15/2023	55	65	55	79	51	80	109	73	80	78	71	100	102	94	88	86	81	91	87	80	83	78	76	76
46	Thu	11/16/2023	87	79	83	81	69	91	96	90	101	99	76	95	97	74	76	75	73	72	72	60	77	77	85	78
46	Fri	11/17/2023	94	83	87	92	77	109	108	53	80	57	85	55	66	51	53	58	86	73	57	83	57	49	44	35
46	Sat	11/18/2023	79	94	87	84	79	80	79	78	55	73	59	72	68	60	58	68	66	67	72	73	58	72	73	57
47	Sun	11/19/2023	72	61	72	72	58	58	72	72	25	25	29	36	72	30	55	23	33	59	23	18	58	24	22	72
47	Mon	11/20/2023	44	27	24	67	72	89	93	51	88	71	44	70	64	84	80	92	47	52	82	49	46	47	48	84
47	Tue	11/21/2023	41	39	74	52	21	61	96	83	105	104	80	102	100	93	90	55	50	54	73	20	66	24	29	65
47	Wed	11/22/2023	47	38	36	71	28	56	91	41	71	60	57	107	72	52	81	74	55	64	91	62	58	89	53	69
47	Holiday	11/23/2023	54	80	84	85	84	85	84	85	84	83	83	84	84	84	83	84	84	84	83	82	84	84	83	84
47	Fri	11/24/2023	94	89	90	88	85	110	118	94	115	108	83	89	89	76	75	74	73	79	80	77	78	77	76	77
47	Sat	11/25/2023	87	85	78	79	77	79	78	71	67	72	73	58	73	74	73	58	73	72	68	73	58	73	58	73
48	Sun	11/26/2023	57	73	69	73	64	73	65	73	72	73	72	58	66	73	26	37	23	24	72	27	30	71	32	26
48	Mon	11/27/2023	42	70	35	41	37	97	81	59	108	72	48	75	70	90	89	91	88	93	86	77	87	78	73	71
48	Tue	11/28/2023	81	83	84	79	79	111	113	94	116	79	63	71	109	61	49	48	50	82	49	49	91	56	53	39
48	Wed	11/29/2023	84	56	54	89	57	71	72	62	83	73	79	56	56	49	50	94	73	58	95	56	73	57	90	90
48	Thu	11/30/2023	103	96	92	98	94	105	79	57	116	73	62	76	83	89	53	48	80	43	51	39	83	84	80	83
48	Fri	12/1/2023	97	93	86	80	84	100	105	55	73	102	44	54	56	43	78	47	46	82	38	32	48	93	55	48
48	Sat	12/2/2023	98	61	65	55	44	85	39	39	71	36	27	25	63	27	28	68	26	34	27	24	72	32	36	73
49	Sun	12/3/2023	34	26	34	73	71	72	68	71	72	72	67	71	38	23	71	30	26	35	33	26	67	36	38	72
49	Mon	12/4/2023	34	44	49	74	36	70	111	50	74	78	59	100	98	93	54	54	87	52	52	49	92	53	55	85
49	Tue	12/5/2023	75	71	54	59	98	78	83	98	70	75	70	102	72	61	93	56	53	56	72	89	60	58	86	53
49	Wed	12/6/2023	66	72	98	98	89	109	122	89	106	114	89	75	68	65	36	39	35	47	79	33	42	89	53	52
49	Thu	12/7/2023	58	107	73	64	86	74	86	69	75	117	63	71	97	62	58	56	89	56	53	86	55	51	48	52
49	Fri	12/8/2023	93	59	53	83	49	76	82	85	103	112	96	94	93	77	54	56	90	56	55	57	96	73	57	85
49	Sat	12/9/2023	68	54	72	53	83	52	54	86	51	55	54	89	90	87	88	87	86	88	87	84	84	84	83	83
50	Sun	12/10/2023	84	84	83	84	84	83	45	51	52	83	50	56	44	83	68	46	83	50	51	51	62	83	85	85
50	Mon	12/11/2023	86	86	91				0	0	51	50	76	68	74	59	84	60	53	89	68	52	53	57	88	61
50	Tue	12/12/2023	71	91	67	53	57	103	82	66	115	75	57	56	78	95	87	56	87	57	54	55	55	84	54	53
50	Wed	12/13/2023	85	58	63	54	87	68	83	88	81	76	69	97	106	68	54	86	59	54	55	85	90	88	87	85
50	Thu	12/14/2023	88	96	92	85	90	101	110	67	84	103	58	79	82	81	93	88	90	92	88	87	91	82	87	84
50	Fri	12/15/2023	95	95	93	84	86	105	102	90	112	106	96	106	113	83	96	95	79	90	95	78	90	98	90	85
50	Sat	12/16/2023	91	105	96	91	83	88	94	74	96	94	72	94	85	73	75	74	75	76	82	82	75	74	74	74
51	Sun	12/17/2023	74	75	74	74	74	74	75	74	74	75	33	46	35	76	58	38	38	39	76	35	31	42	75	34
51	Mon	12/18/2023	36	39	72	42	0	38	38	34	72	70	53	67	69	65	51	54	44	91	72	53	69	56	86	54
51	Tue	12/19/2023	51	91	54	43	38	111	81	55	109	74	73	76	74	70	59	83	51	60	73	47	90	92	82	82
51	Wed	12/20/2023	79	82	74	77	79	105	106	40	71	120	61	74	82	58	80	48	29	76	35	35	29	73	73	60
51	Thu	12/21/2023	62	71	62	71	73	89	101	41	61	90	35	73	59	72	43	47	73	46	40	31	37	76	44	43
51	Fri	12/22/2023	73	51	47	47	75	86	56	85	70	71	49	61	84	69	74	80	73	77	78	72	75	75	70	72
51	Sat	12/23/2023	81	83	82	77	75	80	92	83	95	97	76	80	75	59	73	59	62	58	73	76	71	68	59	73
52	Sun	12/24/2023	61	54	66	60	74	52	72	58	55	59	24	14	24	45	20	15	60	21	15	66	18	18	54	21
52	Holiday	12/25/2023	25	70	16	19	56	58	59	51	58	63	20	58	28	19	59	24	18	58	23	18	55	50	58	64
52	Tue	12/26/2023	59	64	73	56	73	91	72	55	101	72	56	58	94	69	70	40	39	73	32	24	72	42	31	72
52	Wed	12/27/2023	35	57	41	31	42	50	58	82	77	74	53	103	69	40	73	41	47	60	71	44	51	40	73	34
52	Thu	12/28/2023	44	33	35	74	48	59	100	51	72	71	92	53	71	87	48	45	45	54	83	45	49	89	35	35
52	Fri	12/29/2023	44	39	39	31	72	97	72	87	71	75	52	63	100	75	78	44	37	77	56	51	52	52	49	78
52	Sat	12/30/2023	84	88	78	79	82	87	95	90	97	99	84	93	93	74	75	75	75	73	73	79	77	72	71	73
53	Sun	12/31/2023	74	73	72	71	73	72	72	72	73	72	72	50	32	32	45	44	37	33	45	42	32	31	41	35

2.) Figure 2-4. The baseline CFM was then calculated by adding 306.2 CFM to account for the purge flow in the baseline dryer and subtracting 29.8 CFM to account for post-case cooling airflow in the post-case dryer. The 306.2 cfm baseline purge flow is based on a spec sheet for an 1801 CFM heatless desiccant air dryer (CD 850+) whose specification sheet indicates that the air consumption is 17% of the 1801 total CFM. The installed blower purge dryer (BD+850) air dryer is also an 1800 cfm air dyer, that only uses 1.6% of the total CFM (or 28.8 CFM) according to its specification sheets, but also uses an additional 16.4 kW which also needs to be accounted for.

3.) The baseline kW data was calculated from the baseline CFM data using the air compressor curve from the compressed air challenge, adjusted to 115 psi, based on the curve shown in Figure 2-.

4.) The post case total kW was based on adding the post-case air compressor kW (from step 1) to the post-case dryer kW value of 16.4 kW (from specification sheets) for each hour of the year.

5.) The energy savings for each hour of the year was calculated from the difference in the total baseline and post-case kW.

[11413637 / 11983247] Ammonia Refrigeration Measures

65˚ minimum condensing setpoint, rather than 85˚ minimum condensing setpoint

1. The post-case compressor % on data from Figure 2-7 was applied to an 8760 file for each of the 6 refrigeration compressors, and then the Amps vs outdoor air temperature relationships (when on) for each of the 6 compressors were applied to develop an estimated Amp value for each Ammonia compressor for each hour of the year. A random number generator function was used to appropriately assign whether each compressor would be on or off for each hour of the year, based on the percentage of time that ammonia compressor had been on during the evaluation monitoring period.

2. An adjustment was made to the trended Amperage data in the 8670 Amp data from the previous step based on the data from Figure 2-14 through Figure 2-19.

3. The Amp data was converted to kW data using data collected from the evaluator's kW meters. The evaluator's kW meters measures and reports Amps, Volts, PF, and kW on each phase, as well as the total Amps and kW including all three phases. So, the adjusted Amp data from step 2 above was converted to kW data by using the relationship between Average Amps (from the evaluator's kW meters), and total kW (from the evaluator's kW meters). Compressor 2, 5 and 6 did not use evaluator kW meters, only evaluator Amp meters, so for those, the kW vs Amp relationships referred to the evaluator kW data from compressors 1, 3, and 4, respectively.

4. The post-case suction temperature and condensing temperature, from the data shown in Figure 2-5 and Figure 2-6 was applied to the 8760 file, based on the TMY3 weather data for the nearest weather location.

5. The post-case kW data for each of the compressors was converted to tonnage data, using the performance data from the Ammonia compressor manufacturers. Those performance curves are described in Section 2.1, and are a function of the kW, suction temperature, and condensing temperature.

6. The baseline kW for each of the compressors was calculated for a 70˚F minimum condensing temperature, as well as an 85˚F minimum condensing temperature. The 85˚F value was used for the primary results as explained above and the 70˚F value was used for the secondary updated baseline results, while. The calculation was done using the performance curves from the Ammonia compressor manufacturers, as described in Section 2.1, and are a function of the tonnage, suction temperature, and condensing temperature.

For the high-stage compressors, the non-economized compressors with slide-valves for compressors 3,4,5, and 6 were used.

7. Savings were calculated by taking the difference between the baseline kW and the post case kW for the 8760 file.

VFD control for evaporative condenser, rather than two-speed fan control

1. The post-case data from Figure 2-20 that relates total post-case condenser fan kW to outdoor air temperature was applied to TMY3 data in an 8,760 file to estimate the total post-case annual condenser fan kWh.
2. For the energy savings for this measure, the baseline energy was calculated by using the post-case performance curves for the condenser fans described in Section 2.1 that relate condenser capacity in Btu/h of heat rejection to condenser fan kW for the case where the baseline condenser fan would have been controlled by a two speed motor. So, the post-case performance curves were first applied to the post-case kW data to determine a btu/h capacity value for each hour of the year, and then the baseline kW was calculated by applying the baseline capacity vs. kW curve to the capacity calculated using the post-case data. The difference between the two resulted in the energy savings.
3. For the secondary energy savings calculation for this measure, the savings are zero, since the additional ISP research discussed in Appendix A, found that industry standard practice when this project was installed was to have installed a VFD on the condenser fans.

Economized high-stage compressors with VFD control for lead ammonia compressor

1. The same steps for calculating the energy savings for the "*65˚ minimum condensing setpoint, rather than 85˚ minimum condensing setpoint"* that are described above were followed for this measure, only the performance curve for compressor 3 was updated to reflect the ammonia compressor with a VFD on it and economized, and the performance curves for compressors 4, 5, and 6 were updated from non-economized versions to economized versions.

3 FINAL RESULTS

[11216625 / 11983237] (2) 177 HP VFD Air Compressors

The parameters impacting the analysis for the compressed air measure are summarized in Table 3-1.

Table 3-1. Summary of key parameters

Parameter	BASELINE		PROPOSED / INSTALLED	
	Tracking Value(s)	Evaluation Value(s)	Tracking Value(s)	Evaluation Value(s)
Total kWh	904,029	1,076,368	519,929	668,157
Lead Compressor kWh	892,130	1,072,332	405,588	524,903
Lead Compressor Hours	6972	8735	6972	8735
Lead Compressor Average kW	127.96	122.76	58.17	60.09
Lead Compressor Average kW/CFM	0.201	0.200	0.158	0.179
Lead Compressor Average CFM	635	613	369	336
Trim Compressor kWh	11,900	4,036	0	0
Trim Compressor Hours	84	102	0	0
Trim Compressor Average kW	142	39.57	0	0
Trim Compressor Average kW/CFM	0.166	0.39	N/A	N/A
Trim Compressor Average CFM	856	41.07	0	0
Dryer kWh	0	0	114,341	143,254
Dryer kW	0	0	16.4	16.4
Dryer Hours	0	0	6972	8735
Dryer Average CFM	306.17	306.17	28.8	28.8

[11413637 / 11983247] Ammonia Refrigeration Measures

The key parameters impacting the prospective analysis for the refrigeration measures are summarized in

Table 3-2.

Table 3-2. Summary of key parameters – prospective results

Parameter	Component	Baseline	Tracking				Evaluator				
			ECM 1	ECM2	ECM3	Total	Baseline	ECM 1	ECM2	ECM3	Total
Annual Savings	Comp 1		0	0	0	0		0	0	0	0
	Comp 2		0	0	0	0		0	0	0	0
	Comp 3		343,522	0	-240,683	102,838		248,355	0	93,807	342,162
	Comp 4		451,915	0	112,874	564,789		352,948	0	18,231	371,179
	Comp 5		196,636	0	399,274	595,910		43,769	0	402	44,171
	Comp 6		0	0	0	0		3,121	0	-368	2,753
	Condenser Fans		-81,824	140,482	4,660	63,317		0	101,428	0	101,428
	Total		910,249	140,482	276,124	1,326,854		648,193	101,428	112,071	861,693
Annual kWh	Comp 1	345,543	345,543	345,543	345,543		297,805	297,805	297,805	297,805	
	Comp 2	0	0	0	0		154,920	154,920	154,920	154,920	
	Comp 3	1,884,092	1,540,570	1,540,570	1,781,253		1,756,118	1,507,763	1,507,763	1,413,956	
	Comp 4	1,996,587	1,544,672	1,544,672	1,431,798		1,966,429	1,613,481	1,613,481	1,595,250	
	Comp 5	595,910	399,274	399,274	0		206,754	162,985	162,985	162,583	
	Comp 6	0	0	0	0		17,998	14,877	14,877	15,246	
	Condenser Fans	227,340	309,164	168,683	164,023		208,771	208,771	107,342	107,342	
	Total	5,049,472	4,139,223	3,998,741	3,722,617		4,608,796	3,960,602	3,859,174	3,747,103	
Annual Hours	Comp 1	8,760	8,760	8,760	8,760		4,800	4,800	4,800	4,800	
	Comp 2	0	0	0	0		3,165	3,165	3,165	3,165	
	Comp 3	8,760	8,760	8,760	8,760		8,643	8,643	8,643	8,643	
	Comp 4	8,760	8,760	8,760	8,760		7,745	7,745	7,745	7,745	
	Comp 5	4,571	4,571	4,571	4,571		1,081	1,081	1,081	1,081	
	Comp 6	0	0	0	0		163	163	163	163	

Parameter	Component	Baseline	Tracking				Evaluator				
			ECM 1	ECM2	ECM3	Total	Baseline	ECM 1	ECM2	ECM3	Total
	Condenser Fans	8,760	8,760	8,760	8,760		8,760	8,760	8,760	8,760	
Average kW	Comp 1	39.4	39.4	39.4	39.4		62	62	62	62	
	Comp 2	0.0	0.0	0.0	0.0		49	49	49	49	
	Comp 3	215.1	175.9	175.9	203.3		203	174	174	164	
	Comp 4	227.9	176.3	176.3	163.4		254	208	208	206	
	Comp 5	130.4	87.3	87.3	0.0		191	151	151	150	
	Comp 6	0.0	0.0	0.0	0.0		110	91	91	94	
	Condenser Fans	26.0	35.3	19.3	18.7		24	24	12	12	
Average Tons	Comp 1	27	27	27	27		12	12	12	12	
	Comp 2	0	0	0	0		8	8	8	8	
	Comp 3	304	337	314	384		275	275	275	275	
	Comp 4	305	298	298	284		364	364	364	364	
	Comp 5	113	64	108	0		248	248	248	248	
	Comp 6	0	0	0	0		104	104	104	104	
	Condenser Fans	808	777	777	767		782	782	782	782	
Average kW/ton	Comp 1	1.44	1.44	1.44	1.44		5.18	5.18	5.18	5.18	
	Comp 2	N/A	N/A	N/A	N/A		6.06	6.06	6.06	6.06	
	Comp 3	0.71	0.52	0.56	0.53		0.74	0.63	0.63	0.60	
	Comp 4	0.75	0.592	0.592	0.575		0.770	0.607	0.607	0.606	
	Comp 5	1.15	1.37	0.81	0.63		1.07	0.88	0.88	0.90	
	Comp 6	N/A	N/A	N/A	N/A		0.03	0.03	0.02	0.02	

Parameter	Component	Baseline	Tracking				Evaluator				
			ECM 1	ECM2	ECM3	Total	Baseline	ECM 1	ECM2	ECM3	Total
	Condenser Fans	0.03	0.05	0.02	0.02		0.03	0.03	0.02	0.02	

The key parameters impacting the retrospective analysis for the refrigeration measures are summarized in Table 4-5

Table 3-3. Summary of key parameters – retrospective results

Parameter	Component	Baseline	Tracking				Evaluator				
			ECM 1	ECM2	ECM3	Total	Baseline	ECM 1	ECM2	ECM3	Total
Annual Savings	Comp 1		0	0	0	0		0	0	0	0
	Comp 2		0	0	0	0		0	0	0	0
	Comp 3		343,522	0	-240,683	102,838		3,086	0	93,807	96,893
	Comp 4		451,915	0	112,874	564,789		10,334	0	18,231	28,565
	Comp 5		196,636	0	399,274	595,910		4,184	0	402	4,586
	Comp 6		0	0	0	0		513	0	-368	145
	Condenser Fans		-81,824	140,482	4,660	63,317		0	0	0	0
	Total		910,249	140,482	276,124	1,326,854		18,118	0	112,071	130,189
Annual kWh	Comp 1	345,543	345,543	345,543	345,543		297,805	297,805	297,805	297,805	
	Comp 2	0	0	0	0		154,920	154,920	154,920	154,920	
	Comp 3	1,884,092	1,540,570	1,540,570	1,781,253		1,510,849	1,507,763	1,507,763	1,413,956	
	Comp 4	1,996,587	1,544,672	1,544,672	1,431,798		1,623,815	1,613,481	1,613,481	1,595,250	
	Comp 5	595,910	399,274	399,274	0		167,169	162,985	162,985	162,583	
	Comp 6	0	0	0	0		15,390	14,877	14,877	15,246	
	Condenser Fans	227,340	309,164	168,683	164,023		107,342	107,342	107,342	107,342	
	Total	5,049,472	4,139,223	3,998,741	3,722,617		3,877,292	3,859,174	3,859,174	3,747,103	

Parameter	Component	Baseline	Tracking				Evaluator				
			ECM 1	ECM2	ECM3	Total	Baseline	ECM 1	ECM2	ECM3	Total
Annual Hours	Comp 1	8,760	8,760	8,760	8,760		4,800	4,800	4,800	4,800	
	Comp 2	0	0	0	0		3,165	3,165	3,165	3,165	
	Comp 3	8,760	8,760	8,760	8,760		8,643	8,643	8,643	8,643	
	Comp 4	8,760	8,760	8,760	8,760		7,745	7,745	7,745	7,745	
	Comp 5	4,571	4,571	4,571	4,571		1,081	1,081	1,081	1,081	
	Comp 6	0	0	0	0		163	163	163	163	
	Condenser Fans	8,760	8,760	8,760	8,760		8,760	8,760	8,760	8,760	
Average kW	Comp 1	39.4	39.4	39.4	39.4		62	62	62	62	
	Comp 2	0.0	0.0	0.0	0.0		49	49	49	49	
	Comp 3	215.1	175.9	175.9	203.3		175	174	174	164	
	Comp 4	227.9	176.3	176.3	163.4		210	208	208	206	
	Comp 5	130.4	87.3	87.3	0.0		155	151	151	150	
	Comp 6	0.0	0.0	0.0	0.0		94	91	91	94	
	Condenser Fans	26.0	35.3	19.3	18.7		12	12	12	12	
Average Tons	Comp 1	27	27	27	27		12	12	12	12	
	Comp 2	0	0	0	0		8	8	8	8	
	Comp 3	304	337	314	384		275	275	275	275	
	Comp 4	305	298	298	284		364	364	364	364	
	Comp 5	113	64	108	0		248	248	248	248	
	Comp 6	0	0	0	0		104	104	104	104	

Parameter	Component	Baseline	Tracking				Evaluator				
			ECM 1	ECM2	ECM3	Total	Baseline	ECM 1	ECM2	ECM3	Total
	Condenser Fans	808	777	777	767		782	782	782	782	
Average kW/ton	Comp 1	1.44	1.44	1.44	1.44		5.18	5.18	5.18	5.18	
	Comp 2	N/A	N/A	N/A	N/A		6.06	6.06	6.06	6.06	
	Comp 3	0.71	0.52	0.56	0.53		0.64	0.63	0.63	0.60	
	Comp 4	0.75	0.592	0.592	0.575		0.623	0.607	0.607	0.606	
	Comp 5	1.15	1.37	0.81	0.63		0.91	0.88	0.88	0.90	
	Comp 6	N/A	N/A	N/A	N/A		0.02	0.02	0.02	0.02	
	Condenser Fans	0.03	0.05	0.02	0.02		0.02	0.02	0.02	0.02	

3.1 Explanation of differences

The rolled-up deviations at the project level for the primary analysis are shown in Table 3-4.

Table 3-4. Summary of project level deviations for all measures combined

Measure	Discrepancy	Parameter	Impact of deviation	Discussion of deviations
[11216625 / 11983237] (2) 177 HP VFD Air Compressors	Operation	Change in compressor efficiency	-7.78%	Decreased savings –The average change in the kW/CFM between the baseline and post-case was found to be 0.021 kW/CFM, compared to 0.044 estimated by the tracking analysis. The change in the kW/CFM is driven by the amount of time the compressor spends at different loads which fall at different parts of the of compressor efficiency curve.
[11216625 / 11983237] (2) 177 HP VFD Air Compressors	Operation	Operating profile	10%	Increased savings – Evaluator found that the lead compressor operated 8,735 hours/year which is 25% more than the 6,972 hours estimated in the tracking analysis.
[11413637 / 11983247] Ammonia Refrigeration 65˚ minimum condensing setpoint measure	Operation	Minimum Condensing Temperature	-13.07%	Decreased savings – evaluators found post-case minimum condensing temperature to be 71˚F, rather than the 65˚used in the tracking calculations.
[11413637 / 11983247] Ammonia Refrigeration 65˚ minimum condensing setpoint measure	Operation	Ton-hours of cooling	-5.66%	Decreased savings – tracking calculations estimated 6,010,000 annual ton-hours, while the evaluator found approximately 5,570,000 annual ton hours.
[11413637 / 11983247] Ammonia Refrigeration VFD control for evaporative condenser	Operation	Observed change in average operating condenser fan kW	-2.83%	Decreased savings – evaluators found that the average condenser fan kW reduced from 24 kW to 12 kW with the use of VFDs rather than two-speed fan control, whereas the tracking calculations estimated that the fan kW would reduce from 35.3 kW to 19.3 kW
[11413637 / 11983247] Ammonia Refrigeration Economized high-stage	Operation	Change in average compressor efficiency	-5.70%	Decreased savings – evaluator found that the average kW/ton efficiency improvement between the baseline and post-case for this measure went from 0.267 kW/ton to 0.258 kW/ton, for an improvement of 0.009 kW/ton, whereas the tracking savings estimated that the average kW/ton efficiency improvement went from 0.275 kW/ton to 0.256 kW/ton, for an improvement of 0.020 kW/ton. This is driven by the amount of time the refrigerant compressors spend at different loads which

Measure	Discrepancy	Parameter	Impact of deviation	Discussion of deviations
compressors with VFD control				fall at different parts of the of compressor efficiency curves.
[11413637 / 11983247] Ammonia Refrigeration Economized high-stage compressors with VFD control	Operation	Ton-hours of cooling	-0.98%	Decreased savings – tracking calculations estimated 6,010,000 annual ton-hours, while the evaluator found approximately 5,570,000 annual ton hours.

3.2 Lifetime savings

The evaluator classified the measures as new construction. The evaluator calculated applicant and evaluated lifetime savings values using the following formula:

$$\mathrm{LAGI} = \mathrm{FYS} \times [\ \mathrm{outyear}\ \% \times (\mathrm{EUL})]$$

where:

LAGI =	lifetime adjusted gross impact (therms)
FYS =	first year savings (therms)
EUL =	measure life (years)
outyear % =	100% for this single baseline measure

The evaluated lifetime savings are lower than the tracking lifetime savings because the evaluated first year savings are lower than the tracking first year savings.

The total primary lifetime savings summary for all measures are shown in Table 3-5. These are the savings that use the baseline assumptions that reference the 2014 National Grid Baseline Document.

Table 3-5. Application ID: [11216625 / 11983237] and [11413637 / 11983247] – Total Primary Lifetime savings summary

Measure	Factor	Tracking	Application	Evaluator
[11216625 / 11983237] (2) 177 HP VFD Air Compressors	Lifetime savings (kWh)	5,761,500	5,761,500	6,123,165
	First-year savings (kWh)	384,100	384,100	408,211
	Measure lifetime (years)	15	15	15
	Baseline classification	New Construction / ISP	New Construction / ISP	New Construction / ISP
[11413637 / 11983247] Ammonia Refrigeration	Lifetime savings (kWh)	13,268,540	13,268,540	8,616,930
	First-year savings (kWh)	1,326,854	1,326,854	861,693

Measure	Factor	Tracking	Application	Evaluator
	Measure lifetime (years)	10	10	10
	Baseline classification	New Construction / ISP	New Construction / ISP	New Construction / ISP
Total	Lifetime savings (kWh)	19,030,040	19,030,040	14,740,095
	First-year savings (kWh)	1,710,954	1,710,954	1,269,904
	Measure lifetime (years)	11.12	11.12	11.61
	Baseline classification	New Construction / ISP	New Construction / ISP	New Construction / ISP

The total secondary lifetime savings summary for all measures are shown in Table 3-6. The secondary evaluation results shall be used in the expansion analysis and final report provided that Rhode Island Energy does not update the 2014 National Grid Baseline document for the affected refrigeration measures based on the findings described in Appendix A and does not refer to those baseline practices for any similar projects going forward.

Table 3-6. Application ID: [11216625 / 11983237] and [11413637 / 11983247] – Total Secondary Lifetime savings summary

Measure	Factor	Tracking	Application	Evaluator
[11216625 / 11983237] (2) 177 HP VFD Air Compressors	Lifetime savings (kWh)	5,761,500	5,761,500	6,123,165
	First-year savings (kWh)	384,100	384,100	408,211
	Measure lifetime (years)	15	15	15
	Baseline classification	New Construction / ISP	New Construction / ISP	New Construction / ISP
[11413637 / 11983247] Ammonia Refrigeration	Lifetime savings (kWh)	13,268,540	13,268,540	1,301,890
	First-year savings (kWh)	1,326,854	1,326,854	130,189
	Measure lifetime (years)	10	10	10
	Baseline classification	New Construction / ISP	New Construction / ISP	New Construction / ISP
Total	Lifetime savings (kWh)	19,030,040	19,030,040	7,425,055
	First-year savings (kWh)	1,710,954	1,710,954	538,400
	Measure lifetime (years)	11.12	11.12	13.79
	Baseline classification	New Construction / ISP	New Construction / ISP	New Construction / ISP

3.3 Ancillary impacts

There are no ancillary impacts for this project.

4 APPENDIX A
EVALUATOR INDUSTRIAL REFRIGERATION BASELINE INVESTIGATION REPORT FOR RICE2022N013

4.1 Executive Summary

The purpose of this research was to investigate the baselines used by the applicant in estimating savings and to determine how efficient the installed measures are relative to refrigeration standard practices. The methodology involves speaking with knowledgeable experts on the design of large ammonia-based new construction industrial refrigeration projects.

The scope of work (described on November 29, 2023, in an email from Rick Boswell to Dave Jacobson and Ann Clarke), states that the process will involve interviewing approximately 5 industrial refrigeration professionals to determine the standard practice baseline for the three measures listed below. The project that initiated this research effort, RICE22N013, was completed in 2020, while this research effort is occurring in 2024. Interviewees were initially asked about current standard practice design of new construction ammonia refrigeration systems and were subsequently asked how these standard practices have changed over the past 0-5 and 5-10 years. These responses were used to inform DNV's recommended baselines for each of the three measures.

Based on DNV's interviews with (7) industrial refrigeration experts, DNV recommends that the baselines shown in Table 4-1 be used for the three installed measures until such a point that additional information suggests otherwise. Responses to the follow up question about changes over the past 0-5 and 5-10 years indicates that standard practice in new construction ammonia refrigeration design, when talking about the use of VFDs on evaporative condenser fans, and the average minimum condensing temperature, has not changed substantially between 2020, when this project was completed, and now, 2024.

Table 4-1. Recommended Baselines for RICE22N013

ECM #	Measure Description according to RICE22N013	Proposed Baseline Based on Surveys of Experts	Supporting Reason for Proposed ISP Baseline
ECM 1	65˚ minimum condensing setpoint, rather than 85˚ minimum condensing setpoint[5]	70˚ F minimum condensing temperature	Default minimum setpoint in multiple compressor manufacturer control software is 70 ˚F according to two respondents. Average value of all respondents is 71˚F.
ECM 2	VFD control for evaporative condenser, rather than two-speed fan control	VFD control for evaporative condenser	Large majority (6/7) respondents stated that proportion of facilities that would install VFDs to control evaporative condensers is >75%.
ECM 3	Economized high-stage screw compressors with VFD control for lead compressor, rather than non-economized high stage screw compressors with slide-valve for lead compressor.	Non-economized high stage compressors, with slide-valve control for lead compressor.	No strong consensus among respondents. Average of respondents estimate for proportion of NC projects with economized high-stage screw compressors is 59%, and proportion of NC projects with VFDs on lead compressor is 53%.

5 This was based on the 2014 RI baseline document: 2014-baseline-document-for-ma-and-ri.pdf (nationalgridus.com) according to an email from Jim Boyco to Nick Wojcik on November 16, 2020.

4.2 Introduction

In the program year 2022 custom electric evaluation, new construction project RICE22N013 involved the installation of three (3) energy efficiency measures at a refrigerated meat processing facility. The total project savings claimed was 1,326,854 kWh. The measure descriptions and each measure's savings contribution as a percent of the total claimed savings are shown below.

- ECM1: 65˚ F minimum condensing setpoint, rather than 85˚ minimum condensing setpoint for new ammonia refrigeration system **[72%].**
- ECM2: VFD control for evaporative condenser, rather than two-speed fan control **[13%].**
- ECM3: Economized high-stage compressors with VFD control for lead compressor, rather than non-economized high stage compressors with slide-valve for lead compressor [**15%].**

Table 4-2 shows the title, organization, and relevant experience of the individuals who were interviewed as part of this research.

Table 4-2. Title, organization, and experience of interviewed refrigeration experts

[Respondent ID]	Title	Organization	Relevant Experience
[A]	Director of Engineering	Lineage, A Food Processing and Manufacturing, and Cold Storage Warehousing Company	16 years' experience in industrial refrigeration. Worked for Cargill, Danfoss, and design build contractor (Coval Group) for 9 years. Current Chair of the IIAR[6] Sustainability Committee. Has worked on hundreds of industrial refrigeration projects.
[B]	President, Owner	Mr. Ammonia Refrigeration	21 years of experience in industrial refrigeration. Owner of a design build firm specializing in large industrial refrigeration sites. Director at large of the IIAR Natural Refrigeration Foundation Has worked on (designed and/or provided service to) over 200 industrial refrigeration facilities (both existing and new construction).
[C]	VP of Engineering	Northstar Refrigeration	7 years of experience working for a design build firm specializing in large industrial refrigeration. Has worked on 15 new construction industrial refrigeration projects in past 5 years.
[D]	Assistant Director	Industrial Refrigeration Consortium at the University of Wisconsin	25 years of experience with industrial refrigeration. Board of Director at IIAR Has worked on over 100 industrial refrigeration projects.

[6] International Institute of All-Natural Refrigeration. The mission of the IIAR is to provide advocacy, education, and standards for the benefit of the global community in the safe and sustainable design, installation and operation of ammonia and other natural refrigerant systems.

[Respondent ID]	Title	Organization	Relevant Experience
			Co-Author/Contributor to ASHRAE's Guide for Sustainable Refrigerated Facilities and Refrigeration Systems
[E]	Project Engineer	Cascade Energy	16 years of experience in industrial facilities. Has worked on dozens of industrial refrigeration projects.
[F]	Managing Director	Scantec Refrigeration	40 years' experience in industrial refrigeration system design Board of Director at IIAR Technical Reviewer for ASHRAE Guide for Sustainable Refrigerated Facilities and Refrigeration Systems
[G]	Research Engineer	Industrial Refrigeration Consortium at the University of Wisconsin	12 years' experience in industrial refrigeration Co-Author/Contributor to ASHRAE's Guide for Sustainable Refrigerated Facilities and Refrigeration Systems Trainer on courses related to Ammonia Refrigeration Systems

4.3 Results

This section presents the findings from the interviews with the industrial refrigeration subject matter experts.

4.3.1 Recommended Baseline Around Minimum Condensing Temperature Setpoint

This section describes the results of the interview question with the industrial refrigeration experts having to do with typical practice around minimum condensing temperature setpoints in large Ammonia refrigeration systems. The tracking calculations in RICE22N013 only mention a lower condensing setpoint, and do not state that floating head pressure controls based on wet-bulb temperatures were installed. Table 4-3 shows the results of the interviews.

The average typical minimum condensing temperature setpoint in new construction industrial refrigeration projects as reported by the 7 subject matter experts is 71˚F. Two respondents said that the major original equipment manufacturers (OEMs) have 70˚F as the default minimum temperature setpoint in their packaged control software. Based on these responses, DNV recommends that the baseline minimum condensing temperature setpoint be 70˚F.

For project RICE22N013, the claim in the project calculation is that standard practice would be to set the minimum condensing temperature to 85˚F[7]. Reducing the standard practice baseline minimum condensing temperature setpoint from 85˚ F to 70˚F would decrease the energy savings for this measure by approximately 50-75%.

Sixty-two percent (62%) of subject matter experts stated that between 15% and 100% of new construction projects on large (>1,000 HP) industrial refrigeration systems would use floating head pressure controls. For the respondents who

[7] Based on the 2014 RI baseline document: 2014-baseline-document-for-ma-and-ri.pdf (nationalgridus.com) according to an email from Jim Boyco to Nick Wojcik on November 16, 2020.

said the number of new construction projects would be lower (i.e. 15%), their explanation was that “most projects (85%) use a fixed setpoint between 100 – 130 psig (64 - 77˚ F) or 100 – 140 psig (64 - 81˚ F), rather than a floating setpoint that changes as a function of a wet-bulb temperature measurement. On hot days, the systems with the fixed setpoint control can’t achieve the setpoint and the actual head pressure floats higher even if they aren’t using wet-bulb temperature control.”

Table 4-3. Head pressure controls and minimum condensing temperature setpoint results

Respondent ID	Percentage of Similar New Construction Projects Using Floating Head Pressure Controls	Average Minimum Condensing Temperature Setpoint in Standard Practice Design	Notes
[A]	75-95%	70˚ F	Almost an industry standard at this point. All the major compressor original equipment manufacturers (OEMs), such as GEA, Frick, Unity, Danfoss, and M&M, have floating head pressure controls built in. The only small added cost to incorporate this feature is the wiring for the wet-bulb sensor. The default minimum condensing temperature in GEA or Frick's software is 70˚ F. Some design engineers, if they are working for certain clients like Nestle or General Mills, who have their own specifications, state that the minimum condensing temperature must be 60˚ F. This would be the more energy efficient design, whereas 70˚ would be more like the normal standard practice. When design engineers are selecting compressors, the default minimum condensing temperature is 70 to 75 deg F. This is used to size the oil separator. Some engineers and end-users specify down to 60 deg F which typically results in larger oil separator. The percentage that use the default of 70 to 75 deg F is around 90%. Some manufacturers will "okay" going lower, typically down to around 105 psig to 110 psig. The challenge is that there currently not published ppm of oil in the discharge gas to determine how much oil ppm in the discharge gas is acceptable
[B]	10%	64 - 73˚ F	Danfoss, Frick, and GEA will have the option in their software, but it is up to the operators to take advantage of it. It is easier to achieve the same result by using a fixed setpoint that their system may not reach, rather than a floating setpoint that depends on a wet-bulb sensor which is prone to

Respondent ID	Percentage of Similar New Construction Projects Using Floating Head Pressure Controls	Average Minimum Condensing Temperature Setpoint in Standard Practice Design	Notes
			failure. It is better to set the minimum setpoint to 100 psig (64˚F) and let the condenser fans try to make that temperature year-round. In the winter, it will approach that setpoint, but in the summer, it will operate at a higher temperature, while still trying to reach that setpoint. 100-120 psig (64 - 73˚ F) is the range of typical setpoints used.
[C]	95%	70 - 79˚ F	When a new construction plant, or a controls upgrade is done usually they will install a floating head pressure control. Unless there is a wide varying load they might opt to not.
[D]	100%	75 - 80˚ F	With industrial refrigeration, everybody does it. Many people have it, but don't do it all year long. They may not set it as low as they could. Designs will allow it, but operation staff will not, because of the risk of portions of the system not working properly, such as defrost. Most facilities will run at 125-135 psig (75-79˚ F.) while 115 psig. (70˚ F) is the lower limit. I don't think the minimum condensing pressure has changed much over time. There are limits and those limits haven't really changed much. Almost every system could run at 135 psig (79˚F) and push it to 125 psig (75˚ F). 135 psig would have been the limit for liquid injection oil cooling on rotary screw compressors using thermal expansion valves (today they use electronic expansion valves and can run at lower condensing pressures). With reciprocating compressors or externally oil cooled rotary screw compressors 125 psig should be easily attainable. If it is not, it would be due to undersized piping (usually from expansion of the system over time) or oil separator size on the screw compressors (pretty rare in my experience but could happen). There is an ad from Ice & Refrigeration Magazine from December 1910 for a little perspective on lowering condensing pressure…it's not a new idea.

Respondent ID	Percentage of Similar New Construction Projects Using Floating Head Pressure Controls	Average Minimum Condensing Temperature Setpoint in Standard Practice Design	Notes
[E]	15%	70 - 73˚ F	Most projects (85%) use a fixed setpoint between 100 – 130 psig (64 - 77˚ F) or 100 – 140 psig (64 - 81˚ F), rather than a floating setpoint that changes as a function of a wet-bulb temperature measurement. On hot days, the systems with the fixed setpoint control can't achieve the setpoint and the actual head pressure floats higher even if they aren't using wet-bulb temperature control. For systems that use wet bulb approach control, 100-120 psig is the typical minimum setpoint, but similarly it floats higher depending on the weather. The typical median minimum temperature setpoint is 115 – 120 psig (70 - 73˚ F). If I had to guess, I would say 130 psig [77˚ F] was a common lower limit 20 years ago, now 110 psig [69˚ F] is more common, but certainly we still see new construction projects that can't go below 120 psig [73˚F].
[F]	100%	64˚ F	64˚ would be the minimum operating condensing temperature in cool ambient conditions, but operating pressure would typically go to 88˚ F during peak summer conditions. The last 5 years there has been a tendency to float the condensing pressures and allow them to go down to the 64˚ F level we have been talking about. Around 10 years ago this would have shrunk to around 75˚ F. Some will argue that 64˚ is insufficient for hot gas defrost. That may be true for liquid overfeed plants where the evaporators contain significant amounts of NH_3 liquid at defrost commencement. With more modern centralized, low charge NH_3 systems (DX), the evaporators contain so little ammonia that a short pump-down period of ~15 minutes will empty the evaporator of all liquefied refrigerant. A DX NH_3 freezer coil contains 0.006 to

Respondent ID	Percentage of Similar New Construction Projects Using Floating Head Pressure Controls	Average Minimum Condensing Temperature Setpoint in Standard Practice Design	Notes
			0.012 lb. NH3 per ton capacity. When the hot gas injection commences after pump-down, condensation starts instantly on all internal (cold) evaporator surfaces. All that is required for defrost completion is for the NH3 saturation temperature in the evaporator to be a little above the freezing point of water. Since this business started operations in 1996, the condensing pressure has been "floating" in all two stage NH3 refrigerating plants with reciprocating second stage compressors. In plants with second stage screw compressors, the minimum condensing pressure as dictated by the compressor manufacturer for oil circulation has been used.
[G]	30%	68˚ F	Floating head pressure is somewhat rare. Even if a facility has implemented a floating condensing pressure control in the past many revert back to a pressure setpoint. Some facilities will adjust the setpoint seasonally for some benefit. I would estimate <30% of facilities have implemented a floating condensing pressure control that succeeds in reducing energy consumption. I would say a typical summer setpoint is 150 psig and a typical winter setpoint is 110psig (68. Most, 80%+, of facilities are capable of operating at 100-120psig condensing pressure if they choose to use that setpoint during the winter (assuming there are seasons where the facility is located). Note these setpoints are not optimal and may or may not be "efficient". I haven't seen much change in the last 15 years.
Average	**62%**	**71˚ F**	

4.3.2 Recommended Baseline Around VFD on Evaporative Condenser Fans, rather than Two-Speed Fan Control

This section describes the results of the interview question with industrial refrigeration experts having to do with standard practice around how evaporative condenser fans are typically controlled in large new construction industrial refrigeration facilities.

For project RICE22N013, the claim in the project calculation is that standard practice would be to install a two-speed condenser fan to control the condensing temperature, and that the energy efficient design practice would involve installing a VFD. Table 4-4 shows the results of the interviews to this question.

The average of the respondents' estimates for the proportion of large new construction industrial refrigeration systems that would use VFDs to control evaporative condenser fans is 75%.

Based on this result, DNV recommends that variable speed control on evaporative condenser fans be considered the baseline for large new construction industrial refrigeration systems. This would eliminate the energy savings for this measure in project RICE22N013.

One similar comment from two survey respondents was that two-speed control would not be a realistic option in new construction, as this was a pre-cursor to VFDs. They stated that if VFDs were not installed, it would be more likely that the system would use single speed on/off control. Six of the seven respondents stated that 75% or more of new construction projects would use VFD control, one respondent stated that only 15% of large new construction industrial refrigeration project would use VFD control. The one respondent stated that 80% would use single-speed fans controlled in an on/off manner. Despite this different response, the majority of respondents stated that 75% or more of large new construction industrial refrigeration projects would use VFDs to control evaporative condensers. Therefore, DNV recommends that this be considered the baseline for this large new construction industrial refrigeration project.

Table 4-4. VFD on evaporative condenser fans results

Respondent ID	Percentage of Similar New Construction Projects that Would Install VFDs on the Evaporative Condenser Fans, rather than Two-Speed Fans	Notes
[A]	75%	It's a quick payback. Shouldn't even be a question on whether to install it or not. 75% of new construction large ammonia systems would have incorporate VFDS on evaporative condenser fans within the past 5 years. 5-10 years ago, this may have been 50%.
[B]	15%	15% VFDs, 5% electronically commutated motors (ECMs), 80% single-speed on/off control, 0% 2-speed control.
[C]	75%	VFD's are usually preferred, every job is different but usually price wise on a new construction drives price point makes it the common choice.
[D]	95%	95% or more VFD, single speed remainder. Two-speed was a pre-cursor to VFDs, so 0% would use two-speed control. The Northwest Energy Efficiency Alliance (www.neea.org) did some market research and case studies on VFD in the 1999-2002 timeframe. So, given some lag time for contractors, engineers, and manufacturers to respond and for the price of the VFDs to drop I would put the year around 2010 when it was becoming more common on new equipment. The retrofit of VFDs onto existing condensers is still happening. The

Respondent ID	Percentage of Similar New Construction Projects that Would Install VFDs on the Evaporative Condenser Fans, rather than Two-Speed Fans	Notes
		percentages question is too detailed for me to answer. I would assume 2010 would be in the 50-75% range. Less before, more after…
[E]	75%	I would guess that condenser fan VFDs started to become common in new construction in the last 10 years, but there are certainly still new construction projects that still use fan cycling instead of VFDs.
[F]	100%	VFDs on evaporative condensers have been the norm since around 2000
[G]	90%	This is fairly ubiquitous on new construction, 90%+ of installations will include VFDs on condenser fans. They are rarely controlled appropriately and provide only a small benefit from what I've seen. Drives on condenser fans in new systems have been the norm since I entered the industry in 2010. I'm not sure when the transition occurred.
Average	**75%**	

4.3.3 Recommended Baseline Around Economized High-Stage Compressors with VFD Control for Lead Compressor, rather than Non-Economized High Stage Compressors with Slide-Valve Control for Lead Compressor

This section describes the results of the interview question with industrial refrigeration experts having to do with standard practice around control of the lead compressor, and whether the high-stage compressors are typically economized or non-economized in large new construction industrial refrigeration facilities. By economizer, DNV specified that meant subcooling the liquid refrigerant before it reaches the evaporator and injecting the vapor generated during sub-cooling into the compressor partway through the compressor cycle.

For project RICE22N013, the claim in the project calculation is that high efficiency practice would be to install economized high-stage compressors with a VFD on the lead compressor, rather than the baseline of non-economized high-stage compressors with slide-valve control on the lead compressor.

When surveying the industrial refrigeration subject matter experts, DNV asked respondents for their estimates of the proportion of large new construction industrial refrigeration facilities that would use economized high stage compressors versus non-economized high stage compressors. A separate question was asked about the estimated proportion of large new construction industrial facilities that would use VFD to control the capacity of the lead compressor, rather than a slide-valve to control the capacity of the lead compressor. Table 4-5 shows the results of the interviews to this question.

Only four of the seven respondents were comfortable with providing an estimate for the proportion of new construction projects that would use economized high-stage compressors rather than non-economized high stage compressors. Two of those respondents stated that the proportion would be 75% or more, while one respondent stated that the proportion would be 5%. The average proportion among the respondents who provided an estimate was 59%. Due to the apparent non-consensus from the respondents, as well as the low response rate from the other interviewees, DNV recommends that prevailing baseline be used for project RICE22N013, which is non-economized high-stage compressors until such a point that additional information strongly suggests otherwise.

Table 4-5. Economizer on lead compressor results

Respondent ID	Percentage of Similar New Construction Projects that Would Install an Economizer on the Lead Compressor	Notes
[A]	75-95%	In cold storage - this is almost 100%. For process systems, this is about 50%. It depends on the size of the medium temperature load. If there is a large medium temperature load where there are multiple (1-2) compressors to take that medium temperature load, or a 2-stage system, that's where you won't see the use of an economized single stage system.
[B]	5%	This is silly - should just buy the right size compressor. Just buy a 125 HP machine. If this is a 2-stage system, no economizers are needed.
[C]	Not provided	A system this size (>1,000 HP) would most like have either a recirculator or surge drum, but if not then a subcooler would certainly be recommended. I would say price is usually a contributor to this as piping and lay out can be expensive.
[D]	Not provided	Great idea in theory but doesn't work in practice so much. It is a good idea for smaller systems. Whenever economizing is proportional, it is easy for the system to get out of balance. For a cold storage warehouse, the dock load is not proportional to the freezer load. Have seen facilities disable it. It is difficult to operate. I would not recommend that this measure be incentivized. It can be turned off at any time. Its use is very application specific. In cold storage, it is used more frequently than in large food manufacturing. To economize in large system, usually justified to go to 2 stages rather than economize with a single stage. This feature is done so they don't have to buy another compressor.
[E]	75%	N/A

Respondent ID	Percentage of Similar New Construction Projects that Would Install an Economizer on the Lead Compressor	Notes
[F]	Not provided	The trend is towards a mixture of screw compressors and reciprocating compressors within the same system with screw compressors dominating on the first (low temperature) compression stage and reciprocating compressors on the second compression stage of a dual-stage plant. This is all in the interest of maximizing energy efficiency. This eliminates the need for economized screw compressors.
[G]	70%	I don't have a good estimate on compressor economizer usage. There are many ways to achieve subcooling and economizers are often not the best option. Large freezer spaces with relatively small docks and no other intermediate loads seem to do well with economizers, I would say 70%+ of those new systems use them. I have seen many economizers valved out or cut/capped, particularly if they were trying to draw a large sideload.
Average	**59%**	

Table 4-6 shows the results of the interviews to the question about the proportion of large new construction industrial refrigeration projects that would install a VFD on the lead compressor, rather than using slide-valve control. Respondents were not consistent with each other around the proportion of large new construction industrial refrigeration projects that would install a VFD on the lead compressor. Four of the respondents stated that the proportion of projects that would do this would be 50% or less, while two of the respondents stated that 100% of projects would do this. The average proportion across all respondents is 52%. Because there was a larger number of respondents who stated that the proportion of projects that would install a VFD on the lead compressor is 50% or less, DNV recommends that the prevailing standard practice of slide valve control for the lead compressor be used for project RICE22N013.

Table 4-6. VFD on lead compressor results

Respondent ID	Percentage of Similar New Construction Projects that Would Install a VFD on the Lead Compressor, rather than Using a Slide-Valve to Control Capacity	Notes
[A]	25%	It is becoming more standard, but it is typically the first thing that gets cut.
[B]	50%	20% would have VFDs on multiple compressors
[C]	100%	Our standard practice would be for the lead compressor to have a drive. Let's say it's a 4-compressor system.

Respondent ID	Percentage of Similar New Construction Projects that Would Install a VFD on the Lead Compressor, rather than Using a Slide-Valve to Control Capacity	Notes
		Compressor 1 would have a drive and start first once the load hits 80% we would bring on compressor 2 without a drive and then use compressor 1 to trim the load before bringing on another compressor. The other 3 would still have slide valves.
[D]	10%	Usually, the compressor controls and sequencing are bad regardless of the presence of VFDs. VFDs are easier to justify as a retrofit, rather than in a new construction project. On a new system, you don't know what the loads are yet. The bigger the system, the less savings there would be.
[E]	25%	N/A
[F]	100%	All compressors would have VFDs
[G]	60%	This is common, I would estimate 60%+ of new installations utilize a VFD on each suction level. Many do not reduce energy consumption though and would be better off trimming with a slide valve machine depending on loads and compressor sizing.
Average	**53%**	

RHODE ISLAND CUSTOM ELECTRIC SITE-SPECIFIC REPORT
SITE ID: RICE22N050

Report Date: December 22, 2023

Program Administrator	Rhode Island Energy
Application ID(s)	11818303
Project Type	C&I Existing Building Retrofit
Evaluation Type	Ops
Program Year	PY2022
Evaluation Firm	DNV
Evaluation Engineer	Joshua Glick
Senior Engineer	Olav Hegland

1 EVALUATED SITE SUMMARY AND RESULTS

The evaluated project was implemented at an industrial metal fabrication facility and consisted of installing a VFD on the blower of a new 50 HP dust collector which replaced their old 50 HP dust collector. According to the project files, the baseline is new construction with a single baseline. The baseline system is a constant-speed dust collector motor without VFDs and a discharge damper to lower the flow to 83% of design flow. The measure saves energy by working with the airflow controller to ramp up and down based on flow requirements. Before the installation of the VFD, the flow was modulated using a discharge damper. During the initial interview with the site contact, evaluators learned the following:

- The site contact is present on-site and agreed to accommodate an on-site evaluation.
- It is safe to visit the facility and inspect the measure.

Based on the information gathered during the initial interview with the site contact, the evaluator proposed this site be evaluated using on-site verification with full M&V. An on-site audit was used to verify measure installation and operation and long term metering was used to capture the key parameters to conduct the operational adjustments. After reviewing the tracking files and information gathered during the site visit, the evaluator classified this measure as an add-on with single baseline and calculated the project savings using an 8,760 analysis based on the metered power data. The evaluation results are presented in Table 1-1.

Table 1-1. Evaluation results summary

PA application ID	Measure name		Annual electric energy (kWh)	% of energy savings on-peak	Summer on-peak demand (kW)	Winter on-peak demand (kW)
11818303	Blower VFD	Tracked	45,825	47.0%	10.20	10.20
		Evaluated	46,629	76.0%	7.93	10.74
		Realization Rate	101.8%	161.7%	77.7%	105.3%

1.1 Explanation of deviations from tracking

The evaluated savings are 1.8% greater than the tracked savings, primarily due to a lower post case fan motor demand despite lower post case operating hours. The applicant relied on a simple calculation model based on pre-installation run hours from a week of metered data and pre- and post-case fan motor demand from a published study, while the evaluator used an 8760 analysis with both pre and post metering data. Further details regarding deviations from the tracked savings are presented in Section 3.1.

1.2 Recommendations for program designers and implementers

If the size of the project warrants, the evaluator recommends that the implementer perform a post-installation metering period. By conducting a post-installation metering, the implementer can account for the actual post-installation operating hours and motor demand.

1.3 Customer alert

There are no relevant customer alerts.

DNV

2 EVALUATED MEASURES

The following sections present the evaluation procedure, including the findings from an in-depth review of the supplied applicant calculations and the evaluation methodology determined to be the best fit for the site and the information available.

The evaluated measure for this site is summarized in Table 2-1.

Table 2-1. Evaluated measure

Measure	Project ID	Parameter
M1	11818303	Replaced an end of life 50 HP dust collector with a new 50 HP dust collector and installed a VFD on the blower that communicates with the airflow controller of the new dust collector.

2.1 Application information and applicant savings methodology

This section describes the applicant's application information, savings methodology, and the evaluation assessment of the savings calculation algorithm used by the applicant.

2.1.1 Applicant description of baseline

According to the project files, the applicant classified this measure as new construction with a single baseline. Table 2-2 shows the baseline parameters in the model.

Table 2-2. Applicant baseline key parameters

Measure	Parameter	Operation description	Source of parameter value
M1	Baseline system	The baseline system is a constant speed, 50 HP dust collector motor without VFDs, using a discharge damper to lower the flow to 83% of design flow resulting in fan motor demand of 26.4kW with operating hours of 4,489 hrs/yr	Applicant savings analysis- fan motor demand is based on the motor power and the percentage of input power based on the chart from “How to Avoid Overestimating Variable Speed Drive Savings,” an article by Jonathan Maxwell, Figure 2-1, operating hours are based on a week of pre-existing metered data as shown in Table 2-3

Figure 2-1. Fan power curves from "How to Avoid Overestimating Variable Speed Drive Savings," an article by Jonathan Maxwell

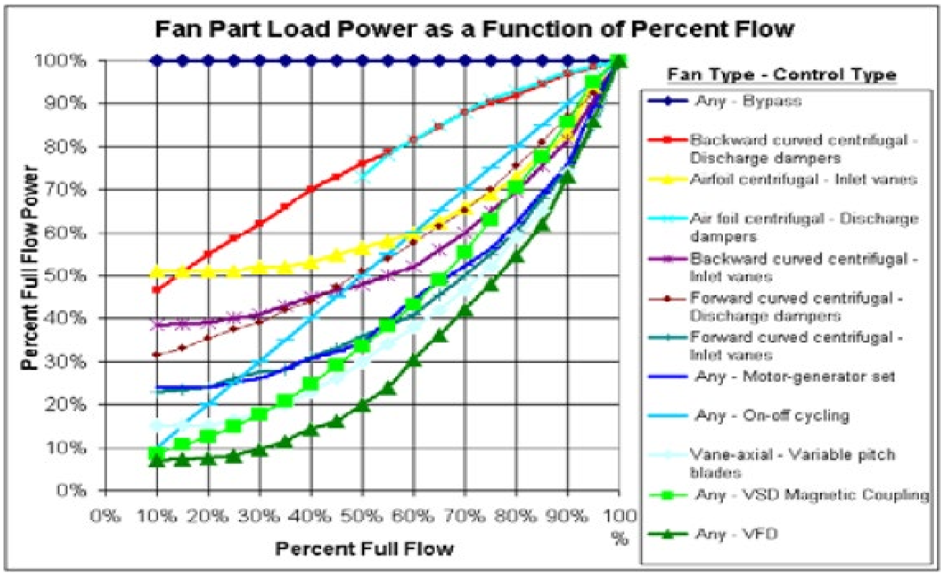


Table 2-3. Average baseline weekly operating profile

	MON	TUE	WED	THU	FRI	SAT	SUN
0	0.00	25.47	19.24	21.69	24.37	0.00	0.00
1	0.00	22.09	22.19	24.36	21.92	0.00	0.00
2	0.00	24.50	23.18	24.42	5.48	0.00	0.00
3	0.00	23.61	21.41	23.37	0.00	0.00	0.00
4	0.00	24.61	23.24	24.94	0.00	0.00	0.00
5	0.00	20.29	22.61	16.02	0.00	0.00	0.00
6	0.00	24.77	21.10	24.82	0.00	0.00	0.00
7	15.78	20.05	16.09	24.38	0.00	0.00	0.00
8	22.25	23.61	25.17	21.35	0.00	0.00	0.00
9	24.83	25.18	23.16	23.09	0.00	0.00	0.00
10	22.01	19.69	23.19	23.05	0.00	0.00	0.00
11	23.47	25.09	20.70	20.82	0.00	0.00	0.00
12	24.24	24.13	21.32	23.82	0.00	0.00	0.00
13	21.74	23.23	21.28	21.48	0.00	0.00	0.00
14	24.71	24.44	22.55	23.73	0.00	0.00	0.00
15	21.75	21.72	24.86	22.36	0.00	0.00	0.00
16	24.74	21.51	24.11	22.51	0.00	0.00	0.00
17	21.05	23.93	23.96	23.97	0.00	0.00	0.00
18	23.75	23.57	24.87	23.88	0.00	0.00	0.00
19	24.35	19.93	22.30	22.74	0.00	0.00	0.00
20	22.00	23.80	21.53	21.68	0.00	0.00	0.00
21	24.26	22.40	21.64	23.83	0.00	0.00	0.00
22	23.52	21.66	25.12	23.41	0.00	0.00	0.00
23	22.78	22.74	23.45	22.69	0.00	0.00	0.00

2.1.2 Applicant description of installed equipment and operation

This project includes installing a VFD on the new dust collector. Table 2-4 presents the main parameters of the proposed case as defined by the applicant.

Table 2-4. Applicant's proposed key parameters

Measure	Parameter	Operation description	Source of parameter value
M1	Proposed system	The installed equipment uses a fan input power of 57% resulting in fan motor demand of 16.2kW with the same operating hours of 4,489 hrs/yr	Applicant savings analysis-fan motor demand is based on the motor power and the percentage of input power based on the chart from "How to Avoid Overestimating Variable Speed Drive Savings," an article by Jonathan Maxwell, operating hours are based on a week of pre-existing metered data

2.1.3 Applicant energy savings algorithm

The applicant used 1 week of metered data on the previous unit to account for typical run time for the unit of 4,489 hrs/yr. The applicant used an estimated flow of 10,000 cfm which is 83% of the design flow of the unit based on expected operating parameters of the vendor. To calculate savings, the customer used a chart from "How to Avoid Overestimating Variable Speed Drive Savings," an article by Jonathan Maxwell as shown in Figure 2-1. This chart related the percentage of full flow to the percentage of full flow power and showed the baseline case used 93% of the input power and the proposed case with VFD used 57% of the input power. These percentages were multiplied by the motor power at design conditions to account for baseline and proposed fan motor demand. With the fan power demand and the operating hours, the customer used the fan motor annual usage and the difference between the baseline and proposed case was used to calculate the energy savings from this measure.

$$\Delta kWh = (Demand_{Base} - Demand_{New}) * Hours$$

Where:

$Demand_{Base}$ is the pre-existing fan motor demand of 26.4 kW

$Demand_{New}$ is the proposed fan motor demand of 16.2 kW

$Hours$ is the annual operating hours of 4,489 hrs

2.1.4 Evaluation assessment of applicant methodology

The evaluator determined that the applicant's calculations to estimate energy savings were appropriate. However, the calculations only used one week of metered data to account for the operating hours. The fan power demand can also be better accounted for through production or actual metered power data rather than a table relating full flow and power.

2.2 On-site inspection and metering

The site contact indicated that it was safe to visit the site and preferred an on-site verification with power metering installed. The evaluator conducted the site visit on October 12, 2023, and was assisted by an employee. Table 2-5 summarizes the findings from the installed measure verification.

Table 2-5. Measure verification.

Measure name	Verification method	Verification result
Blower VFD	On-site inspection	The evaluator verified the installed dust collector and VFD. The installed dust collector is 50HP as shown in Figure 2-2. During the visit, the dust collector was off and the evaluator was able to turn it on and off to watch the dust collector and VFD ramp up and down. The dust collector panel and VFD are shown in Figure 2-3.
Blower VFD	Metering Equipment	The evaluator installed a Dent Elite Pro on the dust collector which was left in place for eight weeks and the meter logged power data in 15-minute intervals.
Blower VFD	Interview the site contact for pre-existing situations	The blower and VFD were installed in May 2022, the site contact confirmed that the existing annual production hours of 4,489 used by the applicant was reasonable. The production is not weather dependent. No significant changes in operation have occurred since installation.

Figure 2-2. Installed Donaldson Torit blower nameplate

Figure 2-3. Dust collector panel and VFD

2.3 Evaluation methods and findings

This section describes the evaluator methods and findings.

2.3.1 Evaluation description of baseline

Based on information provided in the project files and gathered during the site visit, the evaluator determined this measure is a new construction with an add-on single baseline as a VFD was added to a new dust collector. The baseline system is a new dust collector motors with a discharge damper to lower the flow to 83% of design flow.

2.3.2 Evaluation calculation method

The evaluator calculated the project impacts using an 8,760 analysis based on trend data received from the onsite metering. An average week was calculated and extrapolated over the year as the production is not weather dependent.

This analysis showed that the post case production hours are only 3,848 hrs/yr rather than 4,489 hrs/yr that the applicant used based on pre-existing metered data. The average operating profile can be seen in Table 2-6 and shows a lower average fan demand than the 16.2 kW used by the applicant.

Table 2-6. Average weekly post-installation operating profile – from long term monitoring

	MON	TUE	WED	THU	FRI	SAT	SUN
0	0.00	10.56	11.56	12.72	5.12	0.00	0.00
1	0.00	12.51	7.74	7.5	2.02	0.00	0.00
2	0.00	2.81	0.00	0.00	0.00	0.00	0.00
3	0.00	0.01	0.00	0.00	0.00	0.00	0.00
4	0.00	0.01	0.00	0.00	0.00	0.00	0.00
5	0.00	0.02	0.00	0.00	0.00	0.00	0.00
6	2.29	6.09	3.04	0.00	0.00	0.00	0.00
7	8.55	9.51	3.71	0.00	0.00	0.00	0.00
8	10.48	11.2	5.84	0.00	0.00	0.00	0.00
9	10.19	10.48	6.9	0.00	0.00	0.00	0.00
10	10.67	10.03	7.75	0.05	0.00	0.00	0.00
11	10.56	11.47	8.85	0.00	0.00	0.00	0.00
12	10.09	9.04	7.86	0.00	0.00	0.00	0.00
13	10.71	9.85	7.19	0.00	0.00	0.00	0.00
14	9.87	9.42	12.55	0.00	0.00	0.00	0.00
15	10.61	9.81	8.13	0.00	0.00	0.00	0.00
16	10.87	11.72	7.78	0.00	0.00	0.00	0.00
17	13.39	12.6	10.6	0.55	0.00	0.00	0.00
18	13.37	12.91	11.53	4.03	0.00	0.00	0.00
19	12.92	9.53	12.09	6.54	0.00	0.00	0.00
20	10.85	10.67	12.4	6.54	0.00	0.00	0.00
21	13.29	12.89	12.72	6.54	0.00	0.00	0.00
22	13.58	10.77	11.66	6.54	0.00	0.00	0.00
23	11.8	12.64	11.88	6.58	0.00	0.00	0.00

The 8760 analysis took this average weekly operating profile and applied it throughout the year. For the baseline kW data the evaluator took the applicant's weekly operating profile as seen in Table 2-3 and adjusted it to match the schedule of the evaluated operating profile. For any hour where the evaluated kW data was less than 0.1 kW, the applicant baseline kW was set to 0kW. This adjustment can be seen in Table 2-7 and better reflects the post installation operating schedule as seen through the evaluator metering period while leaving the average pre-installation metering kWs intact for the hours when the post-metering indicates operation. The analysis calculated savings by summing the hourly average kW difference from the base case to the installed case. Peak kW savings are calculated by using the RI Energy summer and winter peak periods and taking the hourly savings during those peak periods.

Table 2-7. Average evaluator adjusted baseline weekly operating profile

	MON	TUE	WED	THU	FRI	SAT	SUN
0	0.00	25.47	19.24	21.69	24.37	0.00	0.00
1	0.00	22.09	22.19	24.36	21.92	0.00	0.00
2	0.00	24.50	0.00	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00	0.00	0.00	0.00
4	0.00	0.00	0.00	0.00	0.00	0.00	0.00
5	0.00	0.00	0.00	0.00	0.00	0.00	0.00
6	0.00	24.77	21.10	0.00	0.00	0.00	0.00
7	15.78	20.05	16.09	0.00	0.00	0.00	0.00
8	22.25	23.61	25.17	0.00	0.00	0.00	0.00
9	24.83	25.18	23.16	0.00	0.00	0.00	0.00
10	22.01	19.69	23.19	0.00	0.00	0.00	0.00
11	23.47	25.09	20.70	0.00	0.00	0.00	0.00
12	24.24	24.13	21.32	0.00	0.00	0.00	0.00
13	21.74	23.23	21.28	0.00	0.00	0.00	0.00
14	24.71	24.44	22.55	0.00	0.00	0.00	0.00
15	21.75	21.72	24.86	0.00	0.00	0.00	0.00
16	24.74	21.51	24.11	0.00	0.00	0.00	0.00
17	21.05	23.93	23.96	23.97	0.00	0.00	0.00
18	23.75	23.57	24.87	23.88	0.00	0.00	0.00
19	24.35	19.93	22.30	22.74	0.00	0.00	0.00
20	22.00	23.80	21.53	21.68	0.00	0.00	0.00
21	24.26	22.40	21.64	23.83	0.00	0.00	0.00
22	23.52	21.66	25.12	23.41	0.00	0.00	0.00
23	22.78	22.74	23.45	22.69	0.00	0.00	0.00

3 FINAL RESULTS

The evaluated project consisted of installing a VFD on the new dust collector. The evaluated savings are greater than the tracking values, primarily due to the fan motor demand being less than the applicant assumption despite fewer operating hours. The parameters impacting the analysis are summarized in Table 3-1.

Table 3-1. Summary of key parameters

	BASELINE		PROPOSED / INSTALLED	
Parameter	Tracking Value(s)	Evaluation Value(s)	Tracking Value(s)	Evaluation Value(s)
Operating hours	4,489hrs/yr	3,848hrs/yr	4,489hrs/yr	3,848hrs/yr
Average fan motor demand	26.4kW	22.8kW	16.2kW	8.9kW

3.1 Explanation of differences

The evaluated savings are 1.8% greater than the tracking values because of the difference in operations relating to the decreased fan demand in the post case despite the fewer operating hours. Table 3-2 provides a summary of the differences between tracking and evaluated values.

Table 3-2. Summary of deviations

Measure	Discrepancy	Parameter	Impact of deviation	Discussion of deviations
M1	Operation	Operating Hours	-14 %	Decreased savings – the evaluated findings indicate that the actual operating hours are 641 hours less than the applicant calculations
M1	Operation	Operating Load	+15%	Increased savings – the evaluated findings indicate the actual post case fan motor demand is 7.3 kW less than the applicant calculations and the evaluated baseline demand is 3.6 kW less than the applicant calculations.
M1	Methodology	Savings Calculation methodology	+1%	Increased savings – the evaluated findings are 1% higher based on the more accurate 8,760 analysis used by the evaluator instead of the applicant's formula

3.2 Lifetime savings

The evaluator classified measure the evaluated measure as a new construction with single baseline. The evaluator calculated applicant and evaluated lifetime savings values using the following formula:

$$\mathrm{LAGI} = \mathrm{FYS} \times [\,\mathrm{RUL} + \mathrm{outyear}\,\% \times (\mathrm{EUL} - \mathrm{RUL})]$$

where:

LAGI = lifetime adjusted gross impact (kWh)

FYS = first year savings (kWh)

EUL = measure life (years)

RUL = 1/3 of EUL (years)

outyear % = 100% for this single baseline measure

The evaluated lifetime savings are greater than the tracking lifetime savings because the evaluated first year savings are greater than the tracking first year savings. Table 3-3 provides a summary of key factors that influence the lifetime savings.

Table 3-3. Measure M1 – Application ID: 11818303 – Lifetime savings summary

Factor	Tracking	Application	Evaluator
Lifetime savings (kWh)	458,250	458,250	466,290
First-year savings (kWh)	45,825	45,825	46,629
Measure lifetime (years)	10	10	10
Baseline classification	New Construction	New Construction	New construction -Add on

RI CUSTOM ELECTRIC EVALUATION SITE-SPECIFIC REPORT

DNV SITE ID: RICE22N054

Report Date: 3/14/2024

Application ID(s)	11982673	
Project Type	C&I End of Useful Life Replacement	
Program Year	2022	
Evaluation Firm	DMI	
Evaluation Engineer	Bennett Rose	
Senior Engineer	Mickey Bush	

1 EVALUATED SITE SUMMARY AND RESULTS

This retrofit project consisted of a new energy management system (EMS) at a 3-story 39,000 ft² office building. The retrofit measure implemented at this facility is the installation of a new EMS and was submitted as a custom project with prescriptive EMS savings. The equipment controlled by the EMS is three pre-existing RTUs and 28 VAV boxes.

The electric savings associated with the controls project were calculated using the prescriptive EMS tool. The applicant indicated that the implemented sequences include; 7-day schedule, optimal start/stop, DDC temp control, and night space temperature setbacks. The implication is that the applicant baseline is a control system that does not include these sequences. The savings tool condenses the different control sequences in to one savings algorithm for reducing equipment runtime and another savings algorithm for implementing DDC temperature controls.

The evaluated savings for this project are based on metered data collected by the evaluator. The evaluator confirmed that the baseline sequence of operation for the RTUs considered in the project did not include occupancy schedule controls. The evaluator developed an occupied and unoccupied mode operating profile based on the metered data and compares that profile to a baseline that is in "occupied" mode year-round to calculate savings. The operating profiles of the RTUs developed based on metered data are taken to be reflective of the cumulative impact of the different control sequences implemented with the new EMS.

The evaluated savings are less than the tracking savings for this project primarily because the schedule controls result in fewer unoccupied hours than assumed by the applicant and because the average cooling demand is less than assumed by the applicant.

The evaluation results are presented in Table 1-1.

Table 1-1. Evaluation Results Summary

PA Application ID	Measure Name		Annual Electric Energy (kWh)	% of Energy Savings On-Peak	Summer On-Peak Demand (kW)	Winter On-Peak Demand (kW)
11982673	New EMS	Tracked	64,315	0%	7.3	7.3
		Evaluated	32,718	27.81%	0	0
		Realization Rate	50.87%	N/A	0.00%	0.00%

1.1 Explanation of Deviations from Tracking

The evaluated savings are 49.1% less than the applicant-reported savings. The reduction in savings is due to the evaluation finding that the implemented occupancy schedule for the RTUs results in more annual run hours than assumed by the applicant, and because the average cooling demand is less than assumed by the applicant. This means that the run time reduction savings are less than expected and that the baseline cooling energy is less than expected.

1.2 Recommendations for Program Designers & Implementers

The project was processed as a custom project using the prescriptive EMS savings tool to calculate energy savings. A project should either be considered under the prescriptive program using the prescriptive EMS tool for deemed savings, or the project should be analysed using a custom analysis approach reflecting a unique controls strategy associated with the project and be processed under the custom incentive program.

Many of the issues identified in this evaluation that led to lower savings are a result of treating the project as a prescriptive EMS project, but processing the project as a custom project. Two drivers of the low realization rate for this project are over estimating the cooling profile of the RTUs, and under estimating the installed case operating hours. A more detailed baseline documentation effort would be appropriate for a custom analysis and would have likely provided a more accurate estimation of the cooling profile of the building. The MRD for the project defines the number of points in the EMS and does not specify the scheduled occupancy hours. The number of points is a metric used for the prescriptive EMS project but does not have a direct influence on energy savings, while the implemented schedule does have a direct impact on project savings.

The detail provided in the applicant's baseline documentation rigor, analysis methodology and MRD are appropriate for a prescriptive EMS project, but project was processed as custom. If the project is considered custom, the applicant needs to clearly define the baseline, use an 8,760 or bin model spreadsheet model to calculate savings, and the MRD should be useful in confirming the energy savings parameters of the project.

1.3 Customer Alert

None.

2 EVALUATED MEASURES

The following sections present the evaluation procedure, including the findings from an in-depth review of the supplied applicant calculations and the evaluation methodology determined to be the best fit for the site and the information available.

2.1 Application Information and Applicant Savings Methodology

This section describes the applicant's application information, savings methodology, and the evaluation assessment of the savings calculation algorithm used by the applicant.

2.2 Applicant Description of Baseline

The applicant measure event type is retrofit. The applicant's baseline descriptions are inferred from the prescriptive EMS tool. The baseline considered by the applicant is the existing RTU and VAV box controls that did not include scheduling or space temperature setbacks. The baseline values used in the applicant savings analysis are presented in Table 2-1.

Table 2-1. Applicant baseline key parameters

Measure	Parameter	BASELINE		
		Value(s)	Source of Parameter Value	Note
M1	Connected DX Compressor Connected Load	63.5 kW	Prescriptive EMS tool	
M1	Connected DX Compressor Annual Run Hours	3,360 hours	Prescriptive EMS tool	
M1	Supply Fans Connected Load	13.5 kW	Prescriptive EMS tool	
M1	Supply Fans Annual Run Hours	8,736 hours	Prescriptive EMS tool	
M1	Exhaust Fans Connected Load	0.23 kW	Prescriptive EMS tool	

M1	Exhaust Fans Annual Run Hours	8,736 hours	Prescriptive EMS tool	

2.2.1 Applicant Description of Installed Equipment and Operation

The applicant assumes that an EMS will be installed with 12 gas control points and 106 electric control points. The applicant assumes that the new EMS will implement 7-day schedule, optimal stop start, night setback, and DDC temperature control. The proposed system values used in the applicant savings analysis are presented in Table 2-2.

Table 2-2: Application proposed key parameters

Measure	Parameter	PROPOSED		
		Value(s)	Source of Parameter Value	Note
M1	Connected DX Compressor Connected Load	63.5 kW	Prescriptive EMS tool	
M1	Connected DX Compressor Annual Run Hours	2,100 hours	Prescriptive EMS tool	
M1	Supply Fans Connected Load	13.5 kW	Prescriptive EMS tool	
M1	Supply Fans Annual Run Hours	5,460 hours	Prescriptive EMS tool	
M1	Exhaust Fans Connected Load	0.23 kW	Prescriptive EMS tool	
M1	Exhaust Fans Annual Run Hours	5,460 hours	Prescriptive EMS tool	

2.2.2 Applicant Energy Savings Algorithm

The savings for this measure are calculated using the prescriptive EMS calculator. This calculator has the user enter estimated electric demand, baseline run hours, and proposed run hours for each end use (DX cooling, supply fans, return fans) controlled by the EMS.

The user enters a binary value for each of the control strategies implemented with the EMS. In the case of this project those control strategies include 7-day schedule, optimal start, night setbacks, and DDC temperature controls. And the calculator assigns savings to each of the selected control strategies using fixed formulas.

Table 2-3 summarizes the specifications of the RTUs considered for energy savings associated with this EMS project.

Table 2-3: RTU Summary

ID	Supply Fan Motor HP	Nominal Cooling Capacity Tons	Nominal Heating Capacity MBH
RTU-1	10	25	400
RTU-2	7.5	20	250
RTU-3	5	12.5	250

Twenty-eight VAV terminal units are also included in the scope of the controls project, but are not associated with inputs in the savings calculator.

Table 2-4 summarizes the savings calculator applicant assumptions. All assumptions are entered as values i.e. not calculated in the spreadsheet.

Table 2-4: Savings Calculator Inputs

System Component	Connected Load	Baseline			Proposed		
		Hours per Week	Weeks per Year	Annual Hours	Hours per Week	Weeks per Year	Annual Hours
DX Compressors	63.5	168	20	3,360	105	20	2,100
Supply Fans	13.5	168	52	8,736	105	52	5,460
Exhaust Fans	0.23	168	52	8,736	105	52	5,460

The savings associated with 7-day schedule, optimal start, and night setbacks are consolidated into one formula for equipment runtime savings. Note that this formula is the same regardless of how many or which combination of these control strategies are selected in the calculator. The formulas used to calculate equipment runtime savings is as follows.

$$Equipment\ Runtime\ Savings\ kWh = Connected\ Load\ kW * Annual\ Run\ Hour\ Reduction * \ Savings\ Factor$$

Where Savings Factor is 0.2 for DX Cooling and 1 for Supply and Exhaust Fans

Compressor savings are claimed for DDC temperature control. It is unclear what the basis for energy savings is for this control strategy. The savings are calculated using the following formula.

$$DDC\ Control\ Savings\ kWh = Connected\ Load\ kW * Annual\ Hours_{After\ EMS} * 0.025\ Savings\ Factor$$

The savings associated with reducing equipment runtime and DDC temperature control are calculated for each piece of equipment detailed in table 2-4. Table 2-5 summarizes the energy savings by source of savings for each system component.

Table 2-5: Energy Savings Summary

System Component	Equipment Runtime Savings kWh	DDC Control Savings kWh	Total Savings kWh
DX Compressors	16,002	3,334	19,336
Supply Fans	44,226		44,226
Exhaust Fans	753		753
Total	60,981	3,334	64,315

2.2.3 Evaluation Assessment of Applicant Methodology

The applicant uses a prescriptive savings tool for a custom project. The reason for the use of this tool for custom projects is not clear. It is also not clear if or how each controls strategy that is included in the project is considered in the energy savings analysis. 7-day schedule, optimal start, nighttime setbacks are all included in the project, but the savings algorithm for equipment runtime savings seems to indicate a change from 24/7 operation to seven 15-hours days per week

The basis for the demand inputs used in the tool are unclear, but appear reasonable based on the nameplate information for the three RTUs considered for energy savings. Supply fan kW is 80.4% of the combined nominal fan motor HP for the three RTUs and the compressor demand input assumes a cooling performance of ~1.1 kW/ton which is reasonable. The source of the exhaust fan demand input is not clear as the RTUs do not have exhaust fans.

The basis for energy savings associated with reducing equipment runtime is straightforward, however the basis for energy savings associated with DDC temperature control is not addressed in the prescriptive savings tool.

2.3 On-site Inspection and Metering

This section provides details on the work performed during the on-site inspection. Evaluators were granted access to the site and conducted a full M&V evaluation.

2.3.1 Summary of Site Visit

This section summarizes the site visit.

- The evaluator visited the site on September 12, 2023.
- The evaluator installed kW meters on the 3-phase 480V feeds for RTU-1, RTU-2, and RTU-3. The metered loads include supply fan, compressor, and auxiliary RTU power. Though the metering started at the end of the summer, the period includes 87 hours when the outside air temperature is greater than 75°F and includes a maximum outside air temperature of 85°F.
- After the site visit, the site contact answered the site visit interview questions via email.
- The evaluator returned to the site on January 18, 2023 to retrieve the meters.
- The evaluator requested trend data through the controls contractor for the project following the first site visit. Trend data was not provided by the controls contractor after months of follow-up.

Table 2-6. Measure Verification

Measure Name	Verification Method	Verification Result
M1 – New EMS	Metered RTUs and discussed baseline system with site contact.	Metered data demonstrates that an operating schedule is implemented for all 3 RTUs. Site feedback indicates that no schedule was implemented with the pre-existing controls system.

2.3.2 Measured and Logged Data

Table 2-7 summarizes the metered data collected and the period of useful data collected for each RTU. The metered data from the useful data period is used in the evaluation savings analysis.

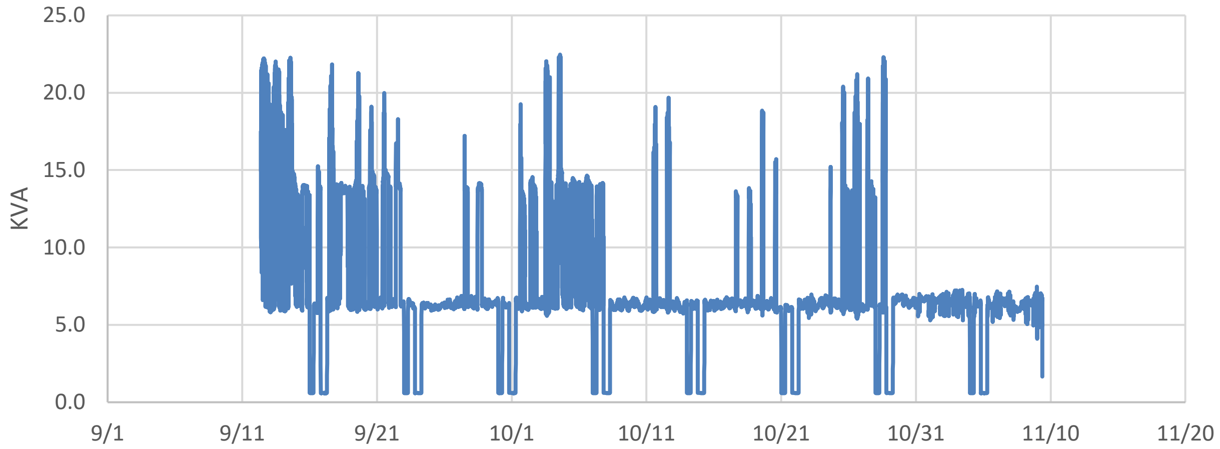
Table 2-7. Metered Data Summary

Equipment	Useful Data Duration
RTU 1 (fan and compressor) KVA (5-minute interval)	58 Days (9/12/2023-11/9/2023)
RTU 2 (fan and compressor)KVA (5-minute interval)	58 Days (9/12/2023-11/9/2023)
RTU 3 (fan and compressor)KVA (5-minute interval)	58 Days (9/12/2023-11/9/2023)

All three RTUs are served from one electrical panel and one three-phase kW meter was used to monitor all three loads separately. The electric meter installed used a single voltage connection and separate amperage channels for each RTU load. When processing the data, it was found there was a phase mismatch between the voltage and amperage readings leading to faulty RMS kW demand readings although the amperage and voltage measurements are accurate. The amperage and voltage data is used to calculate kVA and power factor must be assumed to calculate kW.

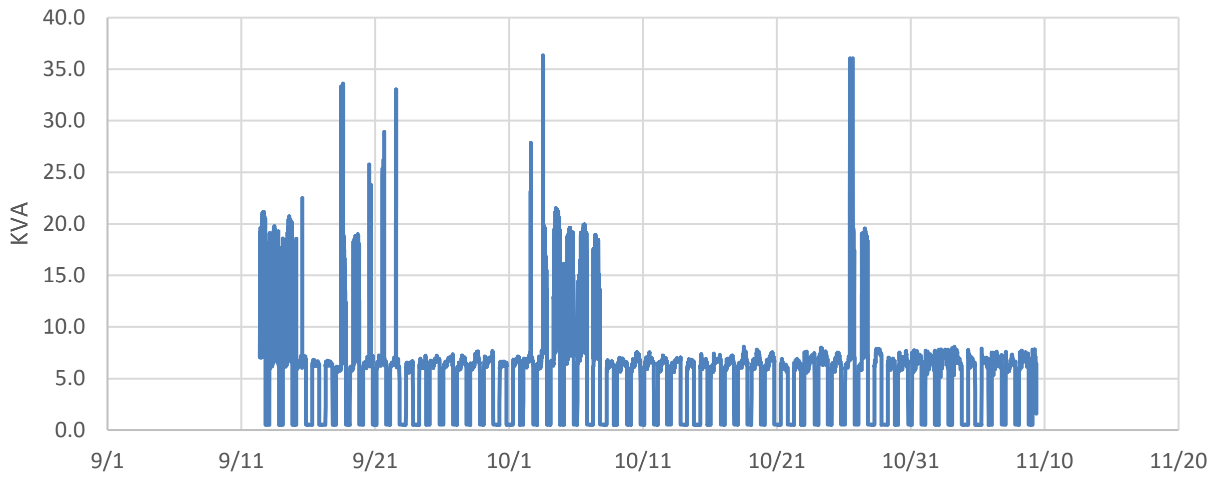
The metered KVA data for RTU-1 is presented in Figure 2-1.

Figure 2-1. RTU-1 Raw KVA Data



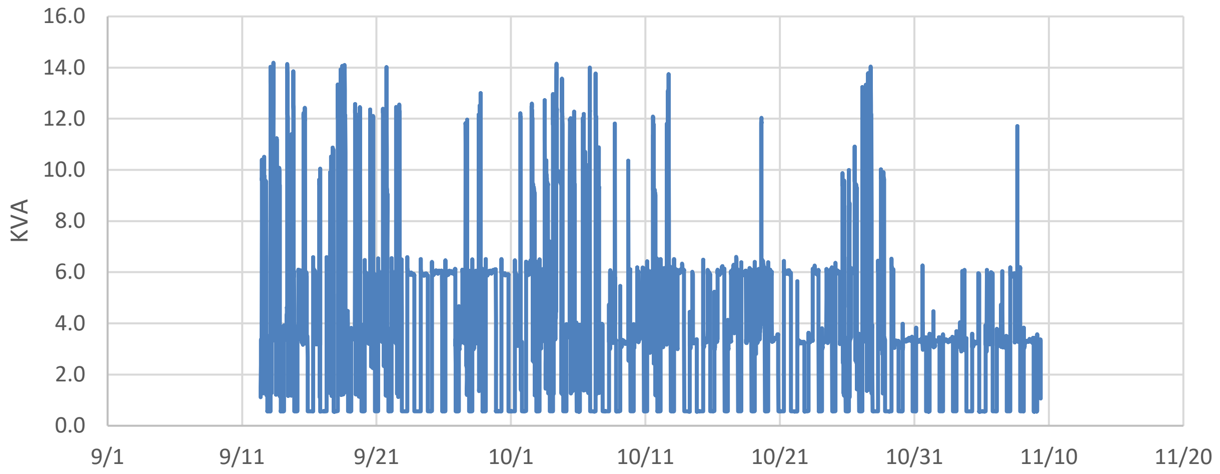
The metered KVA data for RTU-2 is presented in Figure 2-2.

Figure 2-2. RTU-2 Raw KVA Data



The metered KVA data for RTU-3 is presented in Figure 2-3.

Figure 2-3. RTU-3 Raw KVA Data



The site contact referred the evaluator to the controls contractor associated with this project to provide the trend data request associated with this project. The MV plan for this evaluation included a trend data review to confirm schedule controls, RTU airflow and temperature profiles, etc. The controls contractor has not been able to provide trends for the site after months of follow up emails requesting trends and the evaluation analysis was completed without trend data.

RI Energy provided the evaluator with 15-minute electric interval data for the site. The utility meter is a campus meter that includes three buildings; the office building served by the RTUs considered in this evaluation project, an old lab building mostly used for storage now, and a newer lab building.

The site provided the evaluator with 15-minute electric submeter data for the newer lab building.

The interval data collected by the evaluator is summarized in Table 2-8.

Table 2-8. Metered Data Summary

System Component	Buildings Included	Data Description
Utility Interval Data	Office Building, Old Lab Building, New Lab Building	15-minute kW data 1/1/2021-11/7/2023
Lab Building Interval Data	New Lab Building	15-minute kW data 1/1/2021-11/8/2023

2.4 Evaluation Methods and Findings

This section describes the evaluator methods and findings.

2.4.1 Evaluation Description of Baseline

Little to no documentation or description of the existing controls system was collected by the applicant. The preexisting controls system did not have trending capabilities according to the site so data is not available to document RTU operation with the pre-existing controls.

The feedback from the site about the preexisting controls is that the Carrier rooftop units were controlled by a proprietary 33CS Carrier control system. Cooling and heating was dictated by a number of VAVs in the zone and the number could be set manually in a master thermostat with heating mode taking a priority. There was no operating schedule or space temperature setback associated with this control system.

Based on the feedback provided by the site. The evaluator accepts the baseline of RTU operation without a schedule and no nighttime space temperature setbacks.

During the site interview, the site claimed that exhaust fan scheduling was not part of the EMS project. Trends were not provided to the evaluator to confirm or deny this feedback. Based on the available information the evaluator does not include exhaust fan energy in the evaluated savings for this project. The exhaust fan in question is reportedly bathroom exhaust.

2.4.2 Evaluation Calculation Method

The evaluator uses an 8,760-spreadsheet analysis with Providence TMY3 weather data to calculate savings for this project. Savings were calculated by evaluating the operating schedules and operating profiles of the three RTUs included in the new EMS project to compare the equipment runtime savings with 24/7 operation in the baseline. The evaluated savings are based on the cumulative impact of the control strategies implemented, i.e. optimal start and occupancy schedules are not investigated separately. This approach mirrors the methodology of the prescriptive EMS tool in which multiple control strategies are combined into an equipment runtime savings algorithm that considers annual operating hours before and after the installation of the new EMS.

The metered data collected by the evaluator is used to determine the weekly schedule for each of the RTUs. A time of day, day of week matrix of the average demand for each RTU is created to determine the occupancy schedules.

Table 2-8 presents the calculated occupancy schedule for RTU-1

Table 2-8. RTU-1 Occupancy Schedule

Hour	Day of Week						
	Sunday	Monday	Tuesday	Wednesday	Thursday	Friday	Saturday
	1	2	3	4	5	6	7
0	0	1	1	1	1	1	0
1	0	1	1	1	1	1	0
2	0	1	1	1	1	1	0
3	0	1	1	1	1	1	0
4	0	1	1	1	1	1	0
5	0	1	1	1	1	1	0
6	0	1	1	1	1	1	0
7	1	1	1	1	1	1	1
8	1	1	1	1	1	1	1
9	1	1	1	1	1	1	1
10	1	1	1	1	1	1	1
11	1	1	1	1	1	1	1
12	1	1	1	1	1	1	1
13	1	1	1	1	1	1	1
14	1	1	1	1	1	1	1
15	1	1	1	1	1	1	1
16	1	1	1	1	1	1	1
17	1	1	1	1	1	1	1
18	1	1	1	1	1	1	1
19	1	1	1	1	1	1	1
20	1	1	1	1	1	1	0
21	1	1	1	1	1	1	0
22	1	1	1	1	1	1	0
23	1	1	1	1	1	1	0

Table 2-9 presents the calculated occupancy schedule for RTU-2

Table 2-9. RTU-2 Occupancy Schedule

Hour	Day of Week						
	Sunday	Monday	Tuesday	Wednesday	Thursday	Friday	Saturday
	1	2	3	4	5	6	7
0	0	0	0	0	0	0	0
1	0	1	0	0	0	0	0
2	0	1	1	1	1	1	0
3	0	1	1	1	1	1	0
4	0	1	1	1	1	1	0
5	0	1	1	1	1	1	0
6	0	1	1	1	1	1	0
7	1	1	1	1	1	1	1
8	1	1	1	1	1	1	1
9	1	1	1	1	1	1	1
10	1	1	1	1	1	1	1
11	1	1	1	1	1	1	1
12	1	1	1	1	1	1	1
13	1	1	1	1	1	1	1
14	1	1	1	1	1	1	1
15	1	1	1	1	1	1	1
16	1	1	1	1	1	1	1
17	1	1	1	1	1	1	1
18	1	1	1	1	1	1	1
19	1	1	1	1	1	1	1
20	0	0	0	0	0	0	0
21	0	0	0	0	0	0	0
22	0	0	0	0	0	0	0
23	0	0	0	0	0	0	0

Table 2-10 presents the calculated occupancy schedule for RTU-3

Table 2-10. RTU-3 Occupancy Schedule

Hour	Day of Week						
	Sunday	Monday	Tuesday	Wednesday	Thursday	Friday	Saturday
	1	2	3	4	5	6	7
0	0	0	0	0	0	0	0
1	0	0	0	0	0	0	0
2	0	0	0	0	0	0	0
3	0	1	1	1	1	1	0
4	0	1	1	1	1	1	0
5	0	1	1	1	1	1	0
6	0	1	1	1	1	1	0
7	1	1	1	1	1	1	1
8	1	1	1	1	1	1	1
9	1	1	1	1	1	1	1
10	1	1	1	1	1	1	1
11	1	1	1	1	1	1	1
12	1	1	1	1	1	1	1
13	1	1	1	1	1	1	1
14	1	1	1	1	1	1	1
15	1	1	1	1	1	1	1
16	1	1	1	1	1	1	1
17	1	1	1	1	1	1	1
18	1	1	1	1	1	1	1
19	1	1	1	1	1	1	1
20	0	1	1	1	1	1	0
21	0	0	0	0	0	0	0
22	0	0	0	0	0	0	0
23	0	0	0	0	0	0	0

These schedules are used to filter the raw electric data collected for each RTU and to determine the relationship between outside air temperature and RTU demand for each RTU during occupied mode as well as the average standby losses for the RTUs during unoccupied mode which the metered data shows are non-zero. These relationships are used to model the annual operation of the three RTUs considered in this project.

Note that a linear regression is used to calculate the average RTU demand as a function of outside air temperature. The strength of these regressions is poor when assessed based on the R^2 value which is expected. The compressors stage on an off as needed to provide cooling resulting in the data showing discrete KVA levels associated with the different RTU systems (supply fan, DX stage 1 and DX stage 2) cycling on and off. The linear regression represents the average demand accounting for compressor runtime at varying outside air temperature conditions (i.e. varying cooling loads).

Figure 2-4 compares the modelled occupied and unoccupied KVA for RTU-1 compared to the raw kVA metered data.

Figure 2-4. RTU-1 Raw KVA Data vs Modelled RTU-1 KVA

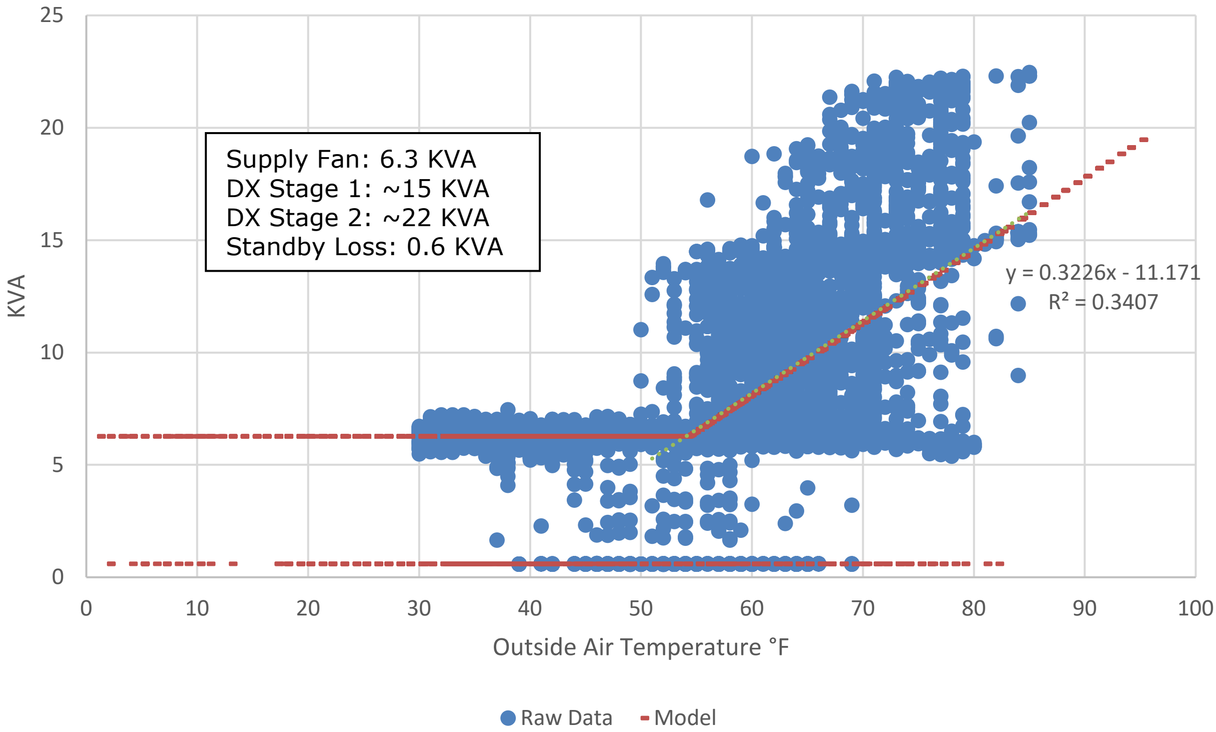


Figure 2-5 compares the modelled occupied and unoccupied KVA for RTU-2 compared to the raw kVA metered data.

Figure 2-5. RTU-2 Raw KVA Data vs Modelled RTU-2 KVA

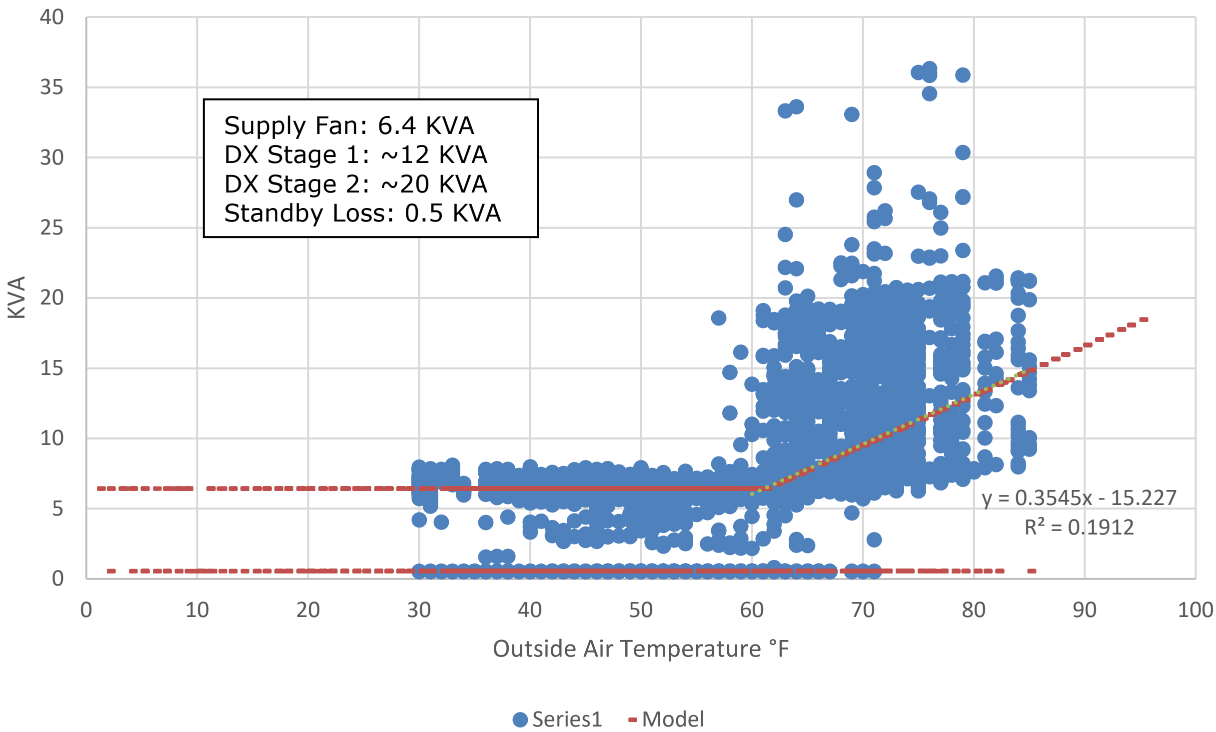
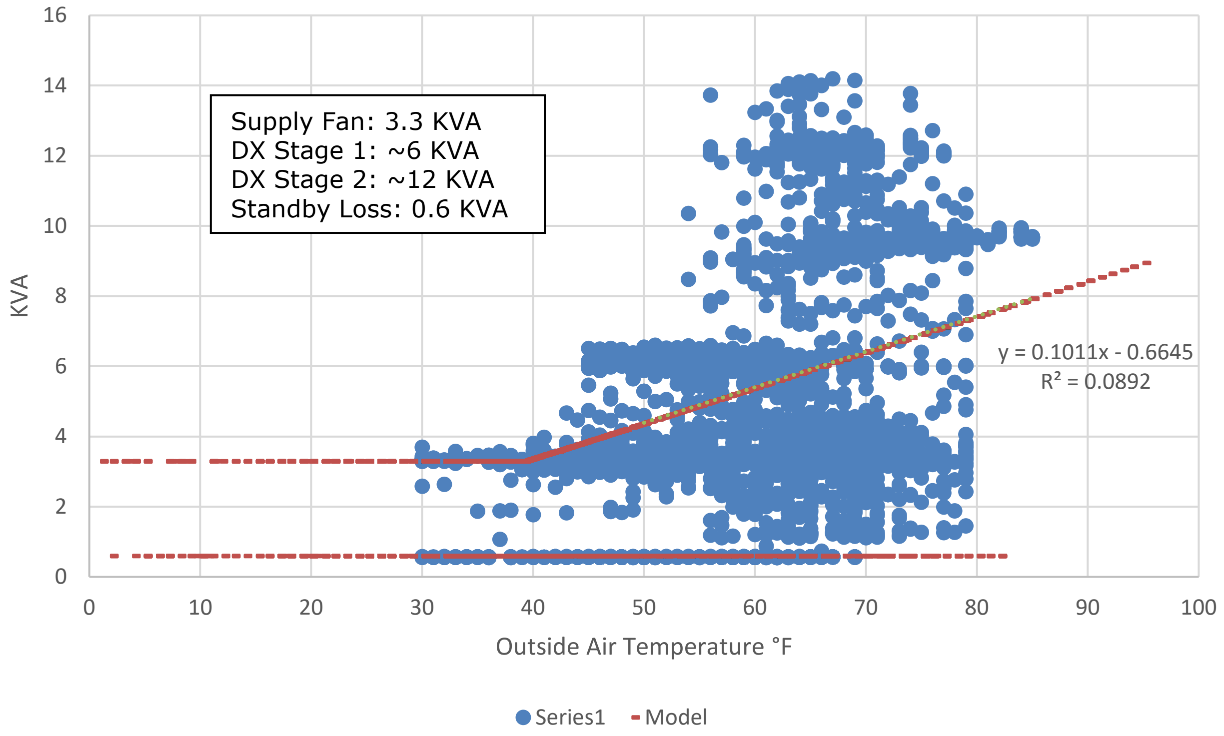


Figure 2-6 compares the modelled occupied and unoccupied RTU for RTU-3 compared to the raw kVA metered data.

Figure 2-6. RTU-3 Raw KVA Data vs Modelled RTU-3 KVA



Energy savings are calculated by assuming that the baseline RTU operation is equivalent to the occupied mode RTU operating profile in the above figures, however the RTUs are in occupied mode 24/7 all year in the baseline. These KVA profiles are converted to kW assuming a power factor of 0.9 during occupied mode and it is assumed that the power factor is 0.5 during unoccupied periods.

On Peak Savings Calculations

The on-peak energy and peak demand savings were calculated with the same 8,760 spreadsheet model that was used to calculate annual savings. Due to the implemented schedules, no on-peak demand savings are evaluated for this project. The annual on-peak energy savings percentage is 27.2%. On peak savings result from RTU-2 operating schedule achieving savings between 8PM-11PM on weekdays and RTU-3 operating schedule achieving savings between 9PM-11PM on weekdays.

Bill Data Discussion

Office building whole building electric demand is calculated using the interval data summarized in Table 2-8. The difference between the utility meter and the new building submeter data is expected to be mostly office building usage. The site estimates that the old lab building accounts for 7% of the difference and the office building accounts for 93% of the difference between these two sets of interval data. Based on this information the evaluated savings are 4.9% of the office building electric usage based on a 3-year average of annual electric usage. Bill data analysis is not useful in assessing this magnitude of savings.

3 FINAL RESULTS

This section summarizes the evaluation results determined in the analysis above. This section includes a summary table of savings by major end-use and application.

Table 3-1. Summary of Key Parameters

Parameter	BASELINE Tracking Value(s)	BASELINE Evaluation Value(s)	PROPOSED / INSTALLED Tracking Value(s)	PROPOSED / INSTALLED Evaluation Value(s)
Supply Fan Demand kW	13.5	RTU-1: 5.65 RTU-2: 5.79 RTU-3: 2.97 Total: 14.41	13.5	RTU-1: 5.65 RTU-2: 5.79 RTU-3: 2.97 Total: 14.41
Annual RTU Operating Hours	8,736	8,760	5,460	RTU-1: 7,818 RTU-2: 6,097 RTU-3: 6,045
Average RTU Cooling Demand kW	12.7 (63.5 kW * 20% Saving Factor)	RTU-1: 4.25 RTU-2: 3.61 RTU-3: 2.00 Total: 9.86	12.7 (63.5 kW * 20% Saving Factor)	RTU-1: 4.33 RTU-2: 3.89 RTU-3: 2.10 Total: 10.32
Annual Cooling Hours	3,360	RTU-1: 4,054 RTU-2: 3,021 RTU-3: 6,164	2,100	RTU-1: 3,707 RTU-2: 2,268 RTU-3: 4,365

3.4 Explanation of Differences

This section describes the key drivers behind any difference in the application and evaluation estimates, annual kWh savings. The following table summarizes these differences. The purpose of this table is to describe how changes to the key parameters influenced the final project savings through the end-use summary analysis. Table 3-2 provides a summary of the differences between tracking and evaluated values.

Table 3-2. Summary of Deviations

Measure	Discrepancy	Parameter	Impact of Deviation	Discussion of Deviations
M1	Operational – Hours of operation	RTU Schedule - Fan Hours	-30.9%	**Decreased savings –** Runtime reduction from equipment scheduling was less than expected per applicant analysis.
M1	Operational – Operating Load	Cooling Profile	-17.1%	**Decreased savings –** RTU cooling demand profile is less than assumed in applicant analysis.
M1	Non-Operational – Operating Load	No Exhaust Fan Savings	-1.2%	**Decreased savings –** Exhaust fans were not part of the EMS project per feedback from the site during the site interview.
Final RR				**-49.1%**

3.5 Lifetime Savings

The evaluators calculated applicant and evaluated lifetime savings values using the following formula:

$$\text{Lifetime Savings kWh} = \text{Annual Savings kWh} * \text{Measure Lifetime Years}$$

The evaluated lifetime savings are smaller than the tracking lifetime savings because the evaluated first-year savings are smaller than the tracking first-year savings. Table 3-3 provides a summary of key factors that influence lifetime savings. The evaluator assumes that the tracking lifetime savings match the lifetime savings from the BCR.

Table 3-3. Measure 11982673 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	643,150 kWh	643,150 kWh	327,180kWh
First year savings	64,315 kWh	64,315 kWh	32,718 kWh
Measure lifetime	10 years	10 years (project BCR)	10 years (TRM)
Baseline classification	Retrofit	Retrofit	Retrofit

3.5.1 Ancillary impacts

This measure includes gas savings. The evaluation finding that RTU runtime reduction is less than expected for this measure will impact gas savings. The finding that the average cooling demand for the RTUs serving the building is less than expected per the applicant analysis may indicate that the heating loads will also be less than expected per the applicant analysis which would result in less gas savings as well.

RHODE ISLAND CUSTOM ELECTRIC SITE-SPECIFIC REPORT
SITE ID: RICE22N087

Report Date: December 22, 2023

Program Administrator	Rhode Island Energy	
Application ID(s)	13815325	
Project Type	C&I Existing Building Retrofit	
Evaluation Type	Ops	
Program Year	PY2022	
Evaluation Firm	DNV	
Evaluation Engineer	Shaobo Feng	
Senior Engineer	George Sorin Ioan	

1 EVALUATED SITE SUMMARY AND RESULTS

The evaluated project was implemented at a supermarket through the GrocerSmart initiative and consisted of adding doors onto refrigerated cases. According to the project files, the baseline is pre-existing vertical open cases. The measure saves energy because less refrigerated air leaks out from the cases, resulting in a load reduction for the refrigeration system.

During the initial interview with the site contact, evaluators learned the following:

- The site contact is present on-site and agreed to accommodate an on-site evaluation.
- It is safe to visit the facility and inspect the measure.

Based on the information gathered during the initial interview with the site contact, the evaluator proposed this site be evaluated using Schedule 3: Base + Add-on #3 – on-site verification with full M&V where an on-site audit was used to verify measure installation and operation and spot measurements were used to capture the key parameters to conduct the operational adjustments. After reviewing the tracking files and information gathered during the site visit, the evaluator classified this measure as an add-on with single baseline and calculated the project savings using the same eQUEST modeling as the tracking estimate but updating input parameters based on on-site findings. The evaluation results are presented in Table 1-1.

Table 1-1. Evaluation results summary

PA application ID	Measure name		Annual electric energy (kWh)	% of energy savings on-peak	Summer on-peak demand (kW)	Winter on-peak demand (kW)
12106805	Refrigerated case doors	Tracked	56,248	40.0%	10.82	13.31
		Evaluated	60,954	40.0%	8.7	9.4
		Realization Rate	108%	100%	80%	71%

1.1 Explanation of deviations from tracking

The evaluated savings are 8% greater than the tracked savings primarily due to updates of operational parameters from on-site spot measurements. The applicant relied on an eQuest calculation model based on pre-installation site audit and less site-specific data instead of making adjustments based on the post installation site specifications and actual site measurements. Further details regarding deviations from the tracked savings are presented in Section 3.1.

1.2 Recommendations for program designers and implementers

The evaluator recommends that the implementer should perform a post installation site-specific model update. By conducting a site-specific adjustment, the implementer can account for factors such as the actual values of installed measure parameters, building layout, usage patterns, and other site-specific variables that may impact the estimated savings.

1.3 Customer alert

There is no relevant customer alert.

2 EVALUATED MEASURES

The following sections present the evaluation procedure, including the findings from an in-depth review of the supplied applicant calculations and the evaluation methodology determined to be the best fit for the site and the information available.

The evaluated measure for this site is summarized in Table 2-1.

Table 2-1. Evaluated measure

Measure	Project ID	Parameter
M1	13815325	Install doors onto refrigerated cases. The impacted cases had 89 doors, and 2 ft wide each.

2.1 Application information and applicant savings methodology

This section describes the applicant's application information, savings methodology, and the evaluation assessment of the savings calculation algorithm used by the applicant.

2.1.1 Applicant description of baseline

According to the project files, the applicant classified this measure as a retrofit with pre-existing conditions as the baseline. Table 2-2 shows the pre-existing key parameters in the model.

Table 2-2. Applicant baseline key parameters

Measure	Parameter	Operation description	Source of parameter value
M1	Refrigerated case	There were 89 two feet wide pre-existing open refrigerated cases without doors. All cases were kept at medium temperature (34°F to 35°F).	Applicant savings analysis, from the pre-installation energy audit conducted by the implementor.

2.1.2 Applicant description of installed equipment and operation

This project includes installing refrigerator case doors to 89 refrigerated cases. Table 2-3 presents the main parameters of the proposed case as defined by the applicant.

Table 2-3. Applicant's proposed key parameters

Measure	Parameter	Operation description	Source of parameter value
M1	Refrigerated case	There would be 89 two feet refrigerated cases equipped with doors. All cases are kept at medium temperature (34°F to 35°F).	Applicant savings analysis, from the pre-installation energy audit conducted by the implementor.

2.1.3 Applicant energy savings algorithm

The applicant used eQUEST modeling software to calculate the savings, following the GrocerSmart program guidelines. In addition, the applicant used the on-site findings to determine a variety of building inputs including refrigeration system type and efficiency, complex building geometry, lighting systems, and HVAC systems to estimate the energy savings.

The refrigeration specs for the eQUEST model were generated using a proprietary audit tool that maps an on-site data collection to a proprietary database. The database is a collection of refrigeration equipment data and specification sheets that have been modified such that the database outputs are compatible with eQUEST parameter keywords. The database outputs shape the eQUEST model to match observed equipment specifications (e.g., number of refrigeration fixtures, case heat conduction rate, case lighting power, suction groups) that were collected through the on-site audit.

Figure 2-1 presents the input differences between baseline and installed models used to model the energy consumption in eQUEST.

Figure 2-1. Baseline and proposed eQUEST models

Component	Reference(s)	Keyword	Array Idx	Baseline	Run #1
Refrigeration Fixture	C5057D2D68228(1 of 7)	SST-SUPPLY-TD	N/A	7.000	4.000
Refrigeration Fixture	C5057D2D68228(1 of 7)	INF-SCH	N/A	Inf_Sched	Inf_Sched
Refrigeration Fixture	C5057D2D68228(1 of 7)	INF-LOAD/LEN	N/A	1,209.530	302.380
Refrigeration Fixture	C5057D2D68228(1 of 7)	CONDUCTION/LEN	N/A	134.392	60.480
Refrigeration Fixture	C5057D2D68228(1 of 7)	CANOPY-KW/LEN	N/A	0.016	0.018
Refrigeration Fixture	C5957D2D68228	SST-SUPPLY-TD	N/A	8.000	4.000
Refrigeration Fixture	C5957D2D68228	INF-SCH	N/A	Inf_Sched	Inf_Sched
Refrigeration Fixture	C5957D2D68228	INF-LOAD/LEN	N/A	1,154.790	288.700
Refrigeration Fixture	C5957D2D68228	CONDUCTION/LEN	N/A	128.310	57.740
Refrigeration Fixture	C5957D2D68228	CANOPY-KW/LEN	N/A	0.005	0.018

Where,

- SST-SUPPLY-TD: defines the design temperature differential between the wet-bulb temperature leaving the evaporator (supply to the fixture) and the saturated-suction temperature.
- INF-SCH: if infiltration changes over the store schedule (i.e., if the display case has a night cover), this keyword defines a scheduling factor (0 to 1) that modifies INF-LOAD/LEN. In this situation, the proposed case “Inf_Sched” schedule is a typical flat 1.0 profile (has no effect on INF-LOAD/LEN because the proposed case now has a door), while the base case “Night Cover_Sch” reduces INFLOAD/LEN by 0.8 from 11p-6a to simulate the night covers that used to be draped over the display cases during store closures. In the evaluated site, there was no night cover in the base case so this parameter was not changed.
- CONDUCTION/LEN: a per length conduction value for the refrigeration fixture (i.e., display case). It defines the design heat gain due to conduction through the fixture surfaces.
- INF-LOAD/LEN: a per length infiltration value for the refrigeration fixture. It defines the design infiltration heat gain due to infiltration (i.e., air exchange between the fixture and the surrounding zone).
- CANOPY-KW/LEN: a per length lighting power of the refrigeration fixture.

Table 2-4 shows how the length of impacted refrigeration fixture cases were modeled in the eQUEST model.

Table 2-4. Impacted fixture case lengths

Refrigeration system	Refrigeration fixture in eQUEST	Actual refrigeration rack	LINE-UP_LENGTH
C5057D2D68228	SG-F01C6	Rack E	24
C5157D2D68228	SG-3BBBD	Rack C	12
C5257D2D68228	SG-3BBBD	Rack C	36
C5357D2D68228	SG-3BBBD	Rack C	64
C5957D2D68228	SG-89E63	Rack E	28
C6257D2D68228	SG-89E63	Rack E	20
C6457D2D68228	SG-89E63	Rack E	46
C6857D2D68228	SG-3BBBD	Rack C	8
Total			238

Since the total actual added door length (178 ft) is different than model estimated (238 ft) based on the applicant estimated in the pre-existing case, the applicant scaled the savings by applying the ratio between actual length and the model estimated length.

$$Savings = Savings\_unit \times Length$$

Where,

$Savings$	= Applicant calculated tracking savings, 56,248 kWh
$Savings\_unit$	= 316 kWh/ft, eQUEST modeled kWh savings per case feet, based on the average of 238 ft
$Length$	= Actual length of impacted case, 178 ft

2.1.4 Evaluation assessment of applicant methodology

The evaluator determined that the applicant's use of eQUEST to estimate energy savings was appropriate. However, the energy model used inputs were not updated based on the actual installed length of the refrigerated case door. The evaluator updated the models by updating input parameters including lengths and temperatures based on the on-site findings and spot measurement data in Section 2.2.

2.2 On-site inspection and metering

The site contact indicated that it was safe to visit the site and preferred an on-site verification with spot measurement of the evaluated measure. The evaluator conducted the site visit on October 23, 2023, and was assisted by the store manager. Table 2-5 summarizes the findings from the installed measure verification.

Table 2-5. Measure verification

Measure name	Verification method	Verification result
Refrigerated case doors	On-site inspection	The evaluator verified the number and length of all installed case doors and took pictures of the door frame nameplates as shown in Figure 2-2 and Table 2-6.
Refrigerated case doors	Spot measurement	For each case, evaluators collected temperature gauge reading. In addition, evaluators used temperature meters and sensors to spot measure both ambient and case temperature values at two units.
Refrigerated case doors	Inspect installed controls	Evaluators gathered details on the control strategy for the refrigeration racks including suction, saturation, and evaporator temperature.
Refrigerated case doors	Interview the site contact for pre-	The dairy cases were installed in 2020 – 2021

Measure name	Verification method	Verification result
	existing situations	and the remaining units were old but still in operating condition. The store manager confirmed the there were no doors on the pre-existing cases.

Figure 2-2. Example of installed doors on a refrigerated case serving fruit

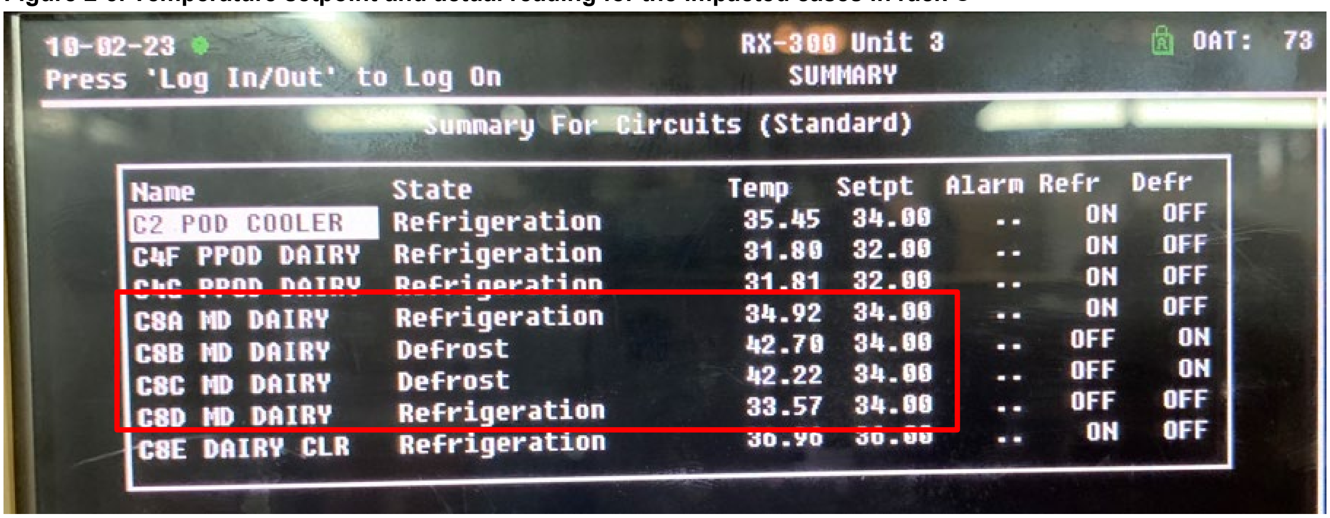
Table 2-6. Verified refrigeration case doors

Case product	Case length (ft)	Verified number of doors
Fruit	6	3
Produce	46	23
Dairy	126	63
Total	178	89

The evaluator's spot measurement for this site included:

1. The evaluator deployed some temperature sensors and use thermal gun in random sales area to spot measure the space temperature. The average space temperature in sales area was 71.95°F.
2. Based on the access to control panel from the refrigeration system (as Figure 2-3 shows), the evaluator collected the case temperature setpoint and the actual case temperature.
3. Collected case gauge temperature reading to cross-verify the actual case temperature in Figure 2-3.

Figure 2-3. Temperature setpoint and actual reading for the impacted cases in rack C



The evaluator updated: the heating set point temperature from the spot temperature measurements; impacted refrigeration case temperature setpoint from the collected system setpoint; and the length of cases impacted by the evaluated project in the eQUEST model as shown in Section 2.3.

2.3 Evaluation methods and findings

This section describes the evaluator methods and findings.

2.3.1 Evaluation description of baseline

Based information provided in the project files and gathered during the site visit, the evaluator determined this measure is an add-on with single baseline. The baseline is single because the measure life (12 years for case doors) is less than 2/3 of the measure life of the underlying refrigeration system (20 years). The baseline is the pre-existing condition.

2.3.2 Evaluation calculation method

The evaluator calculated the project impacts using eQUEST refrigeration software version 3.61 (the same modeling software as used by the applicant) and based on data gathered from the site, updated key input parameters to make the model more site-specific.

The evaluator confirmed there were total of 178 feet of cases equipped with doors. However, instead of incorporating the measure impact for 238 ft of refrigeration fixtures as the applicant did in their model, the evaluator used a total of 178 feet refrigeration fixtures during the parametric run. It should be noted as described above that the applicant model did take into account the fact that the installed cases were 178 feet instead of 238 feet by ratioing the saving down, but this is less accurate than modeling the exact case length in eQuest. Table 2-7 presents the applicant and the evaluated inputs for the refrigerated cases in eQUEST model.

Table 2-7. Impacted fixture case lengths comparison between applicant and evaluated models

Refrigeration system	Refrigeration fixture	Actual refrigeration rack	LINE-UP_LENGTH in applicant model	LINE-UP_LENGTH in evaluated model
C5057D2D68228	SG-F01C6	Rack E	24	6
C5157D2D68228	SG-3BBBD	Rack C	12	12
C5257D2D68228	SG-3BBBD	Rack C	36	36
C5357D2D68228	SG-3BBBD	Rack C	64	64
C5957D2D68228	SG-89E63	Rack E	28	Removed from the parametric run
C6257D2D68228	SG-89E63	Rack E	20	Removed from the parametric run
C6457D2D68228	SG-89E63	Rack E	46	46
C6857D2D68228	SG-3BBBD	Rack C	8	14
Total			238	178

In addition, the evaluator updated the temperature setpoint for the impacted cases and the space heating temperature setpoint in the evaluated model, based on the on-site findings.

Table 2-8. Spot measured space temperature

# of spot measurement	Temperature °F
1	74.08
2	73.22
3	74.66
4	68.74
5	69.04
Average	71.95

3 FINAL RESULTS

The evaluated project consisted of installing doors onto 89 two feet wide pre-existing open refrigerated cases. The evaluated savings are greater than the tracking values, primary due to operational adjustments on the loads calculated in eQuest. The parameters impacting the analysis are summarized in Table 3-1.

Table 3-1. Summary of key parameters

	BASELINE		PROPOSED / INSTALLED	
Parameter	Tracking Value(s)	Evaluation Value(s)	Tracking Value(s)	Evaluation Value(s)
Heating temperature setpoint	71°F	72°F	71°F	72°F
Case temperature setpoint for SG-3BBBD (rack C)	35°F	34°F	35°F	34°F
Modeled cases length	238[1] ft	178 ft	238 ft	178 ft

3.1 Explanation of differences

The evaluated savings are 8% greater the tracking values predominantly because of discrepancies in operations and more accurate modeling of the length of added doors. The evaluator found the actual case temperature setpoint was one degree lower than the applicant estimated. This would slightly increase the refrigeration load and lead the measure save more energy. Table 3-2 provides a summary of the differences between tracking and evaluated values.

Table 3-2. Summary of deviations

Measure	Discrepancy	Parameter	Impact of deviation	Discussion of deviations
M1	Methodology	Methodology	+6%	Increased savings – Evaluator used the actual length of cases impacted by this measure instead of scaling the length after the model simulation.
M1	Operation	Operating profile	+2%	Increased savings – the evaluated findings indicate the actual case temperature was 1°F less than the applicant estimated.

3.2 Lifetime savings

The evaluator classified measure both evaluated measures as an add-on with single baseline. The evaluator calculated applicant and evaluated lifetime savings values using the following formula:

$$\text{LAGI} = \text{FYS} \times [\text{RUL} + \text{outyear}\,\% \times (\text{EUL} - \text{RUL})]$$

where:

[1] The applicant model did take into account the fact that the installed cases were 178 feet instead of 238 feet by ratioing the saving down, but this is less accurate than modeling the exact case length in eQUEST.

LAGI =	lifetime adjusted gross impact (therms)
FYS =	first year savings (therms)
EUL =	measure life (years)
RUL =	1/3 of EUL (years)
outyear % =	100% for this single baseline measure

The evaluated lifetime savings are lower than the tracking lifetime savings because the evaluated first year savings are lower than the tracking first year savings. Table 3-3 provides a summary of key factors that influence the lifetime savings.

Table 3-3. Measure M1 – Application ID: 13815325 – Lifetime savings summary

Factor	Tracking	Application	Evaluator
Lifetime savings (kWh)	731,224	731,224	792,402
First-year savings (kWh)	56,248	56,248	60,954
Measure lifetime (years)	13	13	13
Baseline classification	Retrofit	Retrofit	Add-on retrofit

3.3 Ancillary impacts

A total of 6,820 therms of natural gas savings resulted from installing the evaluated measure.

RHODE ISLAND CUSTOM ELECTRIC SITE-SPECIFIC REPORT
SITE ID: RICE22N093

Report Date: March 22, 2024

Program Administrator	Rhode Island Energy
Application ID(s)	Parent: 9397043; child: 11977866
Project Type	C&I New Construction
Evaluation Type	Full M&V
Program Year	PY2022
Evaluation Firm	DNV
Evaluation Engineer	Shaobo Feng
Senior Engineer	George Sorin Ioan

1 EVALUATED SITE SUMMARY AND RESULTS

The evaluated project was implemented at a materials manufacturer that produces plastic film for packaging. The project consisted of installing a 2,200-ton chilled water plant for a new production line. This chiller plant supplies chilled water for both process and cooling loads, with water side economizing function for the process load. The chiller plant consists of two 1,100-ton variable-speed centrifugal chillers with condenser water reset and three cooling towers equipped with variable frequency drive (VFD) controlled fans. Savings are achieved by installing higher efficiency chillers, higher efficiency cooling towers, and advanced chiller plant controls such as free cooling and condensing water temperature reset.

The applicant classified this measure as new construction and used IECC 2015 code requirement as the baseline. They calculated the savings using an Excel workbook. After reviewing the project files and information gathered during the site visit, the evaluator classified this measure as lost opportunity with single baseline and calculated the project savings using the applicant's model with updated input parameters based on on-site findings. The evaluated savings are less than the tracking values mainly because of the operating load differences. The evaluation results are presented in Table 1-1.

Table 1-1. Evaluation results summary

PA application ID	Measure name		Annual electric energy (kWh)	% of energy savings on-peak	Summer on-peak demand (kW)	Winter on-peak demand (kW)
Parent: 9397043; child: 11977866	New chiller plant	Tracked	2,129,265	47%	241.88	324.30
		Evaluated	1,458,260	52%	347.76	147.87
		Realization rate	68%	110%	144%	46%

1.1 Explanation of deviations from tracking

The evaluated savings are 32% lower than the tracked savings, primarily due to updates of operational parameters. The HVAC load and process load are both lower than the estimate used in the tracked savings calculation, based on measured data collected by the evaluator and trend data provided by the customer. Further details regarding deviations from the tracked savings are presented in Section 3.1.

1.2 Recommendations for program designers and implementers

The evaluator recommends conducting a thorough post verification and commissioning to ensure the installed comprehensive control measure, such as free cooling, is running as proposed with an accurate load estimation.

1.3 Customer alert

There is no relevant customer alert.

2 EVALUATED MEASURES

The following sections present the evaluation procedure, including the findings from an in-depth review of the supplied applicant calculations and the evaluation methodology determined to be the best fit for the site and the information available.

The evaluated measure for this site is summarized in Table 2-1.

Table 2-1. Evaluated measure

Measure	Project ID	Parameter
New chiller plant	Parent: 9397043; child: 11977866	Install two 1,100-ton variable speed centrifugal chillers with condenser water reset and three cooling towers equipped with variable frequency drive (VFD) controlled fans.

2.1 Application information and applicant savings methodology

This section describes the applicant's application information, savings methodology, and the evaluation assessment of the savings calculation algorithm used by the applicant.

2.1.1 Applicant description of baseline

The applicant classified this measure as new construction with IECC 2015 code requirements as the baseline. The baseline cooling tower had three 100-hp fans with VFDs and a performance of 82 gpm/hp at 8,250 gpm of condensing water flow rate. This is greater than the minimum requirement (40.2 gpm/hp) for propeller and axial fan cooling towers from IECC 2015 in Figure 2-1.

Figure 2-1. IECC 2015 cooling tower requirement

TABLE C403.2.3(8) MINIMUM EFFICIENCY REQUIREMENTS: HEAT REJECTION EQUIPMENT

EQUIPMENT TYPE[a]	TOTAL SYSTEM HEAT REJECTION CAPACITY AT RATED CONDITIONS	SUBCATEGORY OR RATING CONDITION[i]	PERFORMANCE REQUIRED[b, c, d, g, h]
Propeller or axial fan open-circuit cooling towers	All	95°F entering water 85°F leaving water 75°F entering wb	≥ 40.2 gpm/hp
Centrifugal fan open-circuit cooling towers	All	95°F entering water 85°F leaving water 75°F entering wb	≥ 20.0 gpm/hp
Propeller or axial fan closed-circuit cooling towers	All	102°F entering water 90°F leaving water 75°F entering wb	≥ 14.0 gpm/hp
Centrifugal fan closed-circuit cooling towers	All	102°F entering water 90°F leaving water 75°F entering wb	≥7.0 gpm/hp

Table 2-2 shows the baseline key parameters in the model.

Table 2-2. Applicant baseline key parameters

Parameter	Value	Source of parameter value
Type of chiller	2 x 1,100-ton constant speed centrifugal chillers	Chiller manufacturer spec sheet

Parameter	Value	Source of parameter value
Chiller efficiency	0.5091 kW/ton at 100% load performance	Chiller manufacturer spec sheet
Condenser water temperature control	75°F minimum. No reset and 5-degree approach to outside air wet-blub temperature	Vendor assumed, from the savings calculation workbook
Free cooling	No free cooling	Vendor assumed, from the savings calculation workbook
Type of cooling tower	3 x 100-hp fans with VFD	Manufacturer spec sheet
Cooling tower efficiency	95.4% motor efficiency and 95% VFD efficiency	Vendor assumed, from the savings calculation workbook

2.1.2 Applicant description of installed equipment and operation

This project includes installing two 1,100-ton variable speed centrifugal chillers with condenser water reset and three oversized cooling towers. Both the baseline and proposed cooling towers are equipped with variable frequency drive (VFD) controlled fans. Table 2-3 presents the main parameters of the proposed case as defined by the applicant.

Table 2-3. Applicant's proposed key parameters

Parameter	Operation description	Source of parameter value
Type of chiller	2 x 1,100-ton variable speed centrifugal chillers	Application document
Chiller efficiency	Full load: 0.5410 kW/ton NPLV: 0.3579 kW/ton	Proposed chiller spec sheet
Condenser water temperature (CWT) control	During mechanical cooling, CWT is 5-degree higher than outside air wet-blub temperature (OAT WB) with 50°F as the minimum. When OAT WB is lower than 35°F, CWT is set at 39°F.	Vendor assumed, from proposed control sequence as given in the savings calculation workbook
Free cooling	Water side economizing when OAT is lower than 50°F.	Vendor assumed, from proposed control sequence as given in the savings calculation workbook
Type of cooling tower	3 x 50-hp fans with VFD, satisfying the same capacity as baseline cooling tower	Manufacturer's spec sheet
Cooling tower efficiency	95.4% motor efficiency and 95% VFD efficiency	Vendor assumed, from the savings calculation workbook

2.1.3 Applicant energy savings algorithm

The applicant used a temperature-based bin analysis model to calculate the savings. The formulas used by the applicant are shown below:

$$kWh\ savings = Baseline\ chiller\ plant\ consumption - Proposed\ chiller\ plant\ consumption$$

The chilled water pumps and condensing water pumps in the baseline and proposed cases are identical; the applicant didn't include pump consumption in the savings calculation.

$$Chiller\ plant\ consumption = Chiller\ energy + Cooling\ tower\ energy$$

For the chiller part:

$$Chiller\ energy = Bin\ hours \times Chiller\ loads \times Chiller\ kW/ton$$

$$Chiller\ loads = Process\ load + HVAC\ cooling\ load$$

$$Process\ load = Average\ demand\ (tons)$$

$$HVAC\ cooling\ load = \sum_{All\ AHU} Supply\ air\ cfm \times (Mixed\ air\ enthalpy - Discharge\ air\ enthalpy) \times \frac{4.5}{12000\ Btu/ton}$$

Where,

$Bin\ hours$ is based on the Providence TMY3 data.

$Average\ Demand\ (tons)$ is based on the new production line estimation.

$Chiller\ kW/ton$ is a normalized value, based on the two-factor line regression from chiller load % and the estimated actual condenser water temperature. The input values are from the performance data of the baseline and proposed chiller spec sheets, respectively. In the baseline case, the minimum condensing water temperature is 75°F and in the proposed case, it is 50°F.

$Supply\ Air\ cfm$ is from AHU specs.

Figure 2-2 shows the process load, HVAC load, and overall chilled water load modeled for an entire year, using local TMY3 weather profile.

Figure 2-2. Applicant model – annual chilled water load

Providence Bin Data				Chilled Water System Load Profile				
DB Mid Point (F)	WB (F)	Enthalpy (BTU/lb)	Hours	Process TW Load (Tons)	Process Chilled Load (Tons)	HVAC Load (Tons)	Total Chilled Water Load (Tons)	Ton-Hours
98	71	35	3	270	654	1,348	2,001	6,004
93	74	38	41	270	654	1,555	2,209	90,571
88	73	36	76	270	654	1,466	2,120	161,127
83	70	34	298	270	654	1,284	1,938	577,652
78	68	32	475	270	654	1,135	1,789	849,667
73	66	31	556	270	654	1,025	1,679	933,281
68	63	28	813	270	654	735	1,389	1,129,148
63	57	25	904	270	654	288	942	851,794
58	52	21	647	270	654	-	654	423,116
53	48	19	758	270	654	-	654	495,707
48	43	17	616	270	654	-	654	402,843
43	39	15	740	270	654	-	654	483,935
38	34	13	921	270	654	-	654	602,303
33	29	11	805	270	654	-	654	526,443
28	24	9	387	270	654	-	654	253,085
23	20	7	375	270	654	-	654	245,238
18	15	5	224	270	654	-	654	146,489
13	10	4	96	270	654	-	654	62,781
8	6	3	24	270	654	-	654	15,695
3	2	1	1	270	654	-	654	654

For the cooling tower part:

$$Cooling\ tower\ energy = Bin\ hours \times Fan\ kW$$

$$Fan\ kW = \frac{Fan\ hp \times 0.746\frac{kW}{hp} \times Load\ factor \times Quantity}{Motor\ efficiency \times VFD\ Efficiency} \times \left(\frac{Cooling\ tower\ load}{Tower\ capacity}\right)^3$$

$$Cooling\ tower\ load = Process\ load + HVAC\ cooling\ load + \frac{Chiller\ loads \times Chiller\ efficiency}{3.516\ kW/ton}$$

Where,

$Fan\ hp$ = 100 hp for baseline case, 50 hp for proposed case

$Load\ factor$ = 0.9

$Motor\ efficiency$ = 93%

$VFD\ efficiency$ = 95%

When the temperature falls below 60 °F, there is no HVAC load and the cooling plant only supplies cooling energy to meet the process load. In the baseline case, there is no free cooling feature. In the proposed case, when the outside air temperature falls below 40°F, the water side economizer supplies cooling energy to meet the process load.

The applicant calculated the baseline and proposed systems annual electric energy consumption as if the plant would operate 8,760 hours then reduced the result to operating hours of 8,000 hours per year.

The summer and winter peak kW were based on the temperature-based bin analysis and local TMY3 data, assuming 1:00pm-5:00pm weekday as the peak time.

$$Summer\ peak\ reduction = \frac{June\ peak + July\ peak + August\ peak}{3}$$

$$Winter\ peak\ reduction = \frac{December\ peak + January\ peak}{2}$$

$$Monthly\ peak\ kW = \frac{Monthly\ peak\ kWh}{Monthly\ peak\ hours}$$

Figure 2-3 and Figure 2-4 are the screenshot of the applicant developed workbook for the baseline and installed case. The yellow cells in Figure 2-4 are hard-coded values from the proposed chiller spec sheet, not the value calculated from the above formula.

Figure 2-3. Applicant calculation workbook – baseline case

	Base Chiller Calcs (Carrier 19XR)													
		Chillers					Cooling Towers						Total System	
	Condenser Water Temp (F)	Qty Chillers Running	% Load/ Chiller	kW/Ton	Chiller Power (kW)	Chiller Energy (kWh)	Cooling Tower Load (Tons)	Capacity Multiplier	Total Tower Capacity (Tons)	% Load	Total Fan kW	Total Fan kWh	kW	kWh
95 to 100	76	2	91%	0.505	1,011	3,033	2,559	1.00	3,048	84%	131.5	395	1,143	3,428
90 to 95	79	2	100%	0.512	1,131	46,383	2,801	1.00	3,048	92%	176.9	7,253	1,308	53,636
85 to 90	78	2	96%	0.508	1,078	81,891	2,697	1.00	3,048	88%	157.9	11,999	1,235	93,890
80 to 85	75	2	88%	0.503	975	290,486	2,486	1.00	3,048	82%	123.7	36,852	1,098	327,338
75 to 80	75	2	81%	0.514	920	436,998	2,321	1.00	3,048	76%	100.6	47,788	1,021	484,785
70 to 75	75	2	76%	0.523	877	487,867	2,198	1.00	3,048	72%	85.5	47,550	963	535,417
65 to 70	75	2	63%	0.545	757	615,267	1,874	1.00	3,048	61%	53.0	43,093	810	658,360
60 to 65	75	1	86%	0.507	478	431,857	1,348	1.00	3,048	44%	19.7	17,837	497	449,694
55 to 60	75	1	59%	0.551	360	233,173	1,027	1.18	3,609	28%	5.2	3,396	366	236,569
50 to 55	75	1	59%	0.551	360	273,176	1,027	1.39	4,238	24%	3.2	2,456	364	275,632
45 to 50	75	1	59%	0.551	360	222,001	1,027	1.59	4,853	21%	2.2	1,329	363	223,330
40 to 45	75	1	59%	0.551	360	266,689	1,027	1.79	5,469	19%	1.5	1,116	362	267,805
35 to 40	75	1	59%	0.551	360	331,920	1,027	2.02	6,154	17%	1.1	975	361	332,894
30 to 35	75	1	59%	0.551	360	290,114	1,027	2.23	6,812	15%	0.8	628	361	290,743
25 to 30	75	1	59%	0.551	360	139,471	1,027	2.46	7,497	14%	0.6	227	361	139,698
20 to 25	75	1	59%	0.551	360	135,146	1,027	2.67	8,140	13%	0.5	171	361	135,318
15 to 20	75	1	59%	0.551	360	80,727	1,027	2.88	8,784	12%	0.4	82	361	80,809
10 to 15	75	1	59%	0.551	360	34,597	1,027	3.11	9,469	11%	0.3	28	361	34,625
5 to 10	75	1	59%	0.551	360	8,649	1,027	3.31	10,085	10%	0.2	6	361	8,655
0 to 5	75	1	59%	0.551	360	360	1,027	3.48	10,602	10%	0.2	0	361	361
			Total 8760 Hours Savings			4,409,806						223,180		4,632,986
			Adjusted Hours kWh:			4,027,220						203,818		4,231,037

Figure 2-4. Applicant calculation workbook – installed case

	Proposed Chiller Calcs (Trane CVHF1300)													
		Chillers					Cooling Towers						Total System	
	Condenser Water Temp (F)	Qty Chillers Running	% Load/ Chiller	kW/Ton	Chiller Power (kW)	Chiller Energy (kWh)	Cooling Tower Load (Tons)	Capacity Multiplier	Total Tower Capacity (Tons)	% Load	Total Fan kW	Total Fan kWh	kW	kWh
95 to 100	76	2	91%	0.435	870	2,610	2,519	1.00	3,426	74%	49.1	147	919	2,757
90 to 95	79	2	100%	0.506	1,119	45,865	2,797	1.00	3,426	82%	62.0	2,544	1,181	48,409
85 to 90	78	2	96%	0.448	949	72,109	2,660	1.00	3,426	78%	53.4	4,055	1,002	76,164
80 to 85	75	2	88%	0.422	817	243,583	2,441	1.00	3,426	71%	41.2	12,287	859	255,870
75 to 80	73	2	81%	0.412	737	350,236	2,269	1.00	3,426	66%	33.1	15,721	770	365,957
70 to 75	71	2	76%	0.393	659	366,375	2,136	1.00	3,426	62%	27.6	15,362	687	381,736
65 to 70	68	2	63%	0.373	518	421,119	1,806	1.00	3,426	53%	16.7	13,582	535	434,701
60 to 65	62	1	86%	0.272	256	231,474	1,285	1.00	3,426	38%	6.0	5,440	262	236,914
55 to 60	57	1	59%	0.248	162	105,118	970	1.00	3,426	28%	2.6	1,675	165	106,793
50 to 55	53	1	59%	0.217	142	107,321	964	1.00	3,426	28%	2.5	1,927	144	109,247
45 to 50	50	1	59%	0.206	135	82,986	962	1.00	3,426	28%	2.5	1,557	137	84,542
40 to 45	50	1	59%	0.206	135	99,691	962	1.00	3,426	28%	2.5	1,870	137	101,561
35 to 40	39	1	Free Cooling	-	-	-	924	1.00	1,377	67%	34.5	31,729	34	31,729
30 to 35	39	1	Free Cooling	-	-	-	924	1.00	1,377	67%	34.5	27,733	34	27,733
25 to 30	39	1	Free Cooling	-	-	-	924	1.00	1,377	67%	34.5	13,332	34	13,332
20 to 25	39	1	Free Cooling	-	-	-	924	1.00	1,377	67%	34.5	12,919	34	12,919
15 to 20	39	1	Free Cooling	-	-	-	924	1.00	1,377	67%	34.5	7,717	34	7,717
10 to 15	39	1	Free Cooling	-	-	-	924	1.04	1,436	64%	30.4	2,918	30	2,918
5 to 10	39	1	Free Cooling	-	-	-	924	1.24	1,714	54%	17.9	429	18	429
0 to 5	39	1	Free Cooling	-	-	-	924	1.41	1,948	47%	12.2	12	12	12
				Total 8760 Hours Savings		2,128,486						172,955		2,301,441
				Adjusted Hours kWh:		1,943,823						157,950		2,101,772

2.1.4 Evaluation assessment of applicant methodology

The evaluator determined that the applicant's temperature-based model is appropriate. The applicant used an ex-ante memo from a different project for the same customer to determine the baseline chiller plant would not include free cooling functionality. The evaluator used the same model with updated input parameters including HVAC load, chiller kW, and cooling tower fan speed based on the collected metered and trend data presented in Section 2.3.

2.2 On-site inspection and metering

The evaluator conducted a site inspection to verify the installation of the incentivized measure and collected BAS trend data on the impacted equipment. This section provides details on the tasks performed during the site visit and on the gathered data.

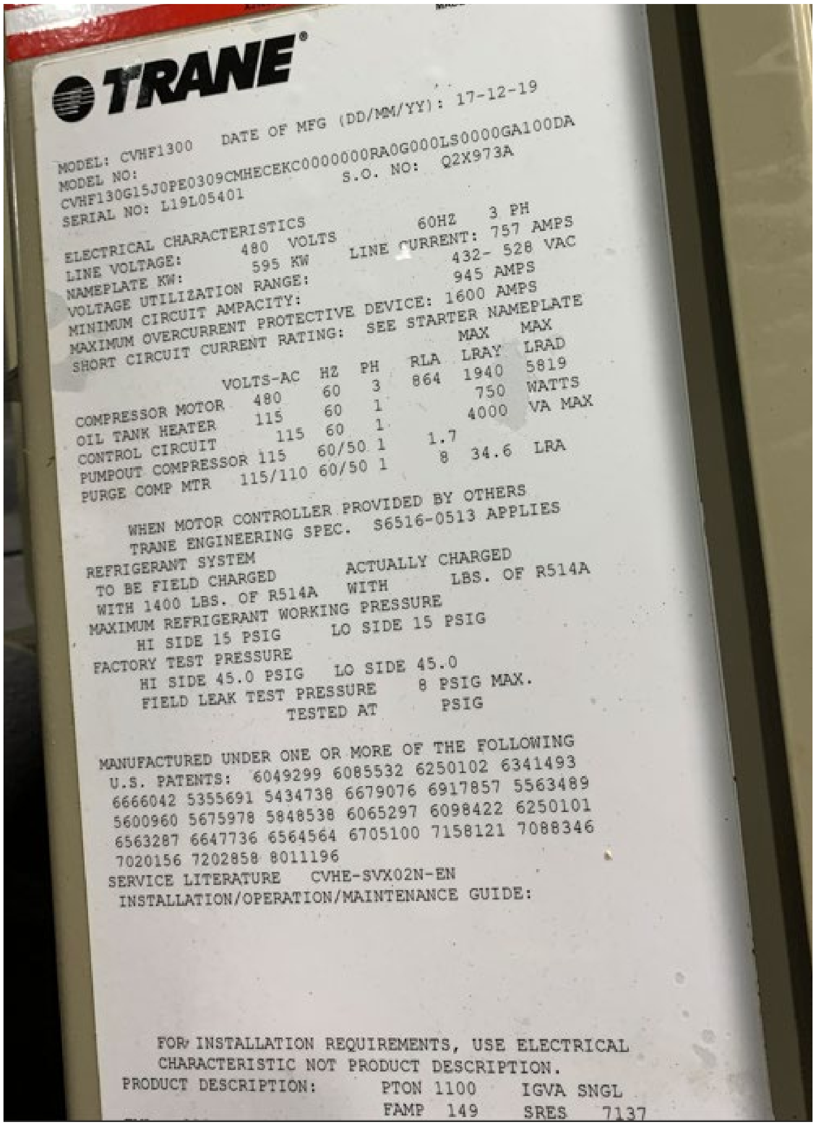
2.2.1 Summary of site visit findings

The site contact indicated that it was safe to visit the site and preferred an on-site verification with installing power meter of the evaluated measure. The evaluator conducted the site visit on August 25, 2023, and was assisted by the facility manager. Table 2-4 summarizes the findings from the installed measure verification.

Table 2-4. Measure verification

Measure name	Verification method	Verification result
New chiller plant	On-site inspection	During the site visit, the evaluator visually verified the installed chiller plant, including 2 new 1,100-ton chillers, new cooling towers with 3 x 50-hp cooling tower fans, and heat exchangers used for free cooling to supply the new production line.
New chiller plant	Nameplate collection	The evaluator collected the nameplate information of the chiller, cooling tower, fans, and heat exchanger and confirmed that model numbers are identical to those the applicant proposed. The installed chiller and its nameplate are shown in Figure 2-6.
New chiller plant	Inspect installed controls	The evaluator accessed the building automation system and gathered details on the control strategy for the new chiller plant. The installed control sequence mainly matched the applicant's proposed case. The evaluator addresses the discrepancy such as chiller operation profile and free cooling temperature threshold in the evaluated analysis in the following section.
New chiller plant	Site interview	During the site visit, the site contact stated that applicant estimated 8,000 hours was underestimated. They normally shut down the chiller plant 1-2 days per month plus extra multi-day maintenance. The evaluator updated the annual operating hours based on collected trended data.
New chiller plant	Power meter installation	The evaluator installed kW meters on two chillers, three cooling tower fans and amperage loggers on 2 of 11 AHUs (AHU-4 and AHU-8A). Detailed information is provided in Section 2.2.2.
New chiller plant	Trend data collection	The evaluator collected one-year trending data on key operational variables from the chiller plant. Detailed information is provided in Section 2.2.2.

Figure 2-5. Installed chiller nameplate



2.2.2 Measured and logged data

With help from the customer's in-house electrician, the evaluator installed kW loggers on both chillers and all three cooling tower fans and installed amperage loggers for AHU-4 (supplying cooling for the office area, with 75°F space air temperature setpoint) and AHU-8A (supplying cooling for the production area, with 85°F space air temperature setpoint). In addition, the evaluator worked with the site contact to go through their energy management system and requested trend data for key variables. The site contact shared trend data from February 2023 to February 2024 for each of the requested variables. Table 2-5 shows the installed loggers and collected trend data information. Due to an unknown issue, the kW meter on cooling tower 2 only captured 5 days of data. To estimate the total kW for the cooling towers, the evaluator calculated the average kW readings from cooling towers 1 and 3 and then multiplied this average by 3.

Table 2-5. Installed loggers with metered variables

Data Type	Description	Unit	Time Interval	From	To	Quantity
DENT kW meter	Chiller kW draw	kW	15-min	9/6/2023	1/11/2024	2
DENT kW meter	Cooling tower fan	kW	15-min	8/25/2023	1/11/2024	3
HOBO meter with CT	AHU-4, AHU-8A current draw	Amp	15-min	8/25/2023	1/11/2024	2
BAS trend data	OAT	°F	15-min	3/1/2023	2/20/2024	1
BAS trend data	Chilled water supply temperature	°F	15-min	2/13/2023	2/20/2024	1
BAS trend data	Chilled water return temperature	°F	15-min	2/13/2023	2/20/2024	1
BAS trend data	Condensing water supply temperature	°F	15-min	2/19/2023	2/20/2024	1
BAS trend data	Condensing water return temperature	°F	15-min	2/19/2023	2/20/2024	1
BAS trend data	Cooling tower #1 fan speed	%	15-min	2/26/2023	2/20/2024	1
BAS trend data	Cooling tower #2 fan speed	%	15-min	2/26/2023	2/20/2024	1
BAS trend data	Cooling tower #3 fan speed	%	15-min	2/26/2023	2/20/2024	1
BAS trend data	AHU-1, 2, 3, 4, 5, 6, 7, 8A, 8B, 9, 10 supply airflow	cfm	15-min	8/12/2023	2/20/2024	1
BAS trend data	AHU-1, 2, 3, 4, 5, 6, 7, 8A, 8B, 9, 10 mix air temperature	°F	15-min	8/12/2023	2/20/2024	1
BAS trend data	AHU-1, 2, 3, 4, 5, 6, 7, 8A, 8B, 9, 10 discharge air temperature	°F	15-min	8/12/2023	2/20/2024	1
BAS trend data	Chiller power	kW	15-min	10/10/2023	4/24/2024	2

The evaluator compared the metered chiller kW from logger data with the trended chiller kW from customer BAS system. The comparisons are shown as Figure 2-6 and Figure 2-7, revealing that the metered kW profiles for both chillers align closely with the customer's trended power data. This consistency underscores the robustness of the evaluated logger data. Both figures also show that chiller 1 is running more frequently than chiller 2 especially during the non-summer periods.

Figure 2-6 Comparison between metered kW and trended kW for chiller 1

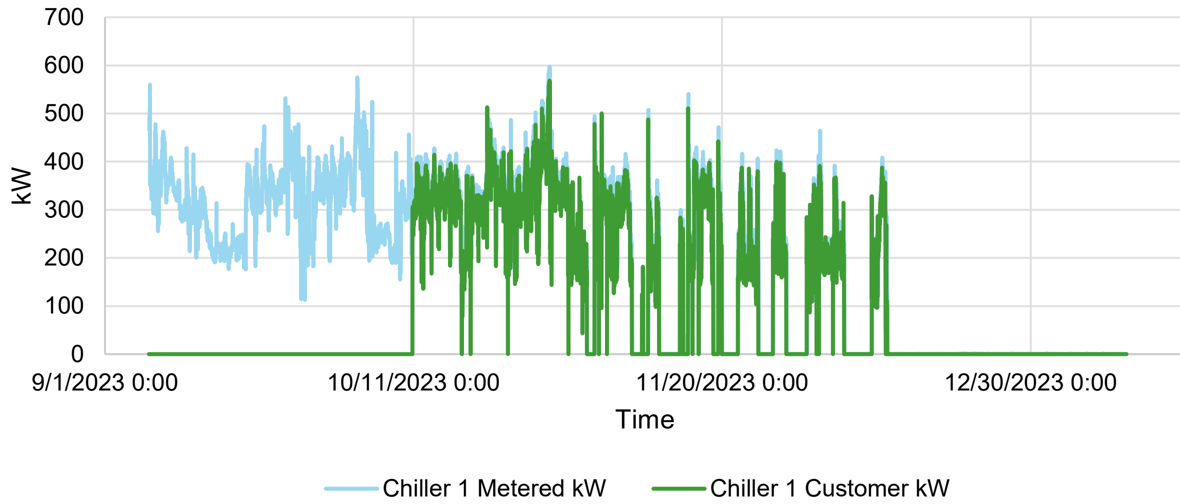


Figure 2-7 Comparison between metered kW and trended kW for chiller 2

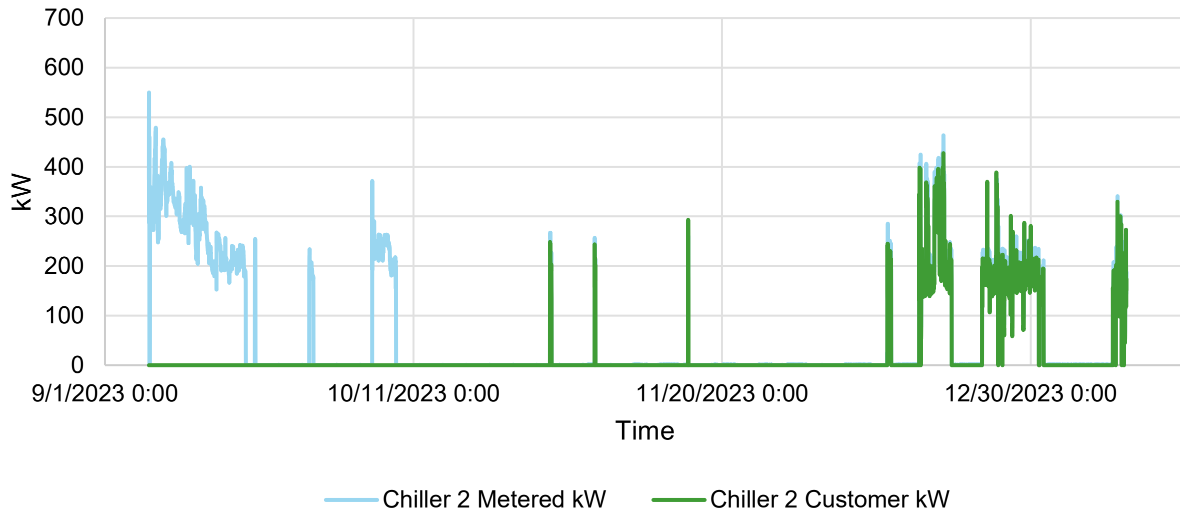
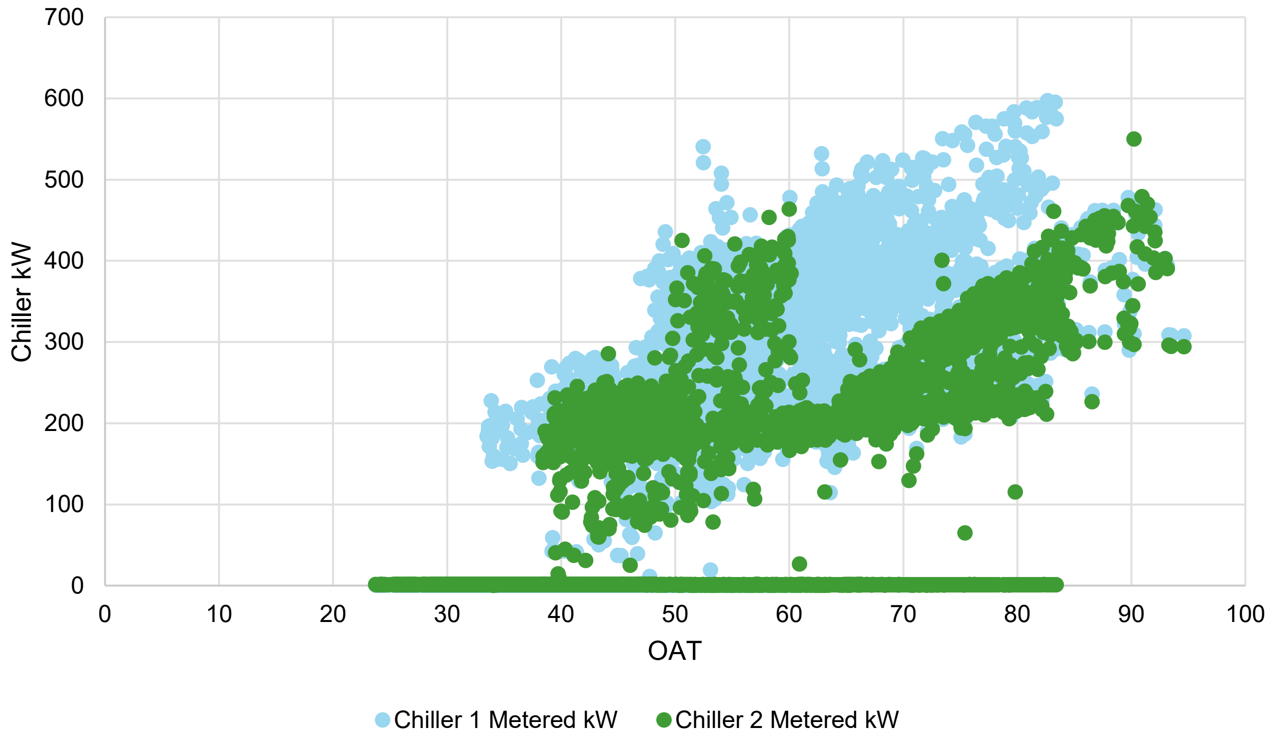


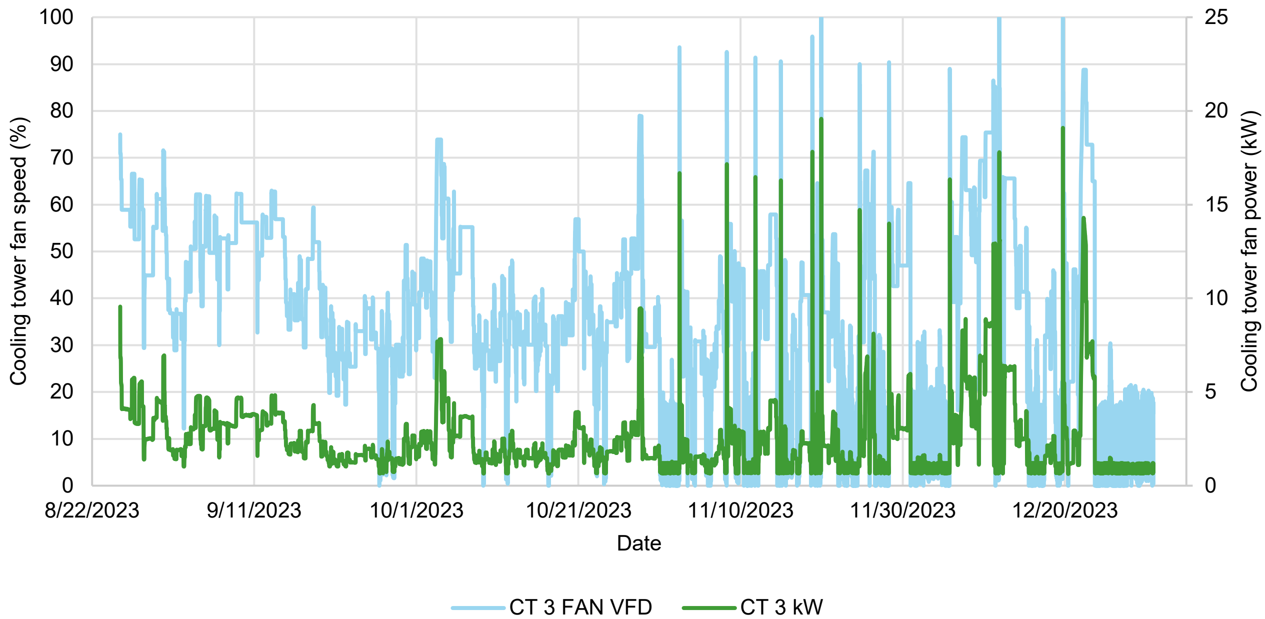
Figure 2-8 plots the charts showing the power for both chillers based on different OAT. Based on the collected data, the evaluator developed a bin profile for all collected metered and trend data using the same 5-degree temperature bin as applicant used. For each bin profile, the evaluator calculated the average metered chiller kW value when it was running, and extrapolated the operational hours from the metered period to an annual profile of 8,760 hours for both chillers in the installed case, based on the metered data.

Figure 2-8. Comparison between OAT and metered chiller kW



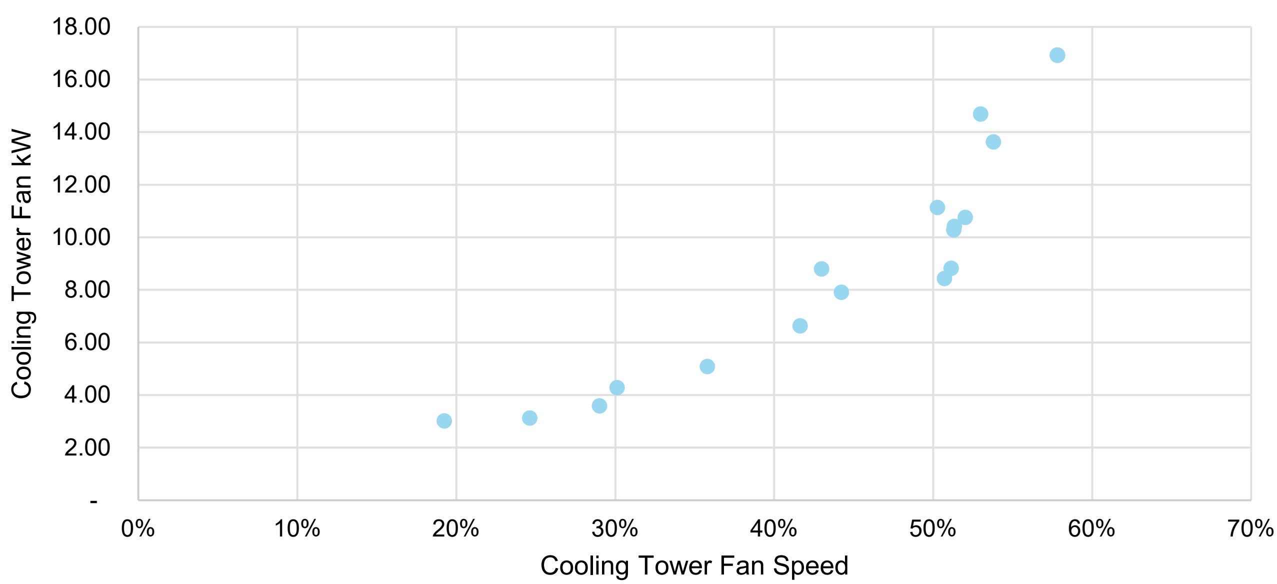
The evaluator compared the metered cooling tower kW with trended cooling tower fan VFD speed, and their trends followed each other closely as Figure 2-9 shows. Since the trended cooling tower speed data was metered for a longer period of time and the evaluator verified the accuracy with the comparison with metered data, trend data was used to generate the bin profile for the cooling tower load in the installed case.

Figure 2-9. Comparison between metered cooling tower 3 kW and trended cooling tower 3 fan speed



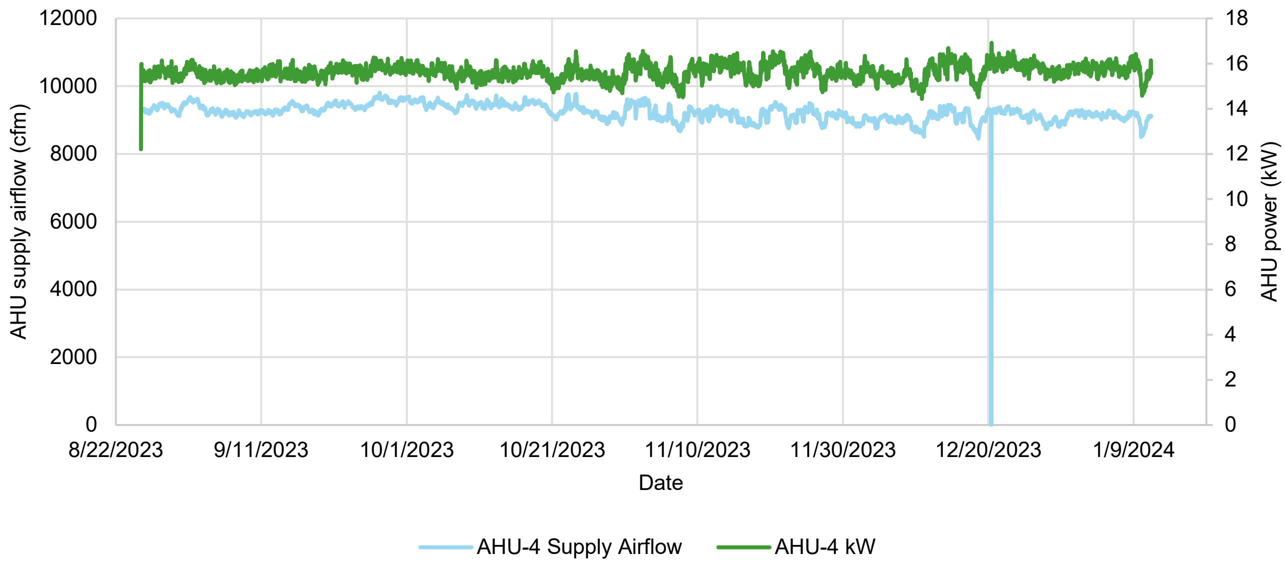
Using the same 5-degree OAT bin framework, the evaluator computed the average cooling tower fan speed from trended data and the average kW drawn from metered data for each cooling tower within each bin profile, then used the average speed and kW as the cooling tower fan speed and kW, for the installed case cooling tower operation profile in the savings model. Figure 2-10 shows the comparison between total cooling tower fan power and the average speed. For baseline cooling tower fan power, the evaluator used the same algorithm used by the applicant, which includes a baseline cooling tower with larger fans, by considering the chilled water load, process cooling tower load, and the heat gain from the chiller(s).

Figure 2-10 Comparison between total cooling tower fan power and the average speed



The evaluator plotted the metered kW and trended supply airflow for AHU-4 in Figure 2-11. Except for a zero-cfm reading in 12/20/2023 10:00 am, the overall trends for both lines follow the same pattern. Therefore, the evaluator used the trended cfm for all AHUs to generate the HVAC load bin profile, instead of extrapolating two metered AHUs current to all units.

Figure 2-11. Comparison between metered AHU-4 kW and trended AHU-4 supply airflow



2.3 Evaluation methods and findings

This section describes the evaluator methods and findings.

2.3.1 Evaluation description of baseline

Based information provided in the project files and gathered during the site visit, the evaluator classified this measure as a lost opportunity with single baseline. The baseline is ISP, and the ISP is IECC2015 code requirement for the chiller plant efficiency. The baseline cooling tower as evaluated is the same as that used by the applicant as it satisfies code minimum requirement in Figure 2-1. The baseline cooling tower is smaller than the installed though with more fan horsepower.

Table 2-6. Evaluated baseline key parameters

Parameter	Value	Source of parameter value
Type of chiller	2 x 1,100-ton constant speed centrifugal chillers	Match installed capacity
Chiller efficiency	0.5091 kW/ton at 100% load performance	ISP, IECC 2015
condenser water temperature control	75°F. No reset and 5-degree approach to outside air wet-blub temperature	ISP, IECC 2015
Free cooling	Without free cooling	ISP, ex-ante memo for the same customer

Parameter	Value	Source of parameter value
Type of cooling tower	3 x 100-hp fans with VFD	Applicant spec sheet for tower which meets ISP, IECC 2015
Cooling tower efficiency	82 gpm/hp at 8,250 gpm of condensing water flow rate	ISP, IECC 2015

2.3.2 Evaluation calculation method

The evaluator calculated the project impacts using the same Excel workbook that the contractor developed, updated with data and key input parameters gathered from the site, to make the model reflect the actual operation sequence more accurately.

For HVAC load, the evaluator used the same equation as application by using the enthalpy difference between mixed air and discharge air to estimate the cooling load, to cover both sensible and latent heat.

$$HVAC\ cooling\ load = \sum_{All\ AHU} SAF \times (MAE - DAE) \times \frac{4.5}{12{,}000\ Btu/ton}$$

Where,

SAF = Supply airflow in cfm, the average of the collected data for each AHU from 15-min trend data

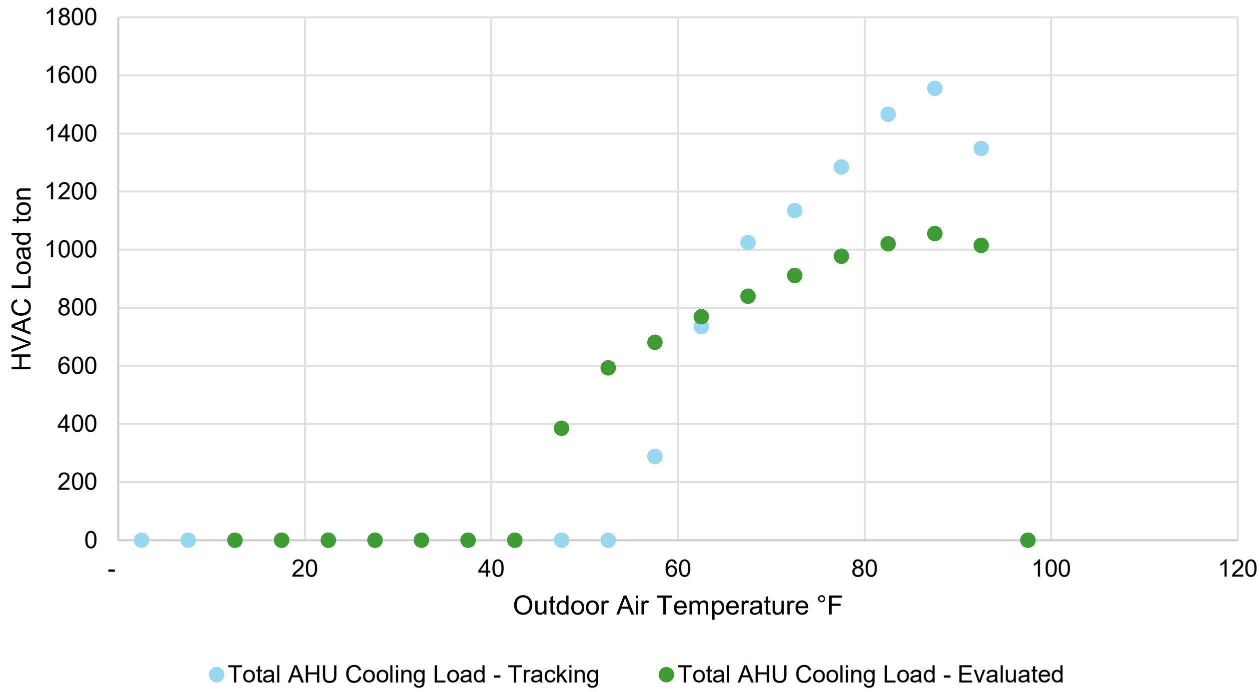
MAE = Mixed air enthalpy in btu/lb. Because the trended data does not cover the relative humidity ratio (RH%), the evaluator created a regression between applicant estimated mixed air dry-bulb temperature and the enthalpy for each AHU, and applied the correlation to the evaluated mixed air dry bulb temperature from trended data to get the evaluated mixed air enthalpy.

DAE = Discharge air enthalpy in btu/lb. The evaluator first used psychrometric chart to get the discharge air RH%, based on the applicant estimated dry-blub and wet-bulb temperature. Then the evaluator estimated the evaluated DAE based on this RH% and the trended discharge air dry-bulb temperature.

4.5 = HVAC factor for typical building air; 4.5 cfm·lb/hr

$12{,}000$ = Conversion factor for 12,000 Btu/hr = 1 ton refrigeration

Figure 2-12 Comparison between tracking HVAC load and evaluated HVAV load



The evaluator developed a two-factor linear relationship between chiller ton, condensing water supply temperature (CWST)[1] and chiller kW derived from the proposed chiller performance table in the spec sheet. The formula is as below:

$$Chiller\ tons = a \times Chiller\ kW + b \times CWT + c$$

With an R-square value of 0.9266 and a multiple R value of 0.9626, the regression demonstrates the effectiveness in explaining and predicting the chiller tons based on both CWST and kW as both CWST and chiller kW are strongly correlated with the chiller ton when considered together. Based on the metered chiller kW and trended CWST, the evaluator calculated the chiller load in tons by applying the regression equation and calculated the average installed chilled water load for each 5F OAT bin. Subsequently, the evaluator calculated the chiller process load ton by subtracting the HVAC load, calculated as described above, from the total chilled water load. The chiller process load ton is used to estimate cooling tower process load, and the total chilled water load is also used to estimate the baseline cooling tower load in the following paragraph. The evaluated chiller load is used to estimate the chiller kW in the base case, following the same regression result as applicant did from the baseline chiller performance table. The baseline chiller operation hours are based on the evaluator collected cooling tower operation profile, since there was no free cooling in the base case and cooling tower operation can reflect the chiller plant operation to satisfy both process load and HVAC load.

The evaluator calculated evaluated/applicant estimation ratio for process chilled load as 63%, by dividing the applicant estimated value by the evaluated value. And the evaluator applied this 63% to get the evaluated process cooling tower load (in tons) to estimate the cooling tower consumption in the base case.

For condensing water temperature setpoint and free cooling control logic, the evaluator determined the free cooling operation is enabled when outside air temperature falls below 42°F, and disabled when outside air temperature is above

[1] Including the load profile for 80°F, 75°F, 70°F, 65°F, 60°F, 55°F and 50°F CWS.

50°F, with 90 second delays, as Figure 2-13, Figure 2-14, and Figure 2-15 show. This aligns with the metered chiller operation status seen at varying OAT conditions.

Figure 2-13. Free cooling logic in BAS – activation threshold

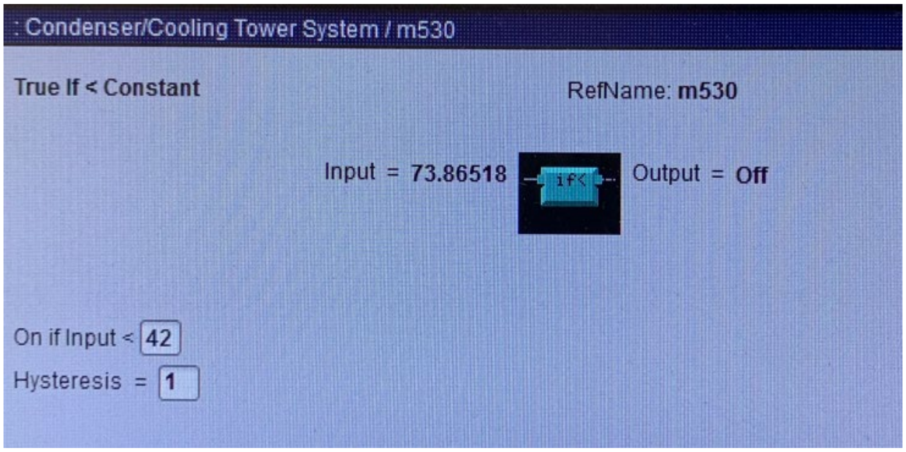


Figure 2-14. Free cooling logic in BAS – deactivation threshold

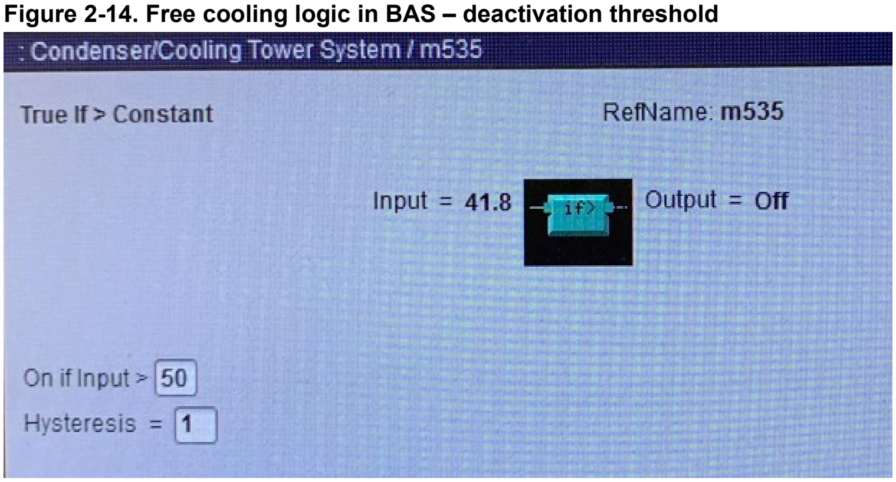
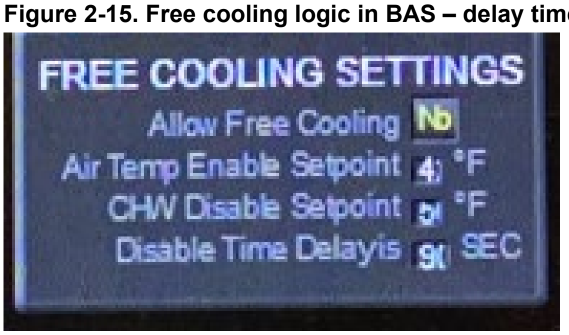


Figure 2-15. Free cooling logic in BAS – delay time



The evaluator compared the temperature difference between supply and return chilled water, and supply and return condensing water temperature to determine the annual operating hours for the chiller plant. When the chilled water return temperature is 0.5°F or more higher than the supply chilled water, and the condensing water return temperature is also 0.5°F or more higher than the supply condensing water at the same time, we considered the chiller plant is in operational status. Then we extrapolated the trend period into 8,760 hour and got the updated annual operation of 8,505 hours per year.

For peak demand savings calculations, the evaluator used the same methodology as that developed by the applicant and re-calculated savings by updating the variables discussed above.

Figure 2-16, Figure 2-17, and Figure 2-18 are screenshots of the evaluator updated workbook, with highlighted in red as the updated variables from collected data. Figure 2-16 represents the evaluated load for both baseline and installed cases. Figure 2-17 and Figure 2-18 demonstrate the chiller and cooling tower consumption in the base and installed cases, respectively.

Figure 2-16. Evaluator updated calculation workbook – chiller plant load

	Providence Bin Data							Chilled Water System Load Profile				
	DB Mid Point (F)	WB (F)	Enthalpy (BTU/lb)	# of Hours	Installed Chiller 1 Hours	Installed Chiller 2 Hours	Installed CT Hours	Process TW Load (Tons)	Process Chilled Load (Tons)	HVAC Load (Tons)	Total Chilled Water Load (Tons)	Ton-Hours
95 to 100	98	71	35	3	3	3	3	171	594	1,015	1,608	4,825
90 to 95	93	74	38	41	41	41	41	171	553	1,055	1,608	65,939
85 to 90	88	73	36	76	76	76	76	171	549	1,020	1,568	119,203
80 to 85	83	70	34	298	298	243	298	171	378	977	1,356	403,962
75 to 80	78	68	32	475	475	358	475	171	345	911	1,256	596,788
70 to 75	73	66	31	556	556	322	556	171	261	840	1,101	612,039
65 to 70	68	63	28	813	813	391	813	171	253	769	1,023	831,421
60 to 65	63	57	25	904	899	87	903	171	202	680	883	797,980
55 to 60	58	52	21	647	597	81	647	171	222	593	815	527,351
50 to 55	53	48	19	758	613	132	750	171	336	385	721	546,304
45 to 50	48	43	17	616	308	199	574	171	507	-	507	312,313
40 to 45	43	39	15	740	138	222	672	171	494	-	494	365,235
35 to 40	38	34	13	921	44	36	829	171	508	-	508	467,770
30 to 35	33	29	11	805	16	0	758	171	440	-	440	354,344
25 to 30	28	24	9	387	0	0	366	171	440	-	440	170,349
20 to 25	23	20	7	375	0	0	375	171	440	-	440	165,067
15 to 20	18	15	5	224	0	0	224	171	440	-	440	98,600
10 to 15	13	10	4	96	0	0	96	171	440	-	440	42,257
5 to 10	8	6	3	24	0	0	24	171	440	-	440	10,564
0 to 5	3	2	1	1	0	0	1	171	440	-	440	440
Total				8,760	4,876	2,192	8,481		414			
								270	654			
								171	63%			

DNV

Figure 2-17. Evaluator updated calculation workbook – base case

	Base Chiller Calcs (Carrier 19XR)													
		Chillers					Cooling Towers						Total System	
	Condenser Water Temp (F)	Qty Chillers Running	% Load/ Chiller	kW/Ton	Chiller Power (kW)	Chiller Energy (kWh)	Cooling Tower Load (Tons)	Capacity Multiplier	Total Tower Capacity (Tons)	% Load	Total Fan kW	Total Fan kWh	kW	kWh
95 to 100	79	2	73%	0.557	895	2,686	2,034	1.00	3,048	67%	19.1	57	914	2,743
90 to 95	79	2	73%	0.557	896	36,752	2,034	1.00	3,048	67%	19.1	785	916	37,536
85 to 90	78	2	71%	0.550	863	65,530	1,985	1.00	3,048	65%	17.9	1,362	881	66,892
80 to 85	75	2	62%	0.547	742	221,145	1,738	1.00	3,048	57%	12.6	3,766	755	224,911
75 to 80	75	2	57%	0.555	697	331,232	1,626	1.00	3,048	53%	10.7	5,075	708	336,308
70 to 75	75	2	50%	0.567	624	346,885	1,449	1.00	3,048	48%	8.1	4,508	632	351,393
65 to 70	75	2	46%	0.573	586	476,150	1,360	1.00	3,048	45%	7.0	5,713	593	481,863
60 to 65	75	2	40%	0.584	515	465,415	1,200	1.00	3,048	39%	5.4	4,897	521	470,312
55 to 60	75	1	74%	0.526	429	277,538	1,108	1.18	3,609	31%	3.6	2,329	433	279,867
50 to 55	75	1	66%	0.541	390	292,517	1,003	1.39	4,238	24%	2.8	2,105	393	294,622
45 to 50	75	1	46%	0.574	291	167,028	761	1.59	4,853	16%	2.4	1,483	293	168,511
40 to 45	75	1	45%	0.576	284	190,816	745	1.79	5,469	14%	2.4	1,769	286	192,585
35 to 40	75	1	46%	0.573	291	241,311	762	2.02	6,154	12%	2.4	2,205	294	243,517
30 to 35	75	1	40%	0.584	257	194,901	684	2.23	6,812	10%	0.2	186	257	195,087
25 to 30	75	1	40%	0.584	257	94,106	684	2.46	7,497	9%	0.2	67	257	94,173
20 to 25	75	1	40%	0.584	257	96,363	684	2.67	8,140	8%	0.1	51	257	96,413
15 to 20	75	1	40%	0.584	257	57,561	684	2.88	8,784	8%	0.1	24	257	57,585
10 to 15	75	1	40%	0.584	257	24,669	684	3.11	9,469	7%	0.1	8	257	24,677
5 to 10	75	1	40%	0.584	257	6,167	684	3.31	10,085	7%	0.1	2	257	6,169
0 to 5	75	1	40%	0.584	257	257	684	3.48	10,602	6%	0.1	0	257	257
			Total 8760 Hours Savings			3,589,029						36,393		3,625,421

Figure 2-18. Evaluator updated calculation workbook – installed case

	Proposed Chiller Calcs (Trane CVHF1300)														
		Chillers						Cooling Towers						Total System	
	Condenser Water Temp (F)	Qty Chillers Running	% power/ Chiller	kW/Ton	Chiller 1 Power (kW)	Chiller 2 Power (kW)	Chiller Energy (kWh)	Cooling Tower Load (Tons)	Capacity Multiplier	Total Tower Capacity (Tons)	% Load	Total Fan kW	Total Fan kWh	kW	kWh
95 to 100	83	2	73%	-	406	405	2,433	1,722	1.00	3,426	50%	11.13	33	417	2,467
90 to 95	83	2	66%	-	406	405	33,257	1,751	1.00	3,426	51%	8.81	361	415	33,618
85 to 90	83	2	67%	-	406	395	60,866	1,737	1.00	3,426	51%	8.43	640	414	61,506
80 to 85	79	2	55%	-	369	331	190,428	1,759	1.00	3,426	51%	10.41	3,103	379	193,531
75 to 80	77	2	51%	-	344	306	272,581	1,782	1.00	3,426	52%	10.76	5,109	355	277,690
70 to 75	74	2	44%	-	320	263	262,710	1,757	1.00	3,426	51%	10.27	5,711	330	268,421
65 to 70	72	2	38%	-	314	226	343,746	1,515	1.00	3,426	44%	7.91	6,429	322	350,175
60 to 65	72	2	33%	-	339	198	321,880	1,226	1.00	3,426	36%	5.08	4,591	344	326,472
55 to 60	71	1	46%	-	319	274	212,677	994	1.00	3,426	29%	3.58	2,318	323	214,995
50 to 55	71	1	40%	-	293	238	211,133	967	1.15	3,928	25%	3.12	2,344	296	213,477
45 to 50	71	1	32%	-	214	192	104,126	897	1.36	4,662	19%	3.01	1,731	217	105,857
40 to 45	70	1	31%	-	208	187	70,163	1,552	1.50	5,155	30%	4.29	2,879	212	73,042
35 to 40	67	1	29%	-	189	169	14,542	2,205	1.55	5,299	42%	6.63	5,494	195	20,037
30 to 35	72	1	29%	-	185	1	2,875	3,010	2.04	7,005	43%	8.80	6,671	194	9,545
25 to 30	41	-	Free Cooling	-	0	1	-	729	1.00	1,377	53%	14.69	5,379	15	5,379
20 to 25	42	-	Free Cooling	-	1	1	-	740	1.00	1,377	54%	13.63	5,110	14	5,110
15 to 20	42	-	Free Cooling	-	-		-	801	1.01	1,386	58%	16.92	3,791	17	3,791
10 to 15	42	-	Free Cooling	-	-		-	980	1.23	1,695	58%	16.92	1,625	17	1,625
5 to 10	42	-	Free Cooling	-	-		-	1,141	1.43	1,974	58%	16.92	406	17	406
0 to 5	42	-	Free Cooling	-	-		-	1,276	1.60	2,207	58%	16.92	17	17	17
			Total Annual Consumption				2,103,418						63,743		2,167,162

The total evaluated first year savings is 1,458,260 kWh. Table 2-7 lists the difference between applicant estimated and evaluated chiller and cooling tower consumption.

Table 2-7. Comparison between applicant and evaluated system consumption

End use	Applicant Baseline kWh	Evaluated Baseline kWh	Applicant Installed kWh	Evaluated Installed kWh
Chiller	4,027,220	3,589,029	1,943,823	2,103,418
Cooling Tower	203,818	36,393	157,950	63,743
Total	**4,231,037**	**3,625,421**	**2,101,772**	**2,167,162**

3 FINAL RESULTS

The evaluated project consisted of installing two 1,100 ton variable speed centrifugal chillers with condenser water reset and three cooling towers with variable frequency drive (VFD) controlled fans. The evaluated savings are lower than the tracking values, primarily due to lower HAVC loads than estimated by the applicant. The applicant assumed the production line running at the design load, and used the design airflow for each AHU. However, the collected average airflow for each AHU was lower than the designed speed. The evaluator identified this discrepancy and considered as a load adjustment through long-term metering and trend data collection during the post installation period. The parameters impacting the analysis are summarized in Table 3-1.

Table 3-1. Summary of key parameters

	BASELINE		PROPOSED / INSTALLED	
Parameter	Tracking Value(s)	Evaluation Value(s)	Tracking Value(s)	Evaluation Value(s)
Annual operation hours	8,000	8,505	8,000	8,505
HVAC load for both chillers, ton	From 60°F OAT, between 288 and 1,348	From 50°F OAT, between 385 and 1,015	From 60°F OAT, between 288 and 1,348	From 50°F OAT, between 385 and 1,015
Process chilled load, ton	654	Between 202 and 594	654	Between 202 and 594
Process cooling tower load, ton	270	171	270	171
Chiller kW	Total: all temperature bins, between 360 and 1,131	Total: all temperature bins, between 153 and 896	Total: from 40°F, between 135 and 1,119	Chiller 1: from 30°F, between 185 and 406 Chiller 2: from 35°F, between 169 and 405
Free cooling	No free cooling	No free cooling	When OAT is below 40°F	When OAT is below 42°F
Cooling tower speed	10% - 92%	6% - 67%	28% - 82%	19% - 58%

3.1 Explanation of differences

The evaluated savings are 32% less the tracking values mainly because of the discrepancies in load and operational hours. The evaluator found the installed load to be lower than the applicant estimated, based on the collected metered and trend data. The reduced chilled water load reduces the demand to the chiller plant, reducing consumption and savings. Table 3-2 provides a summary of the differences between tracking and evaluated values.

Table 3-2. Summary of deviations

Measure	Discrepancy	Parameter	Impact of deviation	Discussion of deviations
New chiller plant	Operating load	Operating profile	-38%	**Decreased savings** – The evaluated HVAC load and process load are smaller than the value the applicant used in the tracking savings calculations, primarily because the TA used full design airflow.
New chiller plant	Operating load	Hours of operation	+6%	**Increased savings** – the evaluated annual operation hours are slightly more than the applicant estimated.

3.2 Lifetime savings

The evaluator classified measure both evaluated measures as lost opportunity with single baseline. The evaluator calculated applicant and evaluated lifetime savings values using the following formula:

$$\mathrm{LAGI} = \mathrm{FYS} \times [\,\mathrm{RUL} + \text{outyear}\,\% \times (\mathrm{EUL} - \mathrm{RUL})]$$

where:

LAGI = lifetime adjusted gross impact (therms)

FYS = first year savings (therms)

EUL = measure life (years)

RUL = 1/3 of EUL (years)

outyear % = 100% for this single baseline measure

The evaluated lifetime savings are lower than the tracking lifetime savings because the evaluated first year savings are lower than the tracking first year savings. Table 3-3 provides a summary of key factors that influence the lifetime savings.

Table 3-3. Application ID: Parent: 9397043; child: 11977866 – Lifetime savings summary

Factor	Tracking	Application	Evaluator
Lifetime savings (kWh)	42,585,300	42,585,300	29,165,194
First-year savings (kWh)	2,129,265	2,129,265	1,458,260
Measure lifetime (years)	20	20	20
Baseline classification	New construction	New construction	Lost opportunity

3.3 Ancillary impacts

There are no ancillary impacts for this evaluated project.

RI CUSTOM ELECTRIC EVALUATION SITE-SPECIFIC REPORT

DNV SITE ID: RICE22N095

Report Date: 6/5/2024

Application ID(s)	11655059/13321748
Project Type	Retrofit
Program Year	2022
Evaluation Firm	DMI
Evaluation Engineer	Zac Cragan
Senior Engineer	Mickey Bush

1 EVALUATED SITE SUMMARY AND RESULTS

This retrofit project consisted of two energy efficiency measures at a university building. The building is a 20,800 ft² research and lab facility with peak occupancy during the typical school year. The project, TA study, and post inspection were completed in late 2021 and early 2022; therefore, the project was split into parent (11655059) and child (13321748) applications. The evaluated savings in Table 1-1 are the total parent and child application savings. The evaluated energy savings measures are as follows:

M1 New Controls for FCUs, AHUs, Air Valves – This measure covered the installation of a new energy management system which includes controls to decrease the laboratory space airflow served by AHU-4 by 50% during unoccupied hours. The new controls were also proposed to include AHU-4 discharge air temperature reset from 53.7°F to 65°F during unoccupied hours as well as a shutoff of fan coil units during unoccupied hours.

M2 Hot Water Pump VFD – This measure covered the replacement of pneumatically controlled fan coil unit hot water valves with new electronic control valves and the installation of variable speed drives on the two 20-hp hot water pumps.

The electric and gas savings associated with these two measures were calculated by the applicant using a custom spreadsheet. The evaluated savings for this project are based on metered data collected by the evaluator.

The evaluated savings are less than the tracking savings for this project primarily because of lower chilled water savings than calculated by the applicant and a lower AHU-4 fan baseline power than used by the applicant.

The evaluation results are presented in Table 1-1.

Table 1-1. Evaluation Results Summary

PA Application ID	Measure Name		Annual Electric Energy (kWh)	% of Energy Savings On-Peak	Summer On-Peak Demand (kW)	Winter On-Peak Demand (kW)
11655059/13321748	M1- New Controls for FCUs, AHUs, Air Valves	Tracked	141,661	46%	2.63	19.10
		Evaluated	52,050	11%	0.0	0.0
		Realization Rate	36.7%	23%	0%	0%
11655059/13321748	M2-HW Pump VFD	Tracked	36,047	53%	7.3	1.55
		Evaluated	16,920	50%	3.54	0.44
		Realization Rate	46.9%	95.4%	49%	21%
Total		Tracked	177,708	47.42%	9.93	20.65
		Evaluated	68,970	20.27%	3.54	0.44
		Realization Rate	38.8%	42.75%	35.7%	2.12%

N/A = Not applicable

1.1 Explanation of Deviations from Tracking

The evaluated savings are 61.2% less than the applicant-reported savings. The evaluated savings are less than the tracking savings for this project primarily because of lower reduction in chilled water load than calculated by the applicant and a lower AHU-4 fan baseline power than used by the applicant. For more information regarding the deviations, see Section 3.1.

1.2 Recommendations for Program Designers & Implementers

This was a retrofit project and the baseline was the pre-retrofit conditions. Large retrofit projects generally should include operational data to calibrate the pre-retrofit energy calculations. This could be in the form of power metering, trend data, or VFD display readings such as speed or power. In this case, the applicant modeled the air handler baseline fan power using the manufacturers specified data which resulted in a higher calculated fan motor power than what the evaluator assumed to be the baseline fan power which was based on metered data during occupied periods. There was a similar discrepancy with the hot water pump measure with the applicant assuming a higher baseline pump demand than what was metered.

1.3 Customer Alert

None.

2 EVALUATED MEASURES

The following sections present the evaluation procedure, including the findings from an in-depth review of the supplied applicant calculations and the evaluation methodology determined to be the best fit for the site and the information available.

2.1 Application Information and Applicant Savings Methodology

This section describes the applicant's application information, savings methodology, and the evaluation assessment of the savings calculation algorithm used by the applicant.

An overview of the building HVAC system, which is impacted by these measures is below.

Space conditioning for a majority of the non-lab rooms is provided by four pipe fan coil units with rooms 101 and 301 being served by AHUs 1 and 2 respectively. (AHUs 1 and 2 are not impacted by any of the measures.) Ventilation air for the fan coil units and two AHUs is provided by an 100% outside air unit, AHU-4. AHU-4 includes a hot water and chilled water coil to condition the ventilation air.

AHU-4 also provides space conditioning and ventilation for the lab spaces (M1). The lab spaces include duct mounted hot water re-heat coils for space heating. There are two exhaust fans serving the lab space fume hoods.

Hot water is provided to the fan coil units and air handlers by two x 3,550 MBH gas fired boilers. There are two 20-hp (568 gpm / 75 ft) hot water circulation pumps (M2). The hot water system runs year-round to serve the lab space reheat coils.

Chilled water is provided by one x 180-ton air cooled chiller.

2.2 Applicant Description of Baseline

The applicant measure event is a retrofit. The baseline considered was the existing conditions of the equipment. The applicant description of the baseline conditions are as follows:

M1 New Controls for FCUs, AHU-4, Air Valves

The existing DDC control system was no longer functional and most of the equipment in the space operated continuously with manual adjustment of HW and CHW valves. AHU-4 operated with a constant volume control 24/7 with a fixed discharge air temperature setpoint of 53°F and the fan coil unit fans ran continuously.

M2 Hot Water Pump VFD

The existing pneumatic controlled hot water 3-way valves were not functioning; therefore, the hot water flow was constant and the hot water pumps ran continuously (8,760 hours) at its design point.

The baseline values used in the applicant savings analysis are presented in Table 2-1.

Table 2-1. Applicant baseline key parameters

		BASELINE		
Measure	**Parameter**	**Value(s)**	**Source of Parameter Value**	**Note**
M1	Summer 24/7 AHU-4 DAT setpoint	53.7°F	Existing Case	
M1	Winter 24/7 AHU-4 DAT setpoint	65°F	Applicant analysis	
M1	AHU-4 Fan Power	18.0 kW	Calculated based on design flow and pressure from drawings	
M1	AHU-4 supply airflow	16,960 CFM	Design drawings	
M1	AHU-4 Occupied Mode Schedule	24 hours per day and 7 days per week	Existing case	
M2	Hot Water Pump Motor Power	14.7 kW	Calculated based on design flow and pressure from the drawings	
M2	Hot water pump occupied mode hours	8,760	Existing case	

2.2.1 Applicant Description of Installed Equipment and Operation

The proposed system values used in the applicant savings analysis are presented in Table 2-2.

M1 New Controls for FCUs, AHU-4, Air Valves

The modeled proposed system included

- Reset supply airflow to the lab spaces served by AHU-4 by 50% during unoccupied hours.
- Reset AHU-4 discharge air temperature setpoint to 65°F during unoccupied hours in the cooling season.
- Shut off fan coil units during unoccupied hours.

The measures resulted in fan savings due to reduced airflow and fan coil time of day control and chiller energy savings due to reduced chilled water loads from all three proposed control sequences.

M2 Hot Water Pump VFD

The applicant calculations do not describe the proposed variable speed hot water pump control sequence. The modeled proposed case is a 50% reduction in the hot water flow when the outside air temperature is above 50°F.

Table 2-2: Application proposed key parameters

		BASELINE		
Measure	**Parameter**	**Value(s)**	**Source of Parameter Value**	**Note**
M1	Unoccupied Summer DAT setpoint	65°F	Proposed Case Assumption	
M1	Unoccupied Winter DAT setpoint	60°F	Proposed Case Assumption	
M1	Unoccupied AHU-4 Fan Power	5.4 kW	Calculated based on proposed airflow reduction	
M1	Unoccupied AHU-4 Airflow	10,525 CFM	50% reduction in lab airflow and no change to remaining spaces	
M1	Unoccupied Hours	4,830 Hours	Proposed Assumption	
M2	Summer Hot Water Pump Motor Power	7.3 kW	Calculated based on 50% flow reduction	
M2	Summer Hours	4,830 hours	Proposed Assumption	

2.2.2 Applicant Energy Savings Algorithm

The savings for these measures are calculated using a custom spreadsheet bin analysis.

M1 New Controls for FCUs, AHUs, Air Valves

1) AHU-4 Fan Savings

Savings are calculated based on a reduction of fan power during unoccupied periods. The applicant calculated the baseline AHU-4 supply fan motor power using the equations below based on fan design flow and pressure from design drawings. The baseline supply airflow and associated fan power are modeled as constant for each hour of the day.

Baseline Fan Power Calculation:

$$Fan\ bhp = (16{,}960\ cfm * 6\ inches) \div (70\% * 6356) = 22.9\ bhp$$

$$Motor\ kW = \left(Fan\ bhp * 0.746 \frac{kW}{HP}\right) \div 95\%\ motor\ efficiency = 18.0\ kW$$

The proposed occupied hours airflow and associated supply fan power is the same as the baseline. The proposed unoccupied hours are modeled with a 50% reduction in airflow to the lab spaces and no airflow reduction to the non-lab spaces. The baseline and proposed airflows are summarized below.

Table 2-3: Airflow Summary

Space Type	Baseline (CFM)	Proposed (CFM)	% Flow
Lab Spaces	12,870	6,435	50%
Other	4,090	4,090	100%
Total	16,960	10,525	62%

The proposed supply fan power is calculated using the equation below.

$$Proposed\ Supply\ Fan\ Motor\ kW = Baseline\ Fan\ kW * \%\ Flow^{2.5} = 5.4\ kW$$

The assumed hours when the airflow will be reduced are 8pm to 5am on weekdays and all day on weekends. Fan savings are calculated using the following formula.

$$Fan\ Savings\ kWh = (18.0\ kW - 5.4\ kW) * 4{,}830\ hours = 60{,}430\ kWh$$

2) AHU-4 Chilled Water Load Savings (Flow and DAT Reset)

Chilled water savings are expected during unoccupied hours based on a discharge air temperature reset and a chilled water coil load reduction from the reduced unoccupied lab airflow. Savings result from a reduction of chilled water ton-hours during unoccupied periods. The formula used to calculate the cooling coil tons is shown below. The discharge air temperature reset is assumed to reduce the cooling coil load by increasing the discharge air enthalpy from 21.1 Btu/lb to 29.0 Btu/lb while the airflow reduction is reduced from 100% in the baseline to 62% in the proposed case.

$$Cooling\ Coil\ Tons = 4.5 * Design\ Flow\ (CFM) * Fan\ Speed\ (\%) * (Cooling\ Coil\ Leaving\ Enthalpy\ \left(\frac{Btu}{lb}\right) - Outside\ Air\ Enthalpy\ \left(\frac{Btu}{lb}\right))$$

Savings are calculated in a temperature bin model using average outdoor air enthalpy data for 5°F bins from New Bedford Regional Airport. Proposed occupied chilled water load is modeled as the same as the constant baseline chilled water load. Chiller savings are calculated using the following formula.

$$Chilled\ Water\ Savings = (Baseline\ CHW\ Tons - Proposed\ CHW\ tons) * 4{,}830\ hours\ * 0.945\ kW/ton = 66{,}788\ kWh$$

3) Fan Coil Units

The applicant calculated the motor power and chilled water loads for the FCUs using the same methods as above. The total fan coil unit design flow is 5,125 cfm with a design pressure and efficiency of 0.75 inches and 68%. The savings for this measure assume that the FCUs will shut off during unoccupied hours (4,830 hours/year). The estimated savings for the fan coil units fan power is 4,273 kWh for fan use and a savings of 10,171 kWh for a reduction in CHW loads.

M2 Hot Water Pump VFDs

1) Unoccupied savings

The second measure assumes savings from reducing the hot water flow by 50% when the outside air temperature is above 50°F. The hot water plant is enabled year-round to be able to provide re-heat for the lab spaces.

The baseline pump motor power was calculated using the formula below at the pump design point of 568 gpm / 100 ft / 80% pump efficiency.

$$Pump\ Power\ (kW) = BHP * \frac{0.746\frac{kw}{BHP}}{91\%\ Motor\ Eff} = 14.7\ kW$$

The pump BHP was found using the following formula.

$$Pump\ BHP = \frac{Flow\ Rate\ (GPM) * Head\ Pressure\ (ft)}{3960 * Pump\ Eff\ (\%)} = 17.9\ BHP$$

Proposed case pumping energy is estimated assuming a 50% flow reduction (GPM) when the outside air temperature is above 50°F (4,905 hours annually). The proposed case hot water flow below 50°F is the same as the baseline; therefore, no savings are modeled. Pump motor demand at 50% flow is calculated using the same equations as the baseline, but with a flow rate of 284 gpm. The savings for the hot water pump VFD measure was calculated using the formula below.

$$Pump\ Savings\ kWh = (14.7\ kW - 7.3\ kW) * 4{,}905\ hours = 36{,}047\ kWh$$

The following table is a summary of the applicant savings estimates by end use category.

Table 2-4. Applicant Savings Summary

End Use	kWh	% of Total	Source
AHU-4 Fan	60,430	34%	Lab zone supply airflow setback
AHU-4 CHW	66,788	38%	Lab zone supply airflow setback and AHU DAT reset
FCU Fan	4,273	2%	Zone space temperature setback
FCU CHW	10,170	6%	Zone space temperature setback
HW Pump	36,047	20%	HW flow setback
Total	177,707	100%	-

2.2.3 Evaluation Assessment of Applicant Methodology

M1 New Controls for FCUs, AHUs, Air Valves

The applicant proposed case model does not include cooling in the 55°F-60°F temperature bin while the baseline model does, so the estimated savings are likely overestimated for that temperature bin.

M2 Hot Water Pump VFDs

Pump motor demand at 50% flow is calculated using the same equations as the baseline, but with a flow rate of 284 gpm. The pump pressure was not modified in the proposed case calculation, but pumping pressure will vary as the flow changes.

2.3 On-site Inspection and Metering

This section provides details on the tests performed during the on-site inspection. Evaluators were granted access to the site and conducted a full M&V evaluation.

2.3.1 Summary of Site Visit

This section summarizes the site visit.

- The evaluator visited the site on August 29, 2023 to install meters and meet with the facility manager. The evaluator returned to the site on April 25, 2024 to retrieve the meters.
- The evaluator installed a kW meter on the chiller associated with M1 and M2.
- The evaluator installed a kW meter on the supply fan for AHU-4 associated with M1.
- The evaluator installed an amperage logger on a sample fan coil unit associated with M1.
- The evaluator installed a kW meter on the two hot water pumps associated with M2.
- After meter retrieval the evaluator reviewed one week of trend data that was provided by the control’s contractor. The evaluator requested additional trend data, but only one week of data was provided after months of follow-up.

Table 2-5. Measure Verification

Measure Name	Verification Method	Verification Result
M1 – New Controls for FCUs, AHUs, Air Valves	kW metering and trend review	Metered data demonstrates that a flow setback schedule is implemented for the lab spaces served by AHU-4. Metered data also shows a difference in the occupied and unoccupied chilled water loads. Trend data shows AHU-4 DAT reset (in cooling mode) has not been implemented. AHU-4 provides a fixed DAT of 55°F when the outside air temperature is above 55°F.
M2 – HW Pump VFDs	kW metering and trend review	Metered data shows that hot water pump demand varies due to the variable speed drive control. The data shows a direct correlation between pumping power and outside air temperature.

2.3.2 Measured and Logged Data

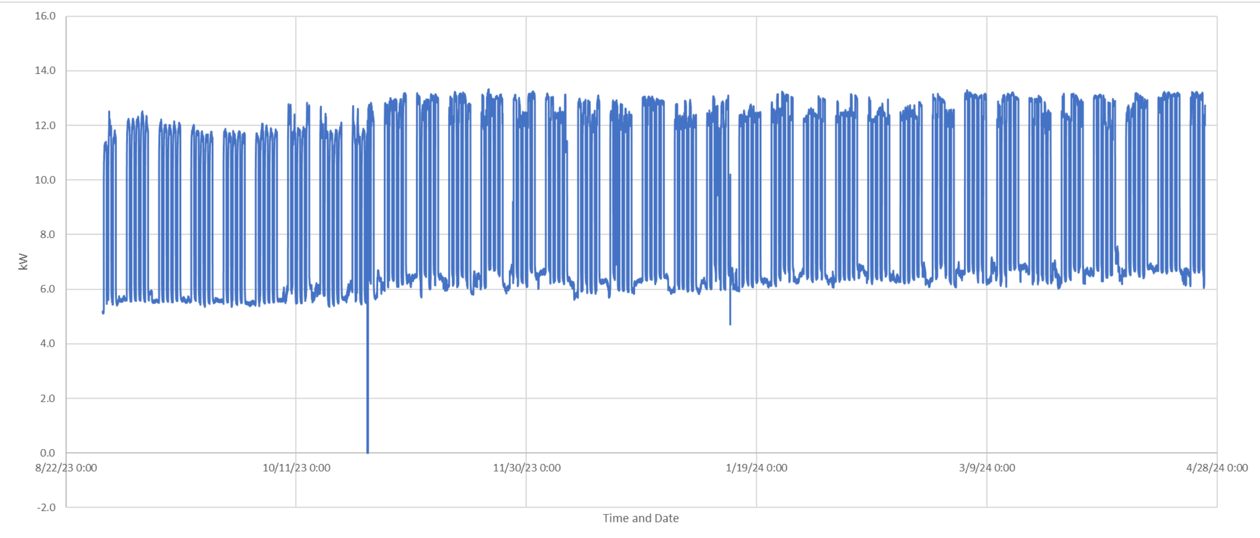
Table 2-6 summarizes the metered data collected and the period of useful data collected for each piece of equipment. The metered data from the useful data period is used in the evaluation savings analysis.

Table 2-6. Metered Data Summary

Metered Load	Data Interval	Metering Period
M1 – AHU-4 Fan Power kW, Chiller kW, FCU amperage	5-minute	8/30/23-4/25/24
M2 – HWP 1 and 2 kW	5-minute	8/30/23-4/25/24

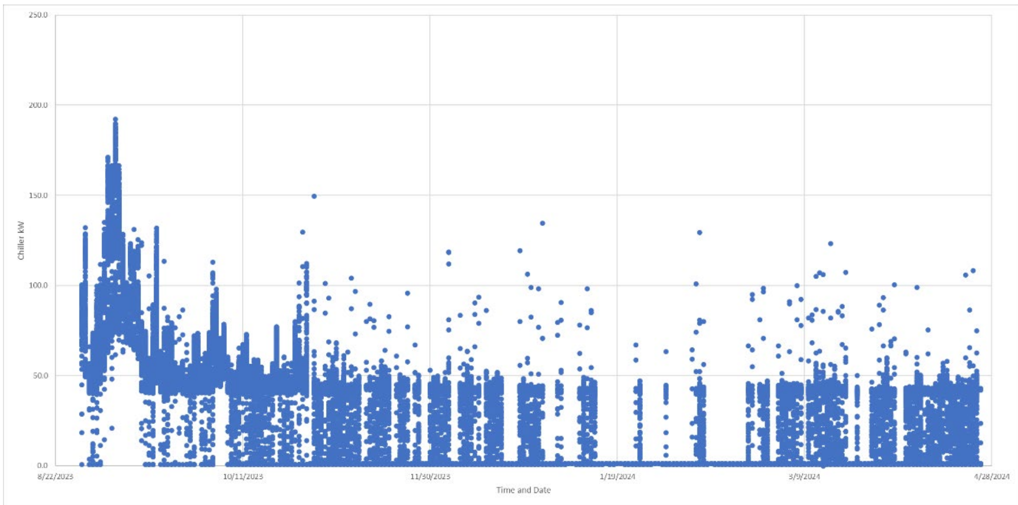
The raw kW data for AHU-4 is shown in Figure 2-1.

Figure 2-1. AHU-4 Raw kW Data



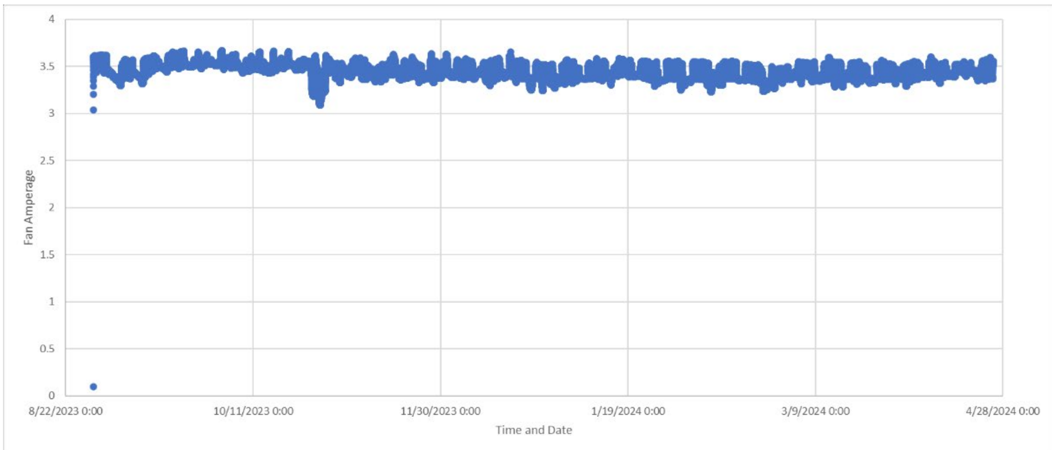
The raw kW data for the chiller is shown in Figure 2-2.

Figure 2-2. Raw Chiller kW Data



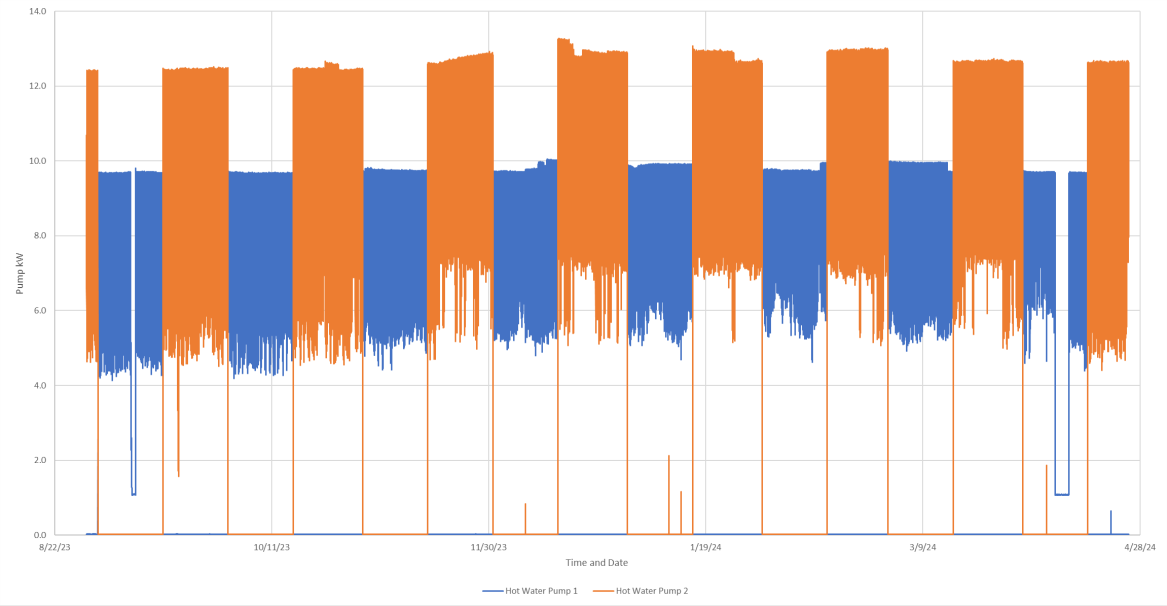
An amperage logger was placed on one sample FCU serving an office space. At the time of meter retrieval, it became apparent to the evaluator that the amperage logger was no longer properly installed. It is unclear how long the meter was disconnected and if the data was impacted. The raw amperage data is shown in Figure 2-3 and was not used in the analysis.

Figure 2-3. Raw Amp Data for Sample FCU



Both hot water pumps are served from one electric panel and one three-phase kW meter was used to collect data on both pumps simultaneously. The pumps operate on a lead/standby system with the primary pump rotated about every 15 days. The two hot water pumps are identical model numbers and run with matching sequence of operations. The evaluator metered both pumps in the same electrical panel using the same kW meter, so it is unclear why there is a discrepancy in average metered kW between the two pumps. Savings for this measure are calculated using an average of the two pumps metered data to account for the alternative operation and the different metered kW values. The raw metered kW data is shown below in Figure 2-4.

Figure 2-4. Raw Hot Water Pumps kW Data



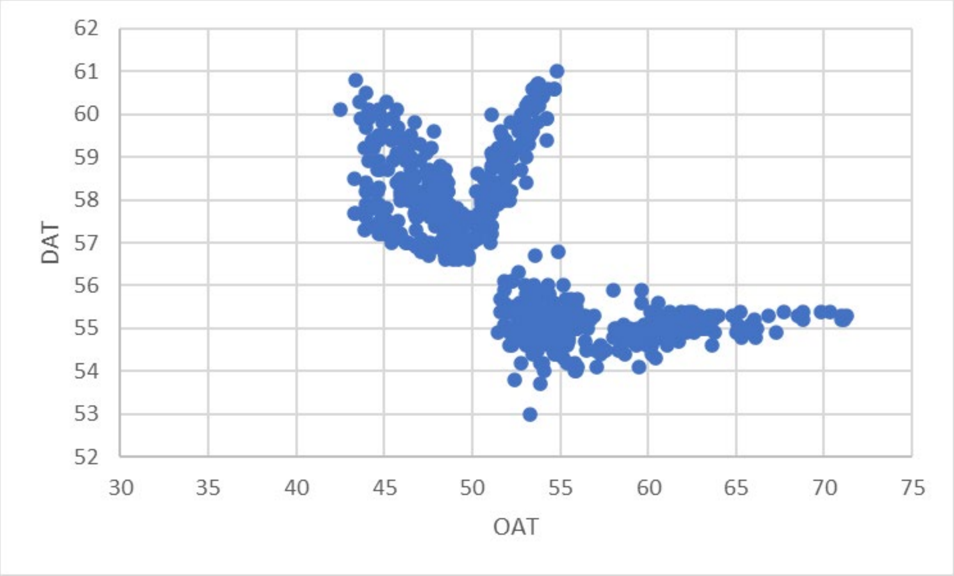
The evaluator also received one week of trend data from the controls contractor. A summary of the trends received are shown below in Table 2-7.

Table 2-7. Trend Data

Data Type	Time Period	Notes
M1 – AHU-4 discharge air temperature, 41 FCUs fan status.	5/7/24-5/14/24	
M1 – 42 FCUs fan status	2/27/22-6/1/22	Post inspection trend data
M2 – Boiler supply and return water temperatures and HWPs VFD speed.	5/7/24-5/14/24	

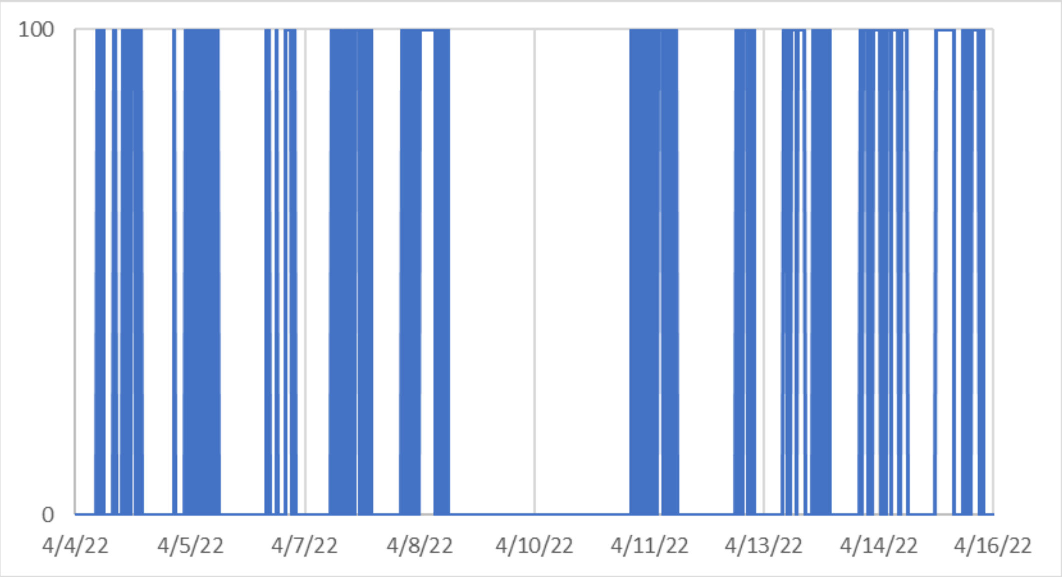
The AHU-4 discharge air temperature plot is shown in Figure 2-5 below. The AHU-4 data was used to investigate a possible DAT reset. M1 cooling savings from airflow setbacks, discharge air temperature reset and FCU unoccupied control are calculated directly from chiller metered data; therefore, the DAT trends were not directly used in the calculations. The AHU-4 DAT trends during AHU cooling mode (OAT > DAT) show the DAT fixed at 55°F indicating DAT reset was not implemented. Discharge air temperature control below 55°F outside air temperature is not clear, but it does not impact the AHU chilled water load.

Figure 2-5. AHU-4 discharge air temperature



The evaluator used three months of post inspection trend data for the FCUs to confirm that the unoccupied fan coil unit controls have been implemented. A sample of the trends showing fan status is shown in Figure 2-6.

Figure 2-6. Representative fan coil unit fan status



The boiler temperatures and VFD speed were used in the analysis of measure 2 (see Figure 2-12 for more information). The one week of data was only used to confirm the VFD sequence of operation for measure 2 as described by Figure 2-11

2.4 Evaluation Methods and Findings

This section describes the evaluator methods and findings.

2.4.1 Evaluation Description of Baseline

The evaluator measure event type is add-on and the baseline is the pre-installation equipment, which is the same as the applicant measure event type. The measure event type is add-on because the efficiency of the existing system was improved but not replaced. The evaluator generally agrees with the applicant baselines.

M1 New Controls for FCUs, AHUs, Air Valves

The evaluator agrees that the baseline for AHU-4 fan control should be a constant operation year-round matching the current occupied period operation.

M2 Hot Water Pump VFDs

The evaluator observed the installation of VFDs on the two hot water pumps. The evaluator agrees with the baseline operation of a constant flow due to the lack of VFDs and the failed pneumatic controls.

2.4.2 Evaluation Calculation Method

M1 New Controls for FCUs, AHUs, Air Valves

AHU-4 Fan Savings

The evaluator collected metered kW data for AHU-4 supply fan. The hourly average kW data was calculated to analyse the time-of-day dependency. The results are shown in Table 2-8 below.

Table 2-8. AHU-4 Metered kW vs Time of Day

Time of Day (Hour)	Monday	Tuesday	Wednesday	Thursday	Friday	Saturday	Sunday
0	6.19	6.21	6.21	6.18	6.25	6.23	6.22
1	6.19	6.19	6.19	6.17	6.24	6.21	6.22
2	6.21	6.20	6.19	6.15	6.22	6.20	6.19
3	6.18	6.20	6.19	6.16	6.19	6.20	6.22
4	6.18	6.19	6.18	6.15	6.18	6.21	6.22
5	8.38	8.38	8.37	8.35	8.35	6.21	6.22
6	12.27	12.19	12.20	12.19	12.21	6.21	6.22
7	12.40	12.25	12.34	12.28	12.31	6.24	6.22
8	12.45	12.34	12.38	12.32	12.36	6.25	6.21
9	12.41	12.43	12.43	12.39	12.45	6.25	6.21
10	12.44	12.48	12.44	12.42	12.49	6.24	6.24
11	12.50	12.45	12.48	12.21	12.48	6.24	6.24
12	12.54	12.47	12.46	12.22	12.47	6.25	6.24
13	12.56	12.51	12.47	12.16	12.48	6.23	6.26
14	12.55	12.51	12.47	12.16	12.48	6.24	6.27
15	12.54	12.49	12.48	12.49	12.49	6.25	6.25
16	12.52	12.49	12.46	12.49	12.47	6.27	6.25
17	12.47	12.49	12.45	12.48	12.47	6.27	6.23
18	12.46	12.50	12.44	12.48	12.48	6.26	6.23
19	12.47	12.48	12.41	12.46	12.47	6.24	6.23
20	10.29	10.29	10.18	10.30	10.31	6.23	6.23
21	6.40	6.41	6.35	6.39	6.43	6.23	6.22
22	6.27	6.31	6.26	6.29	6.32	6.21	6.20
23	6.22	6.25	6.22	6.26	6.26	6.21	6.19

The evaluator is able to verify the time-of-day airflow reset schedule has been implemented and that the supply fan kW is turning down during unoccupied periods. The evaluation analysis assumes that the baseline operation was consistent with the current occupied hours operation with an average fan power of 12.42 kW year-round.

The observed hours when the airflow is reduced are 8pm to 6am on weekdays (1 more hour per day than modeled by the applicant) and all day on weekends. The proposed fan power during unoccupied hours is the average fan power during these unoccupied hours.

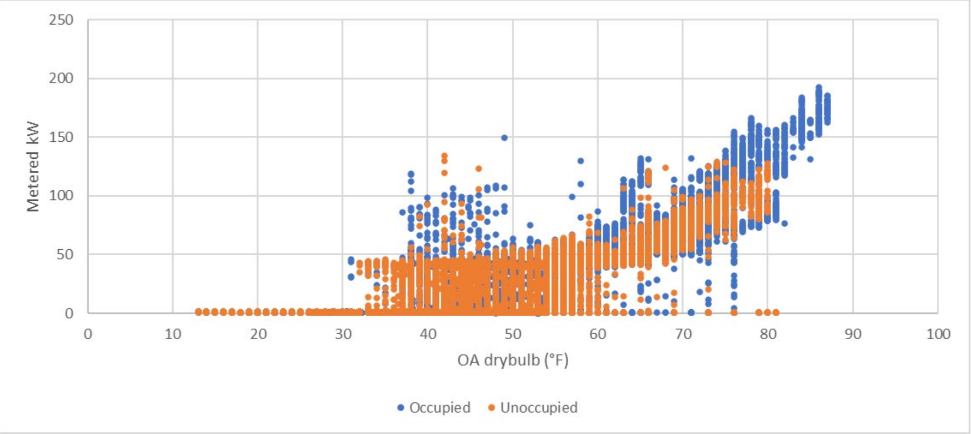
Fan savings are calculated using the following formula.

$$Fan\ Savings\ kWh = (12.4\ kW - 6.5\ kW) * 5{,}120\ hours = 30{,}075\ kWh$$

Chilled Water Reduction

To estimate chiller savings due to the reduction in AHU-4 and fan coil unit unoccupied chilled water loads, the evaluator created a plot of the metered chiller kW data against outside air temperature for occupied and unoccupied periods that correlate with the schedule shown in Table 2-8 above. The results of this correlation are shown in Figure 2-7 below.

Figure 2-7. Chiller kW vs Outside Air Temperature



The metered data was also plotted against time of day to evaluate the correlation between chiller use and time of day. The results are shown in Figure 2-8 below.

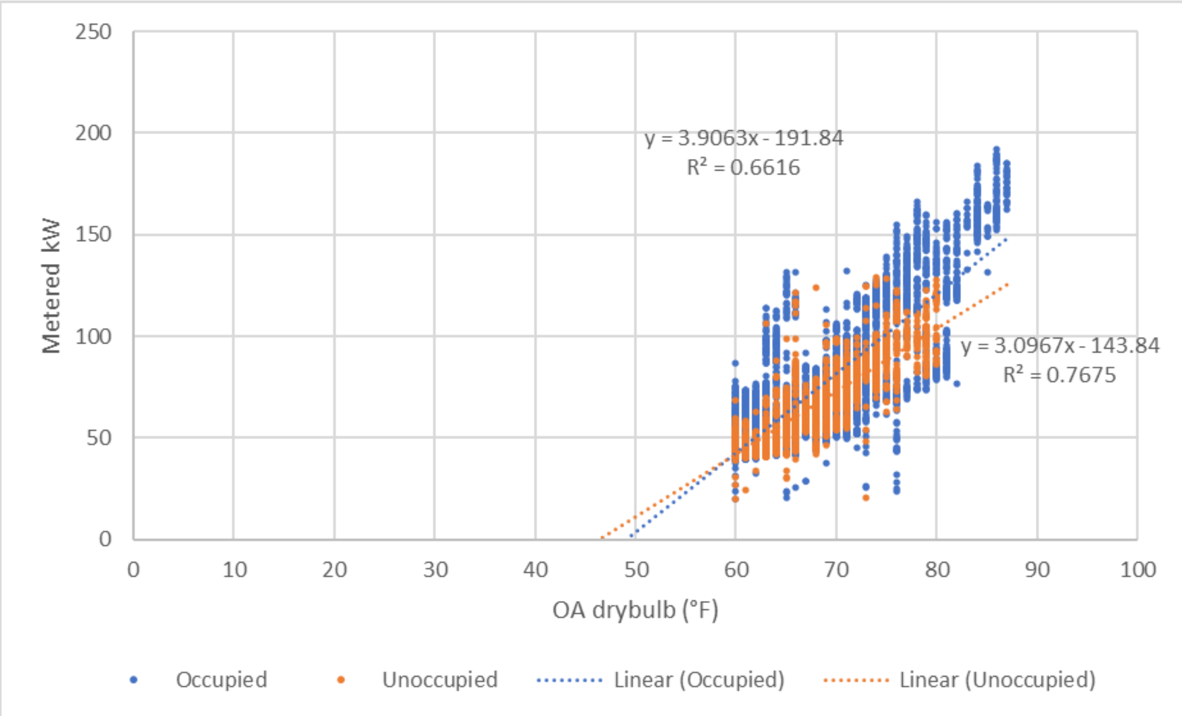
Figure 2-8. Chiller kW vs Time of Day

Time of Day (Hour)	Monday	Tuesday	Wednesday	Thursday	Friday	Saturday	Sunday
0	14.29	11.85	14.42	14.63	12.28	15.45	13.00
1	13.34	11.92	14.62	14.65	13.21	16.14	13.47
2	13.50	11.15	15.08	13.36	12.87	15.71	12.99
3	14.07	11.81	15.09	13.39	12.33	16.26	12.26
4	13.95	10.32	14.47	12.96	12.26	15.77	12.25
5	15.26	11.30	16.13	14.10	13.34	15.57	10.95
6	17.66	13.72	17.21	16.10	15.19	15.97	11.63
7	19.50	16.50	19.69	17.20	19.21	16.43	13.68
8	20.30	18.08	24.13	19.72	22.32	20.12	15.75
9	24.55	22.94	27.53	23.12	24.49	22.41	19.22
10	25.66	27.47	28.06	26.46	25.40	23.52	18.48
11	28.04	26.27	29.35	25.23	27.31	23.87	21.96
12	27.15	28.26	30.50	27.79	28.70	23.92	21.61
13	26.54	28.02	29.36	26.32	29.19	23.06	22.39
14	26.59	27.73	27.24	25.96	28.35	20.27	21.95
15	25.86	27.11	26.71	24.34	25.99	18.83	20.62
16	25.44	27.21	26.39	22.71	25.87	17.29	18.19
17	24.19	24.55	24.87	20.97	24.65	17.66	18.29
18	22.11	23.43	23.70	18.21	23.35	15.94	16.33
19	20.90	22.26	21.36	17.84	22.32	14.98	15.61
20	16.35	19.03	18.67	16.18	19.96	14.96	15.03
21	14.21	16.25	16.64	14.74	16.16	14.81	14.02
22	13.20	16.04	15.90	13.90	15.95	13.70	13.92
23	12.62	15.46	15.58	13.26	16.73	14.20	12.78

Savings are estimated using a bin analysis with the average metered kW taken in 5°F temperature bins for both the occupied and unoccupied periods.

The temperature data used to evaluate the savings was TMY3 data from Providence, RI to normalize the data to a typical year. The baseline energy use is assumed to match the current occupied operation for the entire year because the load reduction is due to AHU-4 unoccupied airflow setback and fan coil unit unoccupied space temperature setback, which matches the applicant baseline assumption. Linear trend lines were used to estimate the chiller kW during high temperature bins that data was not collected at during the metered period. The trend lines were created using the metered data when the outside air temperature was above 60°F to eliminate periods when the chiller was cycling between on and off. The trend lines are shown in Figure 2-9 below.

Figure 2-9. Chiller kW with Trendlines



The reduction in chiller kW during unoccupied periods is a result of AHU-4 flow reduction and FCU space temperature setback. The bin-analysis spreadsheet used to calculate the savings for this measure is shown in Table 2-9 below. The evaluated estimated savings for the chilled water load reduction are 19,219 kWh.

Table 2-9. Chiller Bin Analysis

OA Range		Unoccupied	Unoccupied	Chiller			
				Base	Prop	Saved	
Min	Max	avg db	Hours	kW	kW	kW	kWh
90	95	91.6	3	166.1	139.9	26	79
85	90	87.8	12	151.0	128.0	23	277
80	85	81.7	84	127.4	109.2	18	1,526
75	80	76.7	198	107.6	93.6	14	2,785
70	75	72.3	277	90.7	80.1	11	2,924
65	70	68.0	478	73.9	66.8	7	3,382
60	65	62.5	532	52.4	49.7	3	1,389
55	60	57.0	346	42.6	34.2	8	2,929
50	55	52.4	355	23.9	19.8	4	1,476
45	50	48.0	462	14.9	11.4	3	1,610
40	45	43.1	467	8.6	7.3	1	646
35	40	37.5	529	2.3	1.7	1	304
30	35	32.1	465	1.2	1.3	0	-29
25	30	27.7	215	1.1	1.1	0	-2
20	25	23.1	234	1.1	1.1	0	2
15	20	18.0	121	1.1	1.2	0	-1
10	15	11.7	65		1.2	-1	-76
5	10	8.4	16			0	0
0	5	3.9	1			0	0
Total			4,860	159,179	139,959	119	19,219

Fan Coil Units

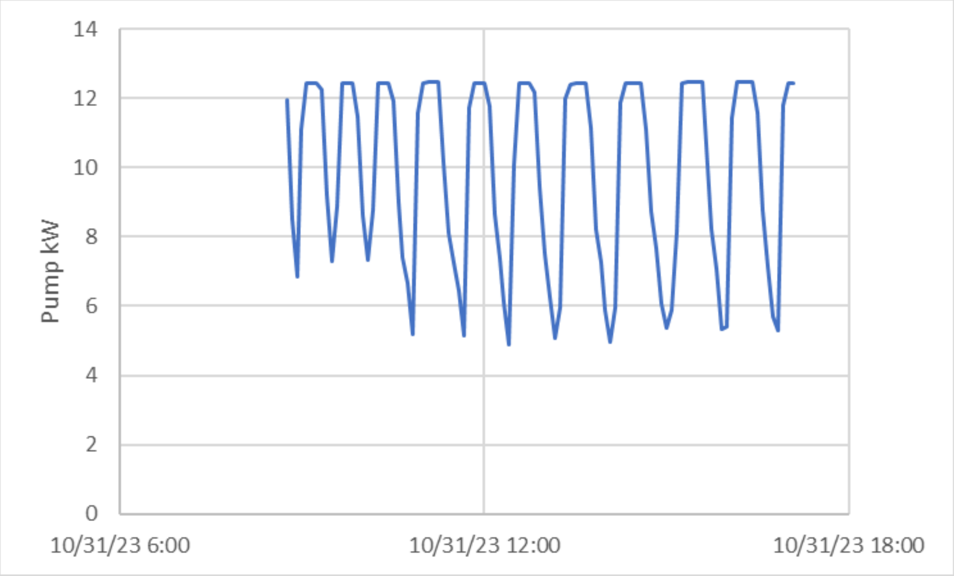
The savings due to the reduced chilled water load on the FCUs was estimated during the review of the metered chiller data. This section considers the savings associated with reduced fan energy during unoccupied periods in the FCUs. Due to the uncertainty and limited quantity of metered FCU fan power data, savings for this measure are estimated using the one week of trend data supplied during May of 2024 as well as three months of post installation trend data for 39 out of the 42 FCUs from March 2022 to April 2022. During the three months of post-installation trend data, it was observed that of the 39 FCUs observed, 25 were cycling off during unoccupied periods, 9 FCUs fans were always off, and 5 FCU fans were always on. Based on this, fan savings are estimated using the assumption that 64.1% of FCU's cycle off during unoccupied hours. The assumed total fan power of the FCUs was based on the applicant estimate due to a lack of documentation for further estimates. The FCU fan savings were estimates using the formula below, where the total savings are equal to 2,756 kWh.

$$FCU\ fan\ Savings\ kWh = (0.9\ kW\ Total\ Fan\ Power) * (8{,}760\ hours - 4{,}860\ unoccupied\ hours) * 64.1\% = 2{,}756\ kWh$$

M2 Hot Water Pump VFDs

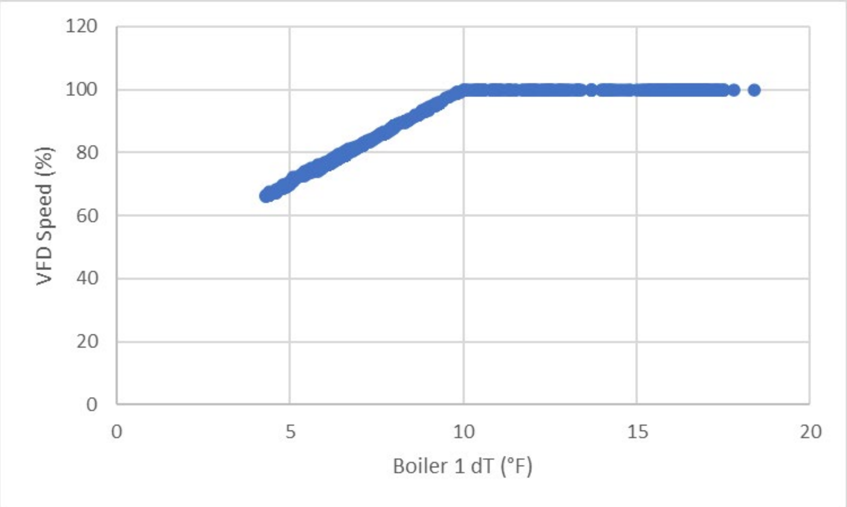
Metered kW data was recorded for both hot water pumps. The pumps run on a lead / lag staging with the lead pump alternating every 15 days. As shown in Figure 2-10 the hot water pumps constantly cycled between 100% speed and part speed operation.

Figure 2-10. HWP VFD Speed Modulation



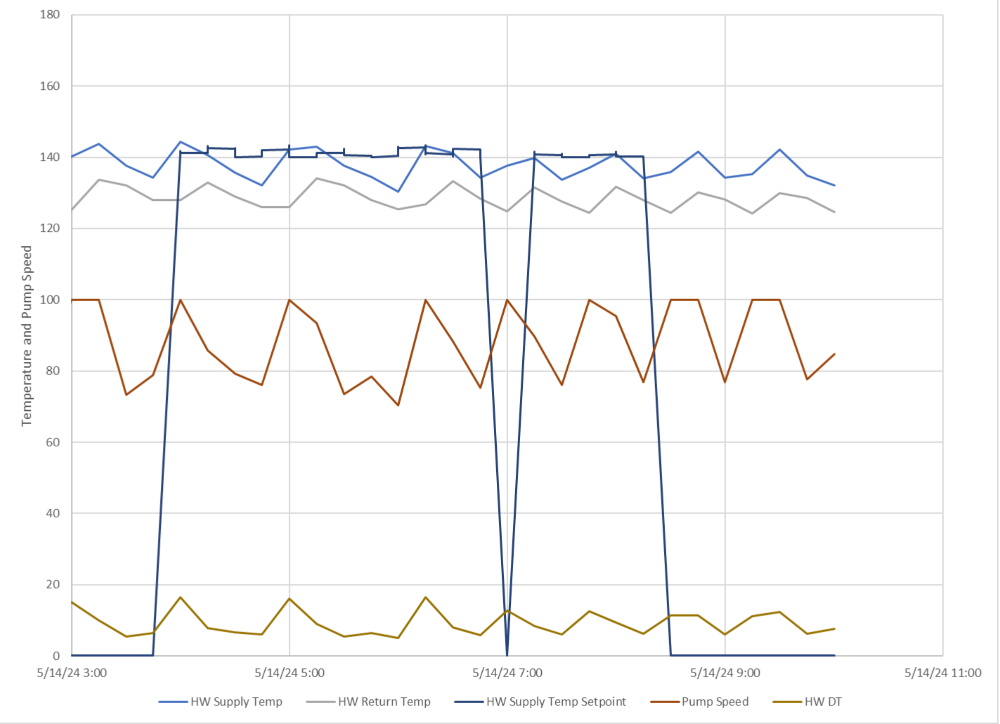
The operation was further investigated using the one week of trend data supplied by the site's controls contractor. The controls contractor stated that there are three-way control valves; therefore, the hot water pump speed is controlled to maintain a hot water loop temperature differential of 10°F.Figure 2-11, confirms that the hot water operation is controlled to a 10°F dT by plotting the active hot water pump's VFD speed to the boiler's difference in supply and return temperature. When the loop differential temperature is above 10°F indicating higher hot water loads the pump speed ramps up to 100% and the when the differential temperature is below 10F (lower loads) the pump speed modulates.

Figure 2-11. Boiler dT vs HWP VFD Speed



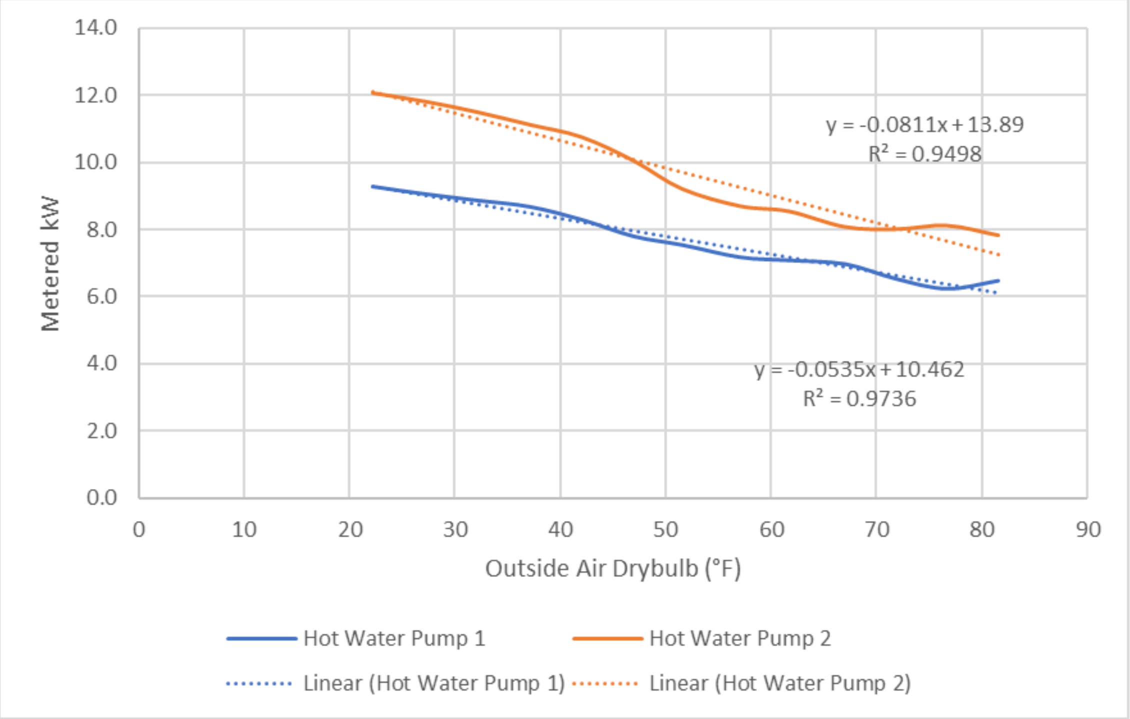
Boiler trend data indicates that the hot water loop differential temperature and associated pump speed are dictated by the boiler cycling. The boiler appears to cycle on and off to maintain the hot water supply temperature setpoint. Figure 2-12 plots the boiler dT and pump VFD for 7 hours of operation. Figure 2-12 confirms that the hot water pumps cycle between 100% VFD speed and slower operation to satisfy the boiler dT. When the boiler cycles on, the hot water supply increases leading to an increase in the loop DT and an associated increase in pump speed. After the boiler cycles off the hot water supply temperature drops leading to a decrease in the loop DT and associated decrease in pump seed.

Figure 2-12. Hot Water System Operation



The metered hot water pump data is plotted against outside air temperature in Figure 2-13 below. Savings for this measure are calculated using an average of the two pumps metered data to account for the alternative operation and the different metered kW values.

Figure 2-13. Hot Water Pumps kW vs Outside Air Drybulb



Savings for this measure are estimates using a bin model with 5°F temperature bins. The active pump runs for 8,760 hours and pump speed varies during all operating hours. Baseline operation is estimated to match the averaged metered 100% VFD operation, with the reduction of a VFD burden, for 8,760 hours. The 100% kW for each pump is determined by averaging the metered kW during periods when the kW readings were within the maximum 5% of their range. The VFD burden was estimated to be 0.513 kW based on a 96.8% efficiency at full load. The formula used to calculate VFD burden is shown below.

$$VFD\ Burden = 20\ hp * 3.2\%\ efficiency\ loss * 0.746 * \left(\frac{1}{93\%\ Motor\ Efficiency}\right) = 0.513\ kW$$

Savings were calculated using the following bin models, there is one model for the occupied building hours and one model for the unoccupied building model.

Table 2-10. HWP Bin Model

				Maximum kW					
				9.7	12.7	10.7			
				Hot Water Pumps					
OA Range		Occupied	Occupied	kW			kWh		
Min	Max	avg db	Hours	HWP1	HWP2	Average	Installed	Baseline	Savings
90	95	92.1	41	5.5	6.4	6.0	245	438	193
85	90	87.8	64	6.9	6.8	6.8	437	684	248
80	85	82.2	214	6.5	7.8	7.1	1,529	2,289	759
75	80	76.9	277	6.2	8.1	7.2	1,987	2,962	975
70	75	72.6	279	6.5	8.0	7.3	2,028	2,984	956
65	70	68.0	335	7.0	8.1	7.5	2,519	3,583	1,063
60	65	62.3	372	7.1	8.5	7.8	2,905	3,978	1,073
55	60	56.9	301	7.2	8.7	7.9	2,388	3,219	831
50	55	52.6	249	7.5	9.2	8.4	2,084	2,663	579
45	50	47.8	308	7.8	10.0	8.9	2,746	3,294	548
40	45	42.9	273	8.3	10.8	9.5	2,599	2,920	321
35	40	37.4	392	8.7	11.1	9.9	3,880	4,192	312
30	35	32.2	340	8.9	11.5	10.2	3,462	3,636	175
25	30	27.8	172	9.0	11.8	10.4	1,794	1,839	45
20	25	23.1	141	9.3	12.1	10.7	1,504	1,508	4
15	20	18.5	103	9.5	12.3	10.9	1,121	1,102	-20
10	15	11.8	31	9.7	12.5	11.1	344	332	-12
5	10	7.7	8	9.7	12.7	11.2	90	86	-4
0	5	-2.5	0	9.7	12.7	11.2	0	0	0
Total			3,900			8.6	33,662	41,709	8,047

				Maximum kW					
				9.7	12.7	10.7			
				Hot Water Pumps					
OA Range		Unoccupied	Unoccupied	kW			kWh		
Min	Max	avg db	Hours	HWP1	HWP2	average	Installed	Baseline	Savings
90	95	91.6	3	5.6	6.5	6.0	18	32	14
85	90	87.8	12	6.9	6.8	6.8	82	128	46
80	85	81.7	84	6.5	7.8	7.1	600	898	298
75	80	76.7	198	6.2	8.1	7.2	1,420	2,118	697
70	75	72.3	277	6.5	8.0	7.3	2,013	2,962	949
65	70	68.0	478	7.0	8.1	7.5	3,595	5,112	1,517
60	65	62.5	532	7.1	8.5	7.8	4,154	5,690	1,535
55	60	57.0	346	7.2	8.7	7.9	2,745	3,700	956
50	55	52.4	355	7.5	9.2	8.4	2,971	3,797	826
45	50	48.0	462	7.8	10.0	8.9	4,120	4,941	821
40	45	43.1	467	8.3	10.8	9.5	4,445	4,994	549
35	40	37.5	529	8.7	11.1	9.9	5,237	5,657	421
30	35	32.1	465	8.9	11.5	10.2	4,734	4,973	239
25	30	27.7	215	9.0	11.8	10.4	2,243	2,299	57
20	25	23.1	234	9.3	12.1	10.7	2,497	2,503	6
15	20	18.0	121	9.5	12.3	10.9	1,319	1,294	-25
10	15	11.7	65	9.7	12.5	11.1	721	695	-26
5	10	8.4	16	9.7	12.7	11.2	179	171	-8
0	5	3.9	1	9.7	12.7	11.2	11	11	-1
Total			4,860			8.9	43,103	51,976	8,873

Savings for this measure are equal to 16,920 kWh.

The following table is a summary of evaluator savings by end use category.

Table 2-11. Evaluator End Use Savings Summary

End Use	kWh	% of Total	Source
AHU-4	30,075	44%	Lab zone supply airflow setback
Chiller	19,219	28%	Chilled water load reduction
FCUs	2,756	4%	Zone space temperature setback
HW Pump	16,920	25%	HW flow setback
Total	68,970	100%	-

Peak Savings Discussion

The bin model was used to calculate summer and winter kW reductions during the demand-peak periods, as well as % on peak savings. Peak demand hours were calculated for each temperature bin in both the occupied and unoccupied bin models. The following peak period definitions were used:

> *Summer Demand Peak: Non-holiday weekdays, 1-5PM, June, July, August*
>
> *Winter Demand Peak: Non-holiday weekdays, 5-7PM, December, January*
>
> *Energy Peak Period: Non-holiday weekdays, 7AM-11PM, year-round*

M1 does not result in a peak demand reduction because all of the savings occur overnight on weekdays (9pm – 6am) and on weekends. Similarly, the M1 percent on peak energy savings are low because there are only savings during the 9pm to 11pm peak hours.

M2 provides small winter demand reduction because the hot water pump runs close to full speed at cold outside air temperatures. M2 summer demand reduction is higher because the hot water load and flow is lower at warmer outside air temperatures. The hot water pump runs continuously therefore there are energy savings throughout the day resulting in 50% of the energy savings occurring during peak hours.

3 FINAL RESULTS

This section summarizes the evaluation results determined in the analysis above. This section includes a summary table of savings by major end-use and application.

Table 3-1. Summary of Key Parameters

		BASELINE		PROPOSED / INSTALLED	
Measure	Parameter	Tracking Value(s)	Evaluation Value(s)	Tracking Value(s)	Evaluation Value(s)
M1	AHU-4 Supply Fan kW	18.0	12.4	5.4	6.6
M1	AHU-4 Supply Fan Hours at reduced flow (unoccupied hours)	4,630	5,096	4,630	5,096
M1	Maximum Unoccupied period Chiller Load kW	137.7	166.1	108.9	139.9
M1	FCU Fan Savings Factor	N/A	N/A	100%	64.1%
M2	HWP kW when savings	14.7	10.7	7.3	8.5
M2	HWP hours when savings	4,905	8,760	4,905	8,760

3.1 Explanation of Differences

This section describes the key drivers behind any difference in the application and evaluation estimates, annual kWh savings. The following table summarizes these differences. The purpose of this table is to describe how changes to the key parameters influenced the final project savings through the end-use summary analysis. Table 3-2 provides a summary of the differences between tracking and evaluated values.

Table 3-2. Summary of Deviations

Measure	Discrepancy	Parameter	Impact of Deviation	Discussion of Deviations
M1	Operating Load	Chiller energy savings	-32.5%	**Decreased savings –** The reduction in chiller load is less than the applicant assumed in the original savings analysis which decreases savings. The lower chilled water load savings are at least partially because AHU-4 DAT reset was not implemented. It is assumed the lower chilled water load savings is also due to a lower AHU-4 airflow reduction during unoccupied hours, but the evaluator was not able to obtain airflow trends to confirm the installed airflow reduction.
M1	Operating Load	Baseline and Proposed kW for AHU-4	-17.1%	**Decreased savings –** The evaluated baseline fan power is lower than the applicant and the evaluated proposed fan power is higher than the applicant, which lead to a decrease in savings.
M1	Measure quantity	Percentage of FCUs cycling fans during unoccupied hours	-0.9%	**Decreased savings –** The quantity of FCUs that are operational during the day and then set back at night is less than the applicant assumed.
M2	Operating Load	HWP Base kW	-11.0%	**Decreased savings –** The pump baseline power with 100% speed is less than the applicant assumption which decreases savings.
M2	Operating Load	HWP Prop kW	-3.9%	**Decreased savings –** The pump proposed power is higher than the applicant assumption which decreases savings.
M2	Hours of operation	HWP Hours	4.2%	**Increased savings –** The pump operating hours at reduced speed are more than the applicant assumed which increases savings.
Final RR				**-61.2%**

3.2 Lifetime Savings

The evaluators calculated applicant and evaluated lifetime savings values using the following formula:

$$\text{Lifetime Savings kWh} = \text{Annual Savings kWh} * \text{Measure Lifetime Years}$$

The evaluated lifetime savings are smaller than the tracking lifetime savings because the evaluated first-year savings are smaller than the tracking first-year savings. Table 3-3 provides a summary of key factors that influence lifetime savings. The evaluator assumes that the tracking lifetime savings match the lifetime savings from the BCR.

Table 3-3. Application 11655059/13321748 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	1,885,221	1,957,315	774,299
First year savings	177,708	177,708	68,970
Measure lifetime*	10.61 years	11.01 years	11.23 years
Baseline classification	Retrofit	Retrofit	Retrofit

**Weighted average of all measure lifetimes included in application*

Table 3-4 provides a summary of key factors that influence lifetime savings for M1.

Table 3-4. Measure M1 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	1,416,610 kWh	1,416,610 kWh	519,839 kWh
First year savings	141,661 kWh	141,661 kWh	52,050 kWh
Measure lifetime	10 years	10 years (project BCR)	10 years (TRM)
Baseline classification	Retrofit	Retrofit	Retrofit

Table 3-5 provides a summary of key factors that influence lifetime savings for M2. The evaluator assumes that the tracking lifetime savings match the lifetime savings from the BCR.

Table 3-5. Measure M2 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	468,611 kWh	540,705 kWh	253,796 kWh
First year savings	36,047 kWh	36,047 kWh	16,920 kWh
Measure lifetime	13 years	15 years (project BCR)	15 years (TRM)
Baseline classification	Retrofit	Retrofit	Retrofit

Ancillary impacts

Measure 1 includes gas savings associated with the reduction in hot water load on the boiler due to the AHU-4 and FCU fan sequencing. There are also gas savings associated with a reduction in the hot water load on the boiler due to the reset of zone space temperatures during unoccupied hours. The applicant reported an estimated 10,470 therms saved for measure 1. The evaluator used the following equations to estimate the gas impact of measure 1:

$$Gas\ Savings = 1.08 * Total\ CFM * (DAT - OA\ Drybulb) * Hours * Boiler\ eff * \left(\frac{1}{100{,}000}\right) = 7{,}916\ therms$$

Where,

Total CFM = AHU-4 CFM combined with total FCU CFM (20,245 CFM)

DAT = AHU-4 and FCU Discharge Air Temp during Unoccupied periods (58.7°F)

OA Drybulb = Average outside air drybulb during unoccupied periods during heating season (less than 80°F) (49.3°F)

Hours = Total Unoccupied Hours (4,860 hours)

Boiler eff = Boiler efficiency from manufacturer specifications (79%)

The evaluator estimated an impact of 7,916 therms (75.6% realization rate) due to the reduction of hot water load on the boilers in M1. The total CFM is based on the applicant assumption for AHU-4 CFM and the evaluator's assumption of FCU CFM reduction. The evaluator was unable to confirm the applicant CFM estimates. There are no ancillary impacts associated with M2.

RI CUSTOM ELECTRIC EVALUATION SITE-SPECIFIC REPORT

DNV SITE ID: RICE22S011

Report Date: 7/3/2024

Application ID(s)	13839869	
Project Type	C&I Retrofit	
Program Year	2022	
Evaluation Firm	DMI	
Evaluation Engineer	Bennett Rose	
Senior Engineer	Mickey Bush	

1 EVALUATED SITE SUMMARY AND RESULTS

This site is a university. The site has an MOU with RI Energy under which the site completes a large number of energy efficiency projects annually and submits them under one application. The 2022 application includes 26 non-lighting energy conservation measures, all of which are retrocommissioning with the exception of three (3) insulation measures, and one chiller replacement. The evaluator is tasked with evaluating a sample of six (6) of the retrocommissioning measures. The sample was selected by DNV and selected to minimize error in the evaluation results. The tracking savings associated with the six (6) measures included in the evaluation sample represent 48% of the tracking savings for the site. All of these measures were completed in separate buildings with the exception of measures M3 and M6 which were completed in the same building. The evaluated energy savings measures are as follows:

M1 Replace FCU Thermostats

Replace existing manual controls with new digital thermostats with BMS communication to control the 36 hydronic FCUs serving an administrative building. Savings are achieved by implementing time of day space temperature setbacks.

M2 Chiller Occupancy Controls

The 10-ton air-cooled chiller serving a house that is used for an academic department is running continuously even at low OAT and the building is usually in heating mode. The controls contractor changed the programming so that the chiller is turned off when either occupied or unoccupied space temperature setpoints are achieved. The dual-temp pumps will also turn off if cooling/heating is not needed.

M3 VAV Controller Replacements

In September 2021, facility staff observed issues with AHU-1 in a Lab/office/classroom building. The unit was maintaining 55F discharge air temperature (DAT) whenever the unit was running since Dec 2020 leading to overcooling of the spaces.

Facility staff found that the DAT reset is based on return air temperature and some VAVs terminal units were giving incorrect return air temperature values. A work order was submitted and it was found that many VAV terminal unit controllers failed and needed replacement. (Out of the total 27 VAVs, 13 were replaced.) Also, construction filters were found on return air ducts which were creating an unnecessary pressure drop in the return duct system.

M4 Chiller OAT Controls

It was found that the chiller at one of the libraries on campus was kept ON during colder weather when it was not required to run to meet the space loads. Savings are achieved by disabling the chiller when OAT is less than a newly programmed chiller enable setpoint.

M5 Dining Area AHU Controls

Retrocommissioning identified that the ventilation systems operate 24/7. Savings are achieved by implementing time of day scheduling controls and reducing operating hours for the AHUs and exhaust fans.

M6 Exhaust Fan Staging

This measure involves retrocommissioning of laboratory building high plume exhaust fans with an outside air bypass. The design intent is to open the outside air bypass damper when the building exhaust is low to maintain a high exhaust exit velocity to prevent re-entrainment of the exhaust air. There are four 30-hp constant speed exhaust fans manifolded together to serve the building exhaust.

Prior to the winter of 2020 the site was operating two fans to meet the building exhaust airflow requirements and maintain the exhaust static pressure setpoint. In the winter of 2020, the outside air bypass damper and the isolation dampers on two of the fans failed leading to three fans being required to run to meet the exhaust static pressure setpoint (baseline condition). The site initially closed the outside air bypass damper and switched to only running the two fans with the failed isolation dampers (proposed condition). After these changes the site was able to run two fans to serve the exhaust loads. Later in 2021 the site repaired all of the dampers to allow the system to return to fully automatic control. The site continued to be able serve the exhaust loads with two fans after the repairs were completed.

Energy savings result from running two fans in the proposed case compared to three fans in the baseline.

These six measures represent a sample of measures installed under this application and represent about 51% of the savings. These six measures were evaluated and the result from this sample (realization rate) was applied to the total tracking savings for the application to calculate the evaluated savings for the application. The evaluation results are presented in Table 1-1.

Table 1-1. Evaluation Results Summary

PA Application ID	Measure Name		Annual Electric Energy (kWh)	% of Energy Savings On-Peak	Summer On-Peak Demand (kW)	Winter On-Peak Demand (kW)
13839869	M1 Replace FCU Thermostats	Tracked	1,492	40.00%	0.34	0.34
		Evaluated	1,609	0.00%	0.00	0.00
		Realization Rate	107.82%	0.00%	0.00%	0.00%
	M2 Chiller Occupancy Controls	Tracked	16,056	40.00%	3.67	3.67
		Evaluated	5,778	94.54%	1.05	0
		Realization Rate	35.99%	236.36%	28.68%	0.00%
	M3 VAV Controller Replacements	Tracked	12,907	40.00%	2.86	2.86
		Evaluated	0	0.00%	0	0
		Realization Rate	0.00%	0.00%	0.00%	0.00%
	M4 Chiller OAT Controls	Tracked	43,329	40.00%	9.92	9.92
		Evaluated	12,830	47.12%	0	5.61
		Realization Rate	29.61%	117.81%	0.00%	56.58%
	M5 Dining Area AHU Controls	Tracked	95,907	40.00%	21.95	21.95
		Evaluated	79,720	30.29%	1.40	3.45
		Realization Rate	83.12%	75.72%	6.37%	15.73%
	M6 Exhaust Fan Staging	Tracked	126,623	40.00%	28.98	28.98
		Evaluated	214,015	46.00%	24.43	24.43
		Realization Rate	169.02%	115.00%	84.30%	84.30%
	Evaluation Sample Total	**Tracked**	**296,314**	**40.00%**	**67.72**	**67.72**
		Evaluated	**313,951**	**42.71%**	**26.88**	**33.50**
		Realization Rate	**105.95%**	**106.79%**	**39.70%**	**49.46%**
	Application Total	**Tracked**	**586,045**	**67.00%**	**190.62**	**49.84**
		Evaluated	**620,928**	**71.55%**	**75.67**	**24.65**
		Realization Rate	**105.95%**	**106.79%**	**39.70%**	**49.46%**

N/A = Not applicable

1.1 Explanation of Deviations from Tracking

The evaluated savings are 6.0% more than the applicant-reported savings.

The reason that evaluated savings for this site are greater than the tracking savings is due to additional information provided to the evaluator by the site for M6. The evaluator revised the time period to be considered as reflective of existing case baseline for the measure and this increased the savings for M6. This increase in savings was greater than the cumulative decrease in savings associated with the other measures and resulted in a realization rate greater that 100% for this site.

The second largest discrepancy was a decrease in savings for M4. The evaluator calculates energy savings using whole building demand trend data that indicates the applicant overestimated the chiller load at low ambient temperatures. The applicant estimated the chiller would be 25% loaded in the winter, but the whole building demand data indicates a lower average chiller demand.

M5 evaluated savings are lower than the tracking due to lower cooling and reheat loads. One reason for the lower loads is that the observed AHU discharge temperature is higher than was used in the savings calculations (63°F vs 55°F).

M2 evaluated savings are lower than the tracking savings due to an over estimation of chiller run hours (applicant assumes chiller runs continuously but chiller cycles to meet the load) in the baseline and the evaluation finding that the implemented controls result in a pump penalty.

M3 evaluated savings are lower than the tracking because there is no basis or evidence of savings attributable to the project.

The applicant claimed the same on-peak demand savings for winter and summer periods. The evaluator found that there was seasonal variation in demand savings for the evaluated measures. The evaluator also found that on-peak demand savings were particularly less than tracking estimates for measures considering primarily unoccupied energy savings due to the majority of energy savings occurring outside of on-peak demand hours. These deviations resulted in a much lower realization rate for demand savings than for energy savings.

1.2 Recommendations for Program Designers & Implementers

The sampling approach used by the TA vendor to verify energy savings calculated by the site is understandable given the number of measures implemented and the varying levels of complexity of the different measures. Due to this sampling approach some of the measures were not reviewed by a TA vendor or RI Energy.

There may be an opportunity to revise the review process. Modifying the MOU structure to include periodically meeting with RI Energy and the TA vendor to discuss ongoing RCx measures and review savings analysis methodology in more detail would likely improve the quality of the savings analysis by providing feedback to the site on an ongoing basis rather than trying to correct everything at one time when the application is submitted. These meetings could function similar to an MRD and provide guidance on the trends needed to demonstrate the energy savings associated with the RCx measures that are being implemented. Areas for improvement that could be addressed specifically through this process are as follows:

1. The TA vendor can provide feedback geared towards improving project descriptions for retrocommissioning measures in the context of energy savings. This would mean clearly documenting and defining the existing case baseline and the basis of savings for the project. Addressing this issue may avoid the basis of savings errors evaluated for M3 and M6.

2. Identify critical assumptions for the site to verify with spot checks or trend data collection. Examples specific to this evaluation project would be unoccupied chilled water load for M2 and winter chiller load for M5 and to confirm if the chillers cycle or run continuously.

3. More TA reviews would provide a layer of scrutiny to some of the site's assumptions that were not caught in the sampling approach such as the chiller operating continuously for M2 or that fan savings were calculated for DAT reset for M3.

1.3 Customer Alert

None.

2 EVALUATED MEASURES

The following sections present the evaluation procedure, including the findings from an in-depth review of the supplied applicant calculations and the evaluation methodology determined to be the best fit for the site and the information available.

2.1 Application Information and Applicant Savings Methodology

This section describes the applicant's application information, savings methodology, and the evaluation assessment of the savings calculation algorithm used by the applicant.

2.2 Applicant Description of Baseline

This section includes the applicant's description of baseline conditions for the six evaluated measures considered for this site.

M1 Replace FCU Thermostats

Existing FCU controls include imprecise manual dial controls and no scheduling capabilities.

M2 Chiller Occupancy Controls

Chiller is enabled continuously during unoccupied times. One pump is running during chiller downtime. Trends indicate zone temperature is ~65F when setpoint is 76F. The chiller serves a dual temperature loop and is only enabled seasonally when the system is in cooling mode. The switchover from heating to cooling is completed manually by the facility operations staff.

M3 VAV Controller Replacements

Existing VAV controllers provide faulty RAT readings resulting in inefficient DAT reset implementation and discharge air temperature setpoint being maintained at 55°F at all times.

M4 Chiller OAT Controls

No enable/disable command from the BAS. Chiller can be manually enabled/disabled on-site but trends show that the chiller is never disabled. Winter building thermal loads do not justify enabling the chiller year round.

M5 Dining Area AHU Controls

Baseline for this measure is based on the following RCx findings:

TAB and trends showed AHUs S3, S4, S5 and S6 and exhaust fans V3 and V4 are on year-round at relatively constant airflow (CAV).

AHUs all have electric reheat enabled to maintain zone temperature set point. The HVAC configuration is one zone per AHU system although these zones are all part of a large dining area.

Note units with relatively low min OA had higher use of electric reheat --due to infiltration from supply/exhaust air imbalance.

M6 Exhaust Fan Staging

An exhaust fan bank of constant speed constant volume strobic exhaust fans provide lab exhaust for an academic lab building. The baseline condition is assumed such that three fans were running continuously for half the time, and cycling between two and three for the remaining time and that this operation is due to faulty dampers.

The baseline values used in the applicant savings analysis are presented in Table 2-1.

Table 2-1. Applicant baseline key parameters

		BASELINE		
Measure	**Parameter**	**Value(s)**	**Source of Parameter Value**	**Note**
M1	Fan coil control strategy	Manual thermostat	Existing Case	
	Conditioned area	10,446 ft²	Building Area	
M2	Chiller Occupancy Schedule	24/7	Existing case	
	Chiller Capacity	11.4 tons	Nameplate	
	Chiller Demand	9.1 kW	Manufacturer data (55F CHW @ 75F OAT)	
	Zone Temperature	65°F	Site observation	
	Zone-Temperature Setpoint	76°F	Existing condition	
M3	Average Return Fan Speed	100%	Assumption informed by trend data	
	Return Fan Motor Size	7.5 HP	Nameplate	
	Average Supply Fan Speed	100%	Assumption informed by trend data	
	Supply Fan Motor Size	20 HP	Nameplate	
	AHU Operating Hours	6,570 hours	Assumes 18 hours/day	
M4	Chiller Operating Hours	8,760 hours	Applicant assumption	
	Average Chiller demand in Winter	10 kW	Applicant assumption 25% average load	
M5	Hours when OAT > Cooling Setpoint	2,262 hours	Bin model analysis	
	Average Chiller Load	177,641 BTU/hr	Bin model analysis	
	Average Exhaust Fan Power	7.6 kW	Calculated from design CFM and total static pressure	
	Average Supply Fan Power	15.5 kW	Calculated from design CFM and total static pressure	
	Annual Fan Hours	8,760	Existing Case schedule	
	Average Reheat Load	76,057 BTU/hr	Bin model analysis	
	Reheat Run hours	2,262 hours	Bin model analysis	
M6	Exhaust Fan Power	30 HP	Nameplate	
	Three Exhaust Fans Running Annual Hours	6,570 hours	Assumption informed by trends	

2.2.1 Applicant Description of Installed Equipment and Operation

M1 Replace FCU Thermostats

New digital thermostats with BMS communication are installed to control all of the hydronic FCUs.

M2 Chiller Occupancy Controls

An occupancy schedule is implemented to disable the chillers during unoccupied hours unless there is a call for cooling in the building.

M3 VAV Controller Replacements

VAV controllers are installed that provide accurate temperature readings. During the implementation of this project, it was identified that construction filters were installed on return ducts and these construction filters were removed.

M4 Chiller OAT Controls

Chilled water system control sequence is modified to include an outside air temperature enable setpoint.

M5 Dining Area AHU Controls

Dining area ventilation system is controlled to operate according to an occupancy schedule that is programmed in the BMS.

M6 Exhaust Fan Staging

Strobic exhaust bank dampers are repaired allowing for improved exhaust fan staging, i.e. fewer average exhaust fans needed to run to maintain exhaust duct static pressure setpoint.

The proposed system values used in the applicant savings analysis are presented in Table 2-2.

Table 2-2: Application proposed key parameters

		BASELINE		
Measure	**Parameter**	**Value(s)**	**Source of Parameter Value**	**Note**
M1	Fan coil control strategy	T-Stat w/ occupancy sensor and BMS communication for schedule control	Installed Case	
	Space Cooling Temperature Setpoint	74°F Occupied 76° Unoccupied	Trend data	
	Conditioned area	10,446 ft²	Building Area	
M2	Chiller Occupancy Schedule	6AM-6:30PM	Assumption informed by trend data	
	Chiller Capacity	11.4 tons	Nameplate	
	Chiller Demand	9.1 kW	Manufacturer data (55F CHW @ 75F OAT)	
	Zone-Temperature Setpoint	Occupied: 73°F Unoccupied: 76°F	Applicant assumption	
M3	Average Return Fan Speed	90%	Assumption informed by trend data	
	Return Fan Motor Size	7.5 HP	Nameplate	
	Average Supply Fan Speed	90%	Assumption informed by trend data	
	Supply Fan Motor Size	20 HP	Nameplate	
	AHU Operating Hours	6,570 hours	Assumes 18 hours/day	
M4	Chiller Operating Hours	5,160 hours	Applicant Assumption	
	Average Chiller demand in Winter	10 kW	Applicant assumption 25% average load	
M5	Hours when OAT > Cooling Setpoint	1,414 hours	Bin model analysis	

	Average Chiller Load	177,641 BTU/hr	Bin model analysis	
	Chiller Performance	1.1 kW/ton		
	Average Exhaust Fan Power	7.6 kW	Calculated from design CFM and total static pressure	
	Average Supply Fan Power	15.5 kW	Calculated from design CFM and total static pressure	
	Annual Fan Hours	5,749 hours	Proposed occupancy schedule	
	Average Reheat Load	76,057 BTU/hr	Bin model analysis	
	Reheat Run hours	1,414 hours	Bin model analysis	
M6	Exhaust Fan Power	30 HP	Nameplate	
	Three Exhaust Fans Running Annual Hours	0 hours	Assumption informed by trends	

2.2.2 Applicant Energy Savings Algorithm

The applicant has an MOU with RI Energy. The applicant implements retrocommissioning measures throughout the year and submits one application to cover all of the measures with savings calculations completed by the applicant. There is a RI Energy review process in which a sample of measures are evaluated using pre-installation and post-installation trend data and the realized savings are compared to the original applicant estimated savings for that measure. The applicant determines a realization ratio to apply across all measures included in the annual MOU savings based on a review of a sample of projects reviewed by the PA. The realization for this program year was 0.93. This realization rate is applied to the measures that were not included in the PA measure review procedure and is used to calculate the tracking savings for M1, M2, M3 and M4 which is why the tracking savings for these measures is slightly less than the result of the applicant's savings analysis. M5 and M6 were included in the PA review sample.

M1 Replace FCU Thermostats

Savings calculations are in a word document. The savings methodology for this measure is from Massachusetts eTRM Programmable Thermostat measure. The referenced value in the savings calculation is the electric saving coefficient: 0.154 kWh/sq. ft. which assumes previous erratic control.

Calculation:

Building square footage = 10,925 sq. ft.

Electric Savings

$$10{,}446\ ft^2 * 0.154\ kWh/ft^2 = 1{,}609.7\ kWh$$

The difference in square footage numbers above is not a typo and reflects the applicant documentation accurately. The reason for different numbers is reflects the applicant excluding unconditioned space in the savings calculation.

M2 Chiller Occupancy Controls

The savings for this measure are in a word document.

1) Unoccupied savings

Unoccupied hours after measure implementation are 6:30 PM to 6 AM on weekdays.

$$11.5\ hours/weekday * 5\ weekdays + 24\ hours/weekend\ day * 2\ Weekend\ days = 105.5\ hours/week$$

Assuming that 20% of the unoccupied hours the building still demands chilled water.

Summer chiller hours unoccupied (4 months or 18 weeks)

$$105.5\ hrs/week\ *\ 18\ weeks/year\ *\ 80\%\ of\ time\ chiller\ can\ be\ shut\ off\ =\ 1519.2\ hours/year$$

Chiller capacity is estimated to be 11.4 tons with a chiller demand of 9.1 kW based on manufacturer's performance data at the average nighttime chilled water supply temperature (55F) and outside air temperature (75F).

Assuming chiller was running at 80% load at night when temps are low the chiller demand

$$80\%\ *\ 9\ kW\ =\ 7.4\ kW$$

Chiller unoccupied energy savings

$$1{,}519\ hours\ *\ 7.4\ kW\ =\ 11{,}242\ kWh$$

Most of this time period – the pump (0.75HP) is observed to be off as well. (10-minute delay)

Assuming pumps cycle between the two, and 10% reduction in OFF times due to the pump-down delay

Pump unoccupied savings

$$90\%\ *\ 0.75\ HP\ *\ 0.746\ kW/HP\ *1{,}519.2\ hours\ =\ 765\ kWh$$

Total unoccupied savings

$$11{,}242\ kWh\ +\ 765\ kWh\ =\ 12{,}007\ kWh$$

2) Occupied savings

The applicant also estimated that chiller run time during mild outside air temperature during occupied hours could be reduced.

Occupied hours in summer operation

$$12.5\ hours/day\ *\ 5\ days/week\ *\ 18\ weeks/year\ =\ 1{,}125\ hours$$

Assuming chiller runs at full load during the day but only 50% of the time

Chiller occupied savings

$$9\ kW\ *\ 50\%\ *\ 1{,}125\ hours\ =\ 5{,}062\ kWh$$

Assuming 40% savings for pump time

Pump occupied savings

$$0.75\ HP\ *\ 0.746\ kW/HP\ *\ 40\%\ *\ 1{,}125\ hours\ =\ 251.78\ kWh$$

Total savings = 5,314 kWh

Total measure savings = 17,321.36 kWh

M3 VAV Controller Replacements

The savings for this measure are in a word document.

The supply fan and return fan were observed to run at 100% speed in the baseline. Fan speed is assumed to reduce to 90% in the proposed case. The fans are assumed to run for 18 hours/day. Annual energy from the reduction in fan speed are calculated using the formula below.

$$(100\%\ baseline\ fan\ speed\ -\ 90\%\ proposed\ fan\ speed) * (20\ SF\ hp\ +\ 7.5\ RF\ hp) * 0.746\ kW/hp * 18\ hours/day * 365\ days\ =\ 13{,}478\ kWh$$

M4 Chiller OAT Controls

The savings for this measure are in a word document.

The savings calculations assume the chiller serving and academic library building is enabled year-round in the baseline and will be shut down from November through March (150 days) in the proposed case.

The chiller is assumed to run at a minimum 25% of capacity in the winter. Manufacturer's data indicates that the chiller demand is 10 kW at 25% load.

Savings from reducing the chiller run hours are shown below.

$$10\ kW * 150\ days * 24\ hours\ =\ 36{,}000\ kWh$$

There are additional savings from shutting of the chilled water pump in the winter.

Pump savings:

2x 2HP motor each, both ran simultaneously when chiller was ON.

Motor savings:

$$4\ HP * 0.746\ kW * 150\ days * 24\ hours\ =\ 10{,}742\ kWh$$

Total savings

$$36{,}000\ +\ 10{,}742\ =\ 46{,}742\ kWh$$

M5 Dining Area AHU Controls

Savings calculations are in a spreadsheet model.

Chiller Savings

Existing Chiller kWh

Hours when OAT>space temperature setpoint = 2,262

Average mixed air temperature when OAT>cooling setpoint = 73.9°F

Supply Airflow = 8,685 CFM

Average Chiller Load

$$8{,}865\ CFM * 1.08 * (73.9°F\ -\ 55°F)\ =\ 177{,}641\ BTU/hr$$

Annual Chiller Load BTU

$$2{,}262\ hours * 177{,}641\ BTU/hr = \ 401{,}825{,}004\ \text{BTU}$$

Chiller kWh

$$401{,}825{,}004\ BTU\ /\ 12{,}000\ BTU/ton\ *\ 1.1\ kW/ton\ =\ 36{,}735\ kWh$$

Proposed Chiller kWh

Hours when OAT>cooling setpoint = 1,225 occupied, 1,037 unoccupied

Average OAT when OAT>cooling setpoint = 73.9°F

Supply Airflow = 8,685 CFM occupied, 2,171 CFM unoccupied

Occupied Chiller Load

$$8{,}865\ CFM\ *\ 1.08\ *\ (73.9°F\ -\ 55°F)\ =\ 177{,}641\ BTU/hr$$

Unoccupied Chiller Load

$$2{,}171\ CFM\ *\ 1.08\ *\ (73.9°F\ -\ 55°F) =\ 44{,}410 BTU/hr$$

Annual Chiller Load BTU = 1,225 hours * 177,641 BTU/hr + 1037 hours * 44,410 BTU/hr

Chiller kWh

$$263{,}697{,}659\ BTU\ /\ 12{,}000\ BTU/ton\ *\ 1.1\ kW/ton\ =\ 24{,}107\ kWh$$

Chiller Savings

$$36{,}735\ kWh - 24{,}107\ kWh\ =\ 12{,}628\ kWh$$

Fan Savings

Fan savings are calculated assuming that the installed schedule results in AHU supply fans and exhaust fans operating 13 hours a day every day of the week and cycling on 25% of the time during unoccupied hours compared to 24/7 occupied. This assumption results in a 3,011 annual operating hour reduction for the fans.

Exhaust Fan Savings

$$7.6\ kW\ *\ 3{,}011\ annual\ hour\ reduction\ =\ 22{,}765\ kWh$$

Supply Fan Savings

$$16.0\ kW\ *\ 3{,}011\ annual\ hour\ reduction\ =\ 48{,}177\ kWh$$

Electric Resistance Reheat Savings

Existing Reheat

Cooling setpoint = 72°F

Reheat Airflow = 3,856 CFM (excludes S5)

Reheat Average Load

$$3{,}856\ CFM\ *\ 1.08\ *\ (72°F - 55°F)\ =\ 70{,}796\ BTU/hr$$

Annual Reheat Load

$$2{,}262\ hours\ *\ 70{,}796\ BTU/hr\ =\ 160{,}140{,}914\ BTU$$

Annual reheat kWh

$$160{,}140{,}914\ BTU\ /\ 3{,}412\ BTU/kWh\ =\ 46{,}935\ kWh$$

Proposed Reheat

Cooling setpoint = 70°F occupied, 76°F unoccupied

Reheat Airflow = 3,856 CFM occupied, 964 CFM unoccupied (excludes S5)

Reheat Average Load Occupied

$$3{,}856\ CFM\ *\ 1.08\ *\ (70°F - 55°F)\ =\ 62{,}467\ BTU/hr$$

Reheat Average Load Unoccupied

$$964\ CFM\ *\ 1.08\ *\ (76°F - 55°F)\ =\ 21{,}864\ BTU/hr$$

Annual Load

$$1{,}225\ hours\ *\ 62{,}467\frac{BTU}{hr} + 1{,}037\ hours * 21{,}864\frac{BTU}{hr} =\ 100{,}088{,}071\ BTU$$

Annual reheat kWh

$$100{,}088{,}071\ BTU\ /\ 3{,}412\ BTU/kWh\ =\ 29{,}334\ kWh$$

Reheat Savings = 17,600 kWh

Total Measure Savings

$$12{,}628\ Chiller\ kWh\ +\ 22{,}765\ Exhaust\ Fan\ kWh\ +\ 48{,}177\ Supply\ Fan\ kWh\ +\ 17{,}600\ Reheat\ kWh = 95{,}907\ kWh$$

The savings for this measure were updated in January 2023 to 102,318 kWh however this update is not reflected in the tracking savings for the project. It is not clear why these savings were updates and the tracking savings were not updated.

M6 Exhaust Fan Staging

M6 was included in the PA review sample. Preinstallation and post installation trend data was used to document the average number of exhaust fans running before and after the damper controls. The trend period is 1/30/2019 through 8/30/2022 and the data is in change-of-value format meaning that each point in the data set reflects a time when one of the four fan statuses changes value (i.e. one of the fans cycles on or off). The applicant calculates that the average number of fans running in the pre-installation case is 2.05 fans. In the post-installation case, the average number of fans running is 1.40 fans. The applicant uses a one-line formula to calculate fan savings.

$$(2.05\ fans\ -\ 1.4\ fans) * 8{,}760\ hours * 30\frac{HP}{fan} * 0.746\frac{kW}{HP} = 126{,}623\ kWh$$

2.2.3 Evaluation Assessment of Applicant Methodology

M1 – Replace FCU Thermostats

The applicant uses TRM calculations for this measure. The evaluator agrees with the applicant that TRM calculations are acceptable for this measure.

M2 – Chiller Occupancy Controls

The applicant references the existing condition as "chiller enable' and "chiller is running" interchangeably. It is possible that there is a misunderstanding and the chiller is enabled but not running during unoccupied periods and if this is the case the energy savings calculated for this measure are likely overstated. The pre-install trends show chiller enable as a point and chilled water supply temperature as a point. The chilled water supply temperature is a better indicator of chiller operation than the chiller enable point and the chilled water supply temperature fluctuates in a ~10°F deadband which indicates the chiller cycling on to meet the cooling load on the chilled water loop. The analysis assumes that the chiller runs continuously at 80% load at night when temperatures are approximately 75°F. Based on the pre-installation chilled water temperature trends the impact of cycling should have been accounted for and used to modify the run hour assumption.

Disabling the chiller during unoccupied does provide savings. These savings are supported by the trend data, however the baseline chiller run hours appear to be overstated because the applicant does not account for cycling.

M3 – VAV Controller Replacements

There are two sources of savings cited by the applicant; fixing failed controllers to achieve accurate RAT readings that allow the DAT to reset and removing construction filters to eliminate a significant and unnecessary pressure drop on the system contributing to fan savings. The savings claimed by the applicant include a simplified approach to calculate fan savings and do not address the DAT reset savings.

The one-line calculation used by the applicant equates to 10% of nominal HP savings for the supply and return fan. The applicant description of the basis of savings is a 10% reduction in average speed however affinity fan laws are not used to convert fan speed reduction to BHP reduction so the description is either inaccurate or the calculation is incorrect.

The trend data used as evidence of savings compares AHU fan speed during September 2021 and November 2022. It is reasonable to expect that the loads on the VAV system would be different in September 2021 and November 2022. It is not clear that this difference in average fan speed is directly related to removing construction filters or replacing VAV controls as opposed to showing that the fan speed is different under different space load conditions.

Trends are not provided that demonstrate the impact of replacing of failed controllers on DAT reset. No savings were calculated for this aspect of the project.

M4 – Chiller OAT Controls

The savings for this measure do not directly reference the whole building kW data that was collected in support of the measure. This represents a missed opportunity to calculate savings with better accuracy than the one line equation used by the applicant.

The savings calculation assumes that pump BHP is equal to the nominal HP of the pump which is not a reasonable assumption.

M5 – Dining Area AHU Controls

The savings for this measure are calculated using a series of single line equations. The claimed savings are almost 100,000 kWh. A more detailed analysis approach using temperature bins would have been more appropriate and would account for the interaction between the proposed operating schedule and the corresponding outside air conditions.

M6 – Exhaust Fan Staging

The applicant calculates average fan power by converting nominal HP to kW using straight forward unit conversion. This methodology omits motor load factor assumptions (fan brakehorsepower is less than motor horsepower) and motor efficiency (or assumes 100% for both which would not be reasonable).

The applicant savings analysis for this measure uses change of value trend data to calculate the average number of fans running before and after the damper repairs were completed. A feature of change of value trends are that the time interval of the trend data is irregular and depends on when there is a change in value of the point being trended. In the case of the data used to analyze this measure, that means if a fan cycles on or off then a point will be generated with a time stamp reflecting when the event occurred. The average number of fans running calculated by the applicant does not account for the total runtime in that state. As an example; if a fan cycles on for an hour and then cycles off it will have the same weight as an instance of a fan cycling on and running for a week before cycling off using the applicant's approach.

The evaluation analysis of this measure uses the same data, but considers runtime in calculating the average number of exhaust fans operating before and after the damper repairs were completed.

The following tables consider the same set of data and show the different results when considering and not considering runtime.

Table 2-3 shows the results of average the number of fans running over the time period before and after damper repairs were made counting each data point with equal weight as done by the applicant.

Table 2-3. Applicant Average number of Fan Results

Parameter	Pre-Installation	Post-Installation
Number of fans running	2.05	1.40

Table 2-4 shows the results when factoring runtime at each state to calculate the average number of fans running before and after the damper repairs. A data point indicates that there is a state change and the value for that data point indicates the new state. The runtime at any given state can be calculated by taking the difference in time stamp between two data points. Note that the exact same set of data and repair date assumptions are used for this table as the previous table.

Table 2-4. Evaluator Corrected Applicant Analysis Average number of Fan Results

Number of fans	Pre-Install		Post-Installation	
	Runtime (Days)	Data Points	Runtime (Days)	Data Points
0	222	213	0	14
1	231	3,244	0	15
2	760	11,866	47	13
3	48	4,583	0	10
Total	1,261	19,906	47	52
Average no. of Fans*	1.82		2.00	

**Average does not include runtime w/ 0 fans operating*

This shows that the evidence for energy savings associated with this measure does not exist when this same dataset is assessed with runtime factored appropriately.

2.3 On-site Inspection and Metering

This section provides details on the tests performed during the on-site inspection. Evaluators were granted access to the site and conducted a full M&V evaluation.

2.3.1 Summary of Site Visit

This section summarizes the site visit.

- The evaluator visited the site on September 29, 2023 to install meters and meet with the Senior Energy Engineer at the facility. The evaluator returned to the site on January 31, 2024 to retrieve the meters.

- The evaluator documented the new thermostats installed associated with M1.
- The evaluator installed a kW meter on the chiller and two (2) dual temperature pumps serving the academic department building associated with M2.
- The evaluator installed kW meters on the supply fan and the return fan for AHU-1 associated with M3.
- The evaluator installed amperage loggers on the four (4) dining area duct heaters and kW meters on the four (4) AHUs (S3, S4, S5 and S6) serving the dining area associated with M5. During meter retrieval, the evaluator spot metered the amperage of one of the two exhaust fans (V3) that serves the dining area.
- For M6 the evaluator spot metered kW demand for the two strobic exhaust fans that were operating during meter retrieval. There are four fans total and two were operating which was expected based on the applicant description of M6 as well as the discussion with the site contact.
- After meter retrieval the evaluator reviewed available trend data with the site contact. Following the site visit, the site contact downloaded the requested trends and provided them to the evaluator.

Table 2-5. Measure Verification

Measure Name	Verification Method	Verification Result
M1 – Replace FCU Thermostats	Confirm installation of new thermostats with BMS communication	Site observation confirms that new thermostats have been installed and trends confirm that the thermostats communicate with the BMS and set the space temperature setpoint back during unoccupied hours (Figure 2-8).
M2 – Chiller Occupancy Controls	kW metering and trend review	Trends show that chiller controls reverted to baseline sequence (Figure 2-11).
M3 – VAV Controller Replacements	kW metering and trend review	Applicant's baseline and basis for savings does not result in energy savings for this measure
M4 – Chiller OAT Controls	Trend review	The evaluation used whole building electric meter trends to confirm savings. Trends were used to verify chiller control sequence includes OAT enable (Figure 2-14).
M5 – Dining Area AHU Controls	kW metering and trend review	Metered data and trend data indicate that a schedule has been implemented as expected and the ventilation systems cycle off during unoccupied mode (Table 2-8).
M6 – Exhaust Fan Staging	kW metering and trend review	Metered data and trend data indicate that recent damper failure impacted measure persistence. Trend data does not support claim that measure reduced average number of fans operating and no savings are evaluated for the measure.

2.3.2 Measured and Logged Data

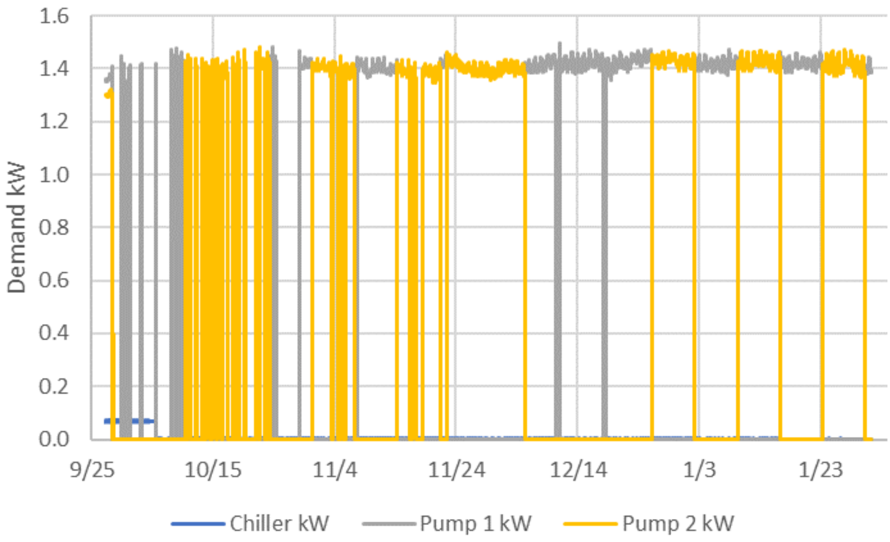
Table 2-6 summarizes the metered data collected and the period of useful data collected for each piece of equipment. The metered data from the useful data period is used in the evaluation savings analysis.

Table 2-6. Metered Data Summary

Metered Load	Data Interval	Metering Period
M2 – Chiller, Pump 1 and Pump 2 kW	1-minute	9/27/23-1/31/24
M3 – AHU Supply Fan and Return Fan kW	1-minute	9/27/23-1/31/24
M5 – S3, S4, S5, S6 supply fan kW and Reheat Amps	5-minute	9/27/23-1/31/24
M5 – Exhaust Fan V3 amperage	5-minute	15-minutes
M6 – Exhaust Fans EF-3 and EF-4 kW	1-second	27-minutes

The metered data for the chiller and two pumps associated with M2 are presented in Figure 2-1.

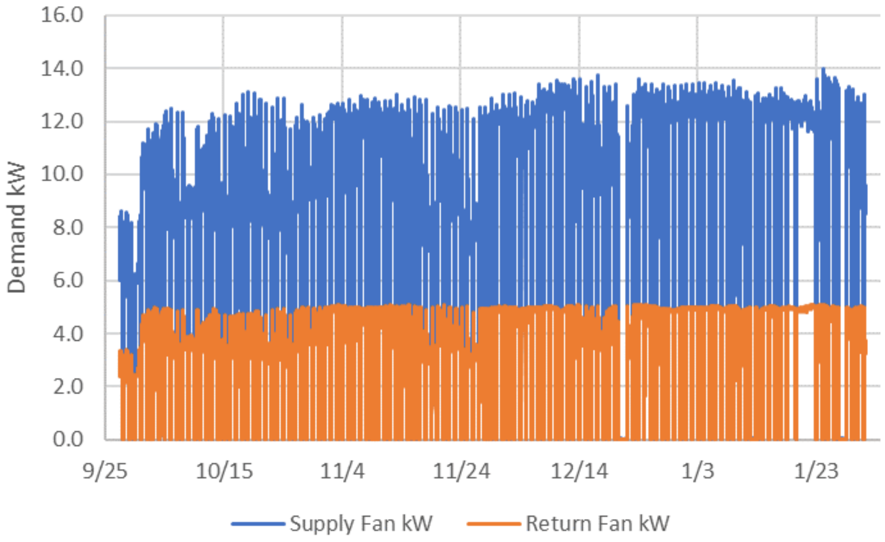
Figure 2-1. M2 Chiller and Pump Metered kW



The chiller is off for the duration of the metering period. The two pumps serve the dual temperature loop in the building. The metered data shows the demand of the constant speed pumps and the operating profile is reflective of pump control when the loop is in heating mode.

The metered data for AHU supply and return fan demand kW associated with M3 are presented in Figure 2-2.

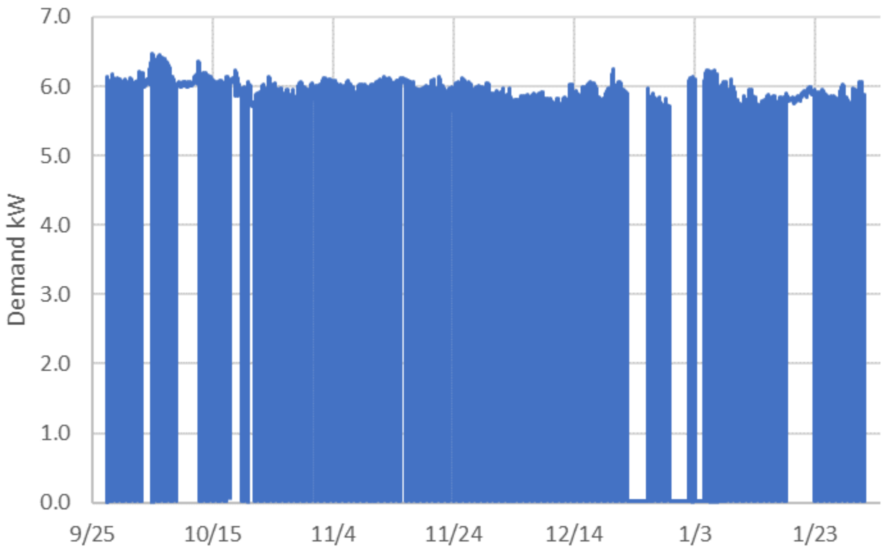
Figure 2-2. M3 AHU Supply and Return Fan kW



The metered data shows that the fans run on a schedule and that the supply fan runs at ~10 kW when it is on and the return fan runs at ~4 kW when it is on.

The metered data for one of the AHU supply fans associated with M5 is presented in Figure 2-3. Metered fan kW data was collected for the other three AHU supply fans associated with this measure.

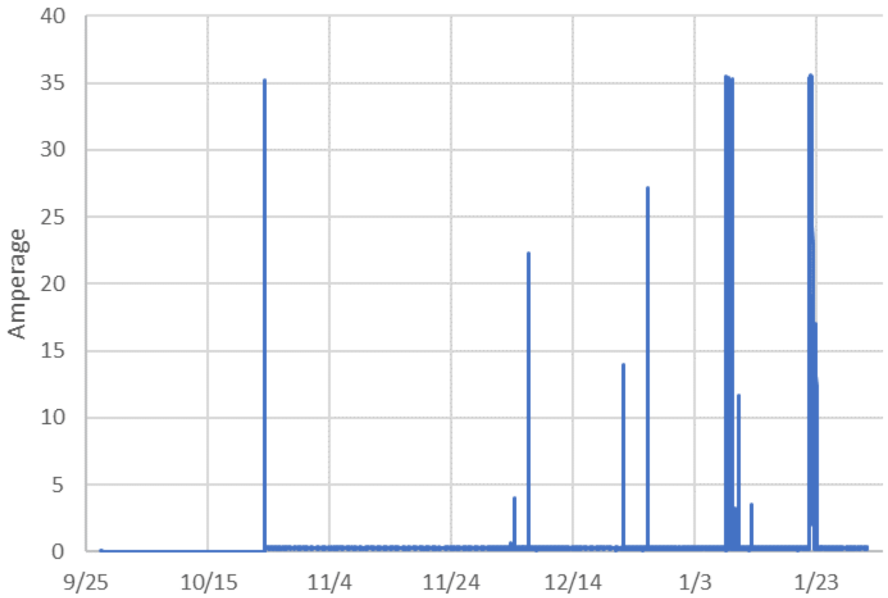
Figure 2-3. M5 AHU S3 Supply Fan kW



This data shows the fan operates on an occupancy schedule and that the operating demand is ~6 kW. This is similar to the other AHU supply fans metered for M5.

The metered data for one of the electric duct heaters associated with M5 is presented in Figure 2-4. Metered heater amperage data was collected for the other three electric duct heaters associated with this measure as well.

Figure 2-4. M5 AHU S3 Duct Heater Amps

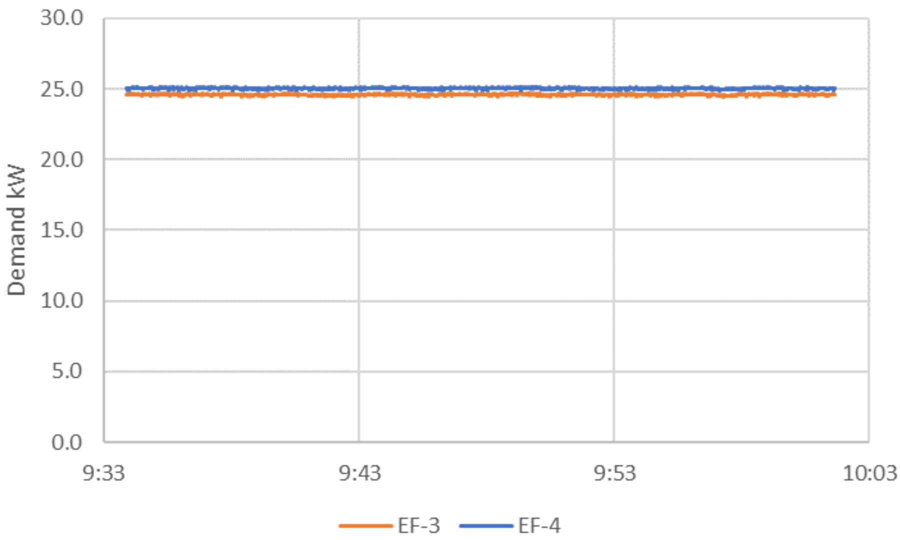


The data shows that the heater rarely runs and that when it cycles on the heater draws ~35 amps.

One of the two constant speed constant volume exhaust fans associated with M5 was spot metered with an amperage data logger. The spot metered data showed an average amperage of 7.5 amps at 208 three-phase voltage.

Two of the four constant speed constant volume strobic lab exhaust fans associated with M6 were spot metered. The spot metered kW data collected for exhaust fans EF-3 and EF-4 is presented in Figure 2-5.

Figure 2-5. M5 Exhaust Fan EF-3 and EF-4 kW



This spot metered data shows the two strobic exhaust fans operate at ~25 kW each.

The trend data collected by the evaluator is summarized in Table 2-7.

Table 2-7. Trend Data

Data Type	Time Period	Interval

M1 – Sample of 6 FCUs, Zone temperature, Zone temperature setpoint, system mode, unit status, Fan Status.	9/1/23 – 9/30/23 and 1/1/24 – 1/31/24	10-minute
M2 – Chiller enable, CHW supply temperature, Pump status	9/1/23 – 9/30/23	10-minute
M2 – Chiller enable, pump enable	6/2/24 – 7/11/24	Change of value
M3 – Discharge air temperature, mixed air temperature, Supply Fan Speed, Return Fan Speed, DAP, SAP setpoint, Reheat Valve Position, Cooling Valve Position, Return air temperature	9/1/23 – 9/30/23 and 1/1/24 – 1/31/24	10-minute
M4 – Whole building kW	1/1/23 – 1/31/24	15-minute
M4 – Cooling system enable, Chilled water supply temperature, Chilled water pump status, outside air temperature	11/30/22 – 11/30/22, 4/1/23 – 4/30/23, 5/1/23 – 5/31/23	15-minute
M5 – for AHUs S3, S4 S5, S6 OAT, Supply Fan Speed, discharge air temperature, return air temperature, mixed air temperature, Reheat Output, Heating Coil Output	9/1/23 – 9/30/23 and 1/1/24 – 1/31/24	10-minute
M5 – Exhaust Fans EF V3 and EF V4 Status	9/1/23 – 9/30/23 and 1/1/24 – 1/31/24	Change of value
M6 – Exhaust fan status for EF1, EF2, EF3, EF4 and number of exhaust fans running	1/1/24 – 1/31/24	10-minute
M6 – Exhaust fan status for EF1, EF2, EF3, EF4 and number of exhaust fans running	Applicant trend data 1/30/2019 – 8/30/2022	Change of value

2.4 Evaluation Methods and Findings

This section describes the evaluator methods and findings.

2.4.1 Evaluation Description of Baseline

The evaluator measure event type for all of the evaluated measures is existing system baseline. All of the measures are retro commissioning measures. The measure implementation is related to controls with a measure life shorter than the expected useful remaining life of the underlying equipment.

M1 Replace FCU Thermostats

Existing FCU controls include imprecise manual dial controls and no scheduling capabilities.

M2 Chiller Occupancy Controls

The chiller serves a dual temperature loop and is only enabled seasonally when the system is in cooling mode. The switchover from heating to cooling is completed manually by the facility operations staff. The chiller is enabled 24/7 when the dual temperature loop is in cooling mode. Note that the evaluator does not equate the terms enable and running. Enabled means that the chilled water system is not disabled and is allowed to cycle on to serve a cooling load.

M3 VAV Controller Replacements

Trend data is not available indicating existing DAT profile for AHU. Applicant asserts that VAV controllers failed and provided incorrect RAT readings resulting in improper DAT reset controls. Applicant describes finding construction filters on return ducts.

M4 Chiller OAT Controls

The site interview was used to verify that prior to this mease, no enable/disable command from the BAS was used to control the chiller at this academic library. The chiller can be manually enabled/disabled on-site but trends show that the chiller is never disabled. Winter building thermal loads do not justify enabling the chiller year round.

M5 Dining Area AHU Controls

No occupancy schedule controlling dining area ventilation equipment. AHUs and exhaust fans operate 24/7 per trend data collected by the applicant.

M6 Exhaust Fan Staging

Site repaired faulty exhaust dampers. Trends are used to calculate average number of exhaust fans running prior to repairs and after repairs.

2.4.2 Evaluation Calculation Method

M1 Replace FCU Thermostats

The evaluator confirmed via the site contact that the existing case FCU control was a manual dial that controlled the output of each FCU and was adjustable by the building occupants. A picture of one of the existing case controllers was provided by the site contact and is presented in the figure below.

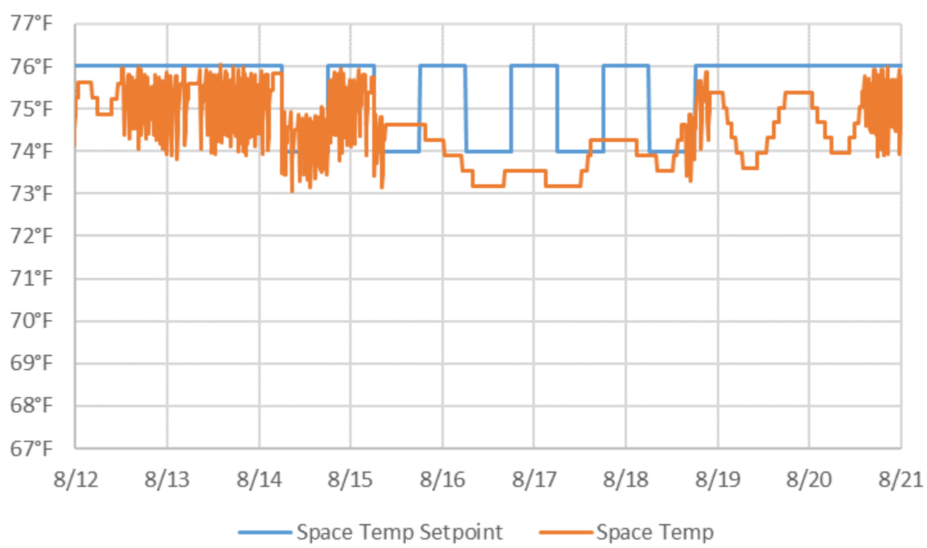
Figure 2-6. Existing Case FCU controls

The evaluator confirmed that the measure is installed. There are new thermostats that communicate with the BMS. The facility staff is able to set space temperature setpoints remotely. The site implements space temperature setbacks based on a 7-day schedule. The evaluator took a picture of one of the new installed thermostats which is presented in the figure below.

Figure 2-7. Installed Case FCU controls

Trend data was collected for a sample of spaces confirming that space temperature setpoints are being setback during scheduled unoccupied hours and that space temperature is responding to the setpoint and schedule set at the BMS. Figure 2-8 presents a week of data for one of the zones that demonstrates the temperature control profile in the installed case for this measure.

Figure 2-8. Room 305 Space Temperature Profile



The basis of savings for this measure is consistent with the TRM approach used by the applicant. The measure improved the accuracy and automation of the temperature controls in the building. The existing manual controls don't allow for scheduling and rely on occupant input or physical changes made by maintenance staff to change settings. Because the basis for savings was confirmed and due to the magnitude of savings for this measure compared to the other measures included in this application, the evaluated savings are calculated using the same methodology as the applicant from the following eTRM – Programmable Thermostat formula using the conditioned area of the build, 10,446 ft². The savings factor for on peak demand savings is 0 for this measure per the eTRM.

$$10{,}446\ ft^2 * 0.154\ kWh/ft^2 = 1{,}609.7\ kWh$$

M2 Chiller Occupancy Controls

The evaluator collected metered kW data for the chiller and two dual-temperature pumps that serve the building. The chiller did not run during the metering period so the data does not reveal any information about the chiller operating profile.

The evaluator has chilled water system trends from three time periods showing chiller enable and chilled water supply temperature. The applicant collected baseline (pre-installation) data in 2021 and installed trend data in 2022 and the evaluation period in summer 2023. The data indicates that the chiller was enabled 24/7 in 2021, occupied controls were implemented in 2022. The occupied controls were manually overridden from 8/25/2023 to 9/27/2023 but this override was removed and occupied controls were enabled after that time period. The chilled water supply temperature is used as an indicator of chiller operation.

Figure 2-10 shows the average hourly chilled water temperature for each day of the week during the pre-installation data set (8/17/2021-8/31/2021) indicating that the chilled water supply temperature setpoint is maintained at 55°F continuously.

Figure 2-9. Baseline (Pre-Installation) Chilled Water Supply Temperature Profile

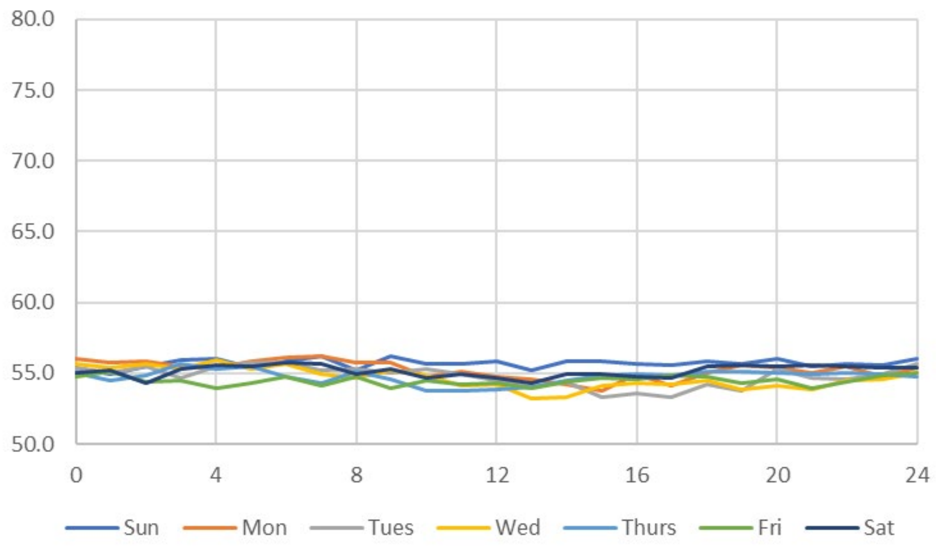
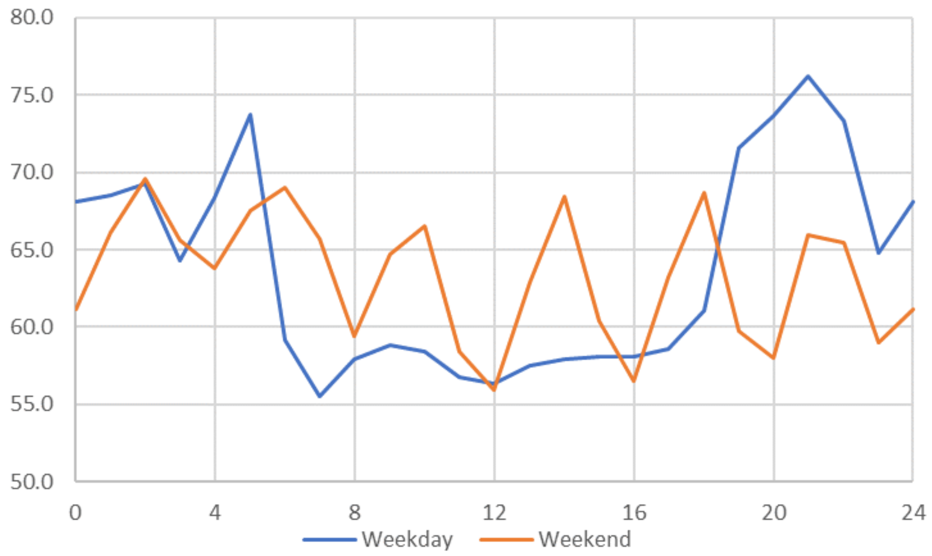


Figure 2-10 shows the average hourly chilled water temperature for each day of the week during the post-installation data set (7/24/2022-8/4/2022).

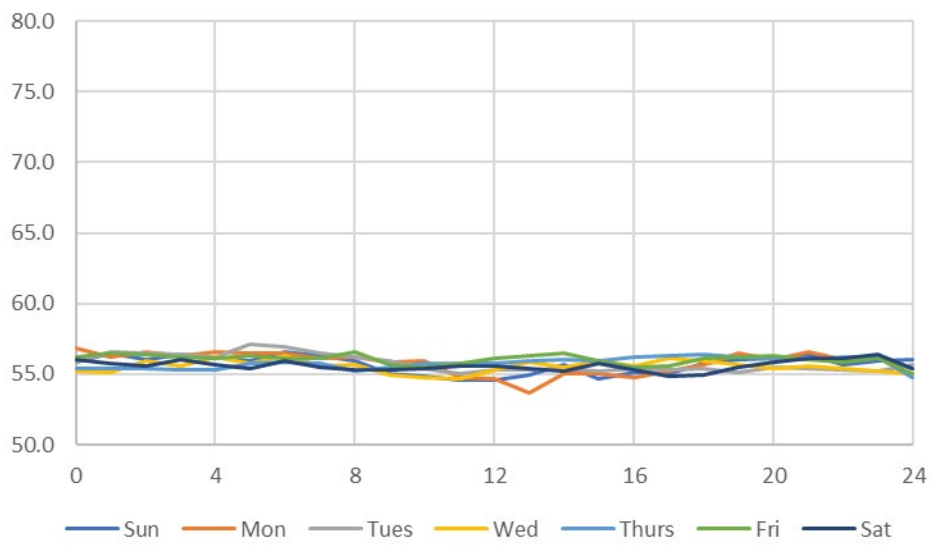
Figure 2-10. Post-Installation Chilled Water Supply Temperature Profile



This data indicates that occupancy controls are impacting the chilled water supply temperature profile. The applicant indicated that the occupied hours are 6AM-6:30PM on weekdays. The chilled water temperature is warmer outside of these hours on weekdays and during the weekends. During unoccupied weekday hours the chilled water temperature floats to warmer temperatures. During weekends the chilled water temperature is generally warmer with the chiller cycling on to provide colder chilled water in response to cooling loads in the building.

Figure 2-11 shows the average hourly chilled water temperature for each day of the week during the evaluation period data set (9/1/2023-9/27/2023).

Figure 2-11. Evaluation Chilled Water Temperature Profile



The site contact provided screenshots confirming that the reversion to baseline operating conditions was a result of a manual override. The occupancy controls were disabled with a manual override of the chiller enable on 8/25/2023 and this override was released on 9/27/2023. This encompasses the evaluation trend period and confirms that this operation is not representative of typical operation. The site contact provided chiller and pump enable trends from 6/3/2024-7/11/2024 that demonstrate the chilled water system and the pumps are enabled and disabled according to the same occupancy schedule observed in the 2022 post-installation trend data.

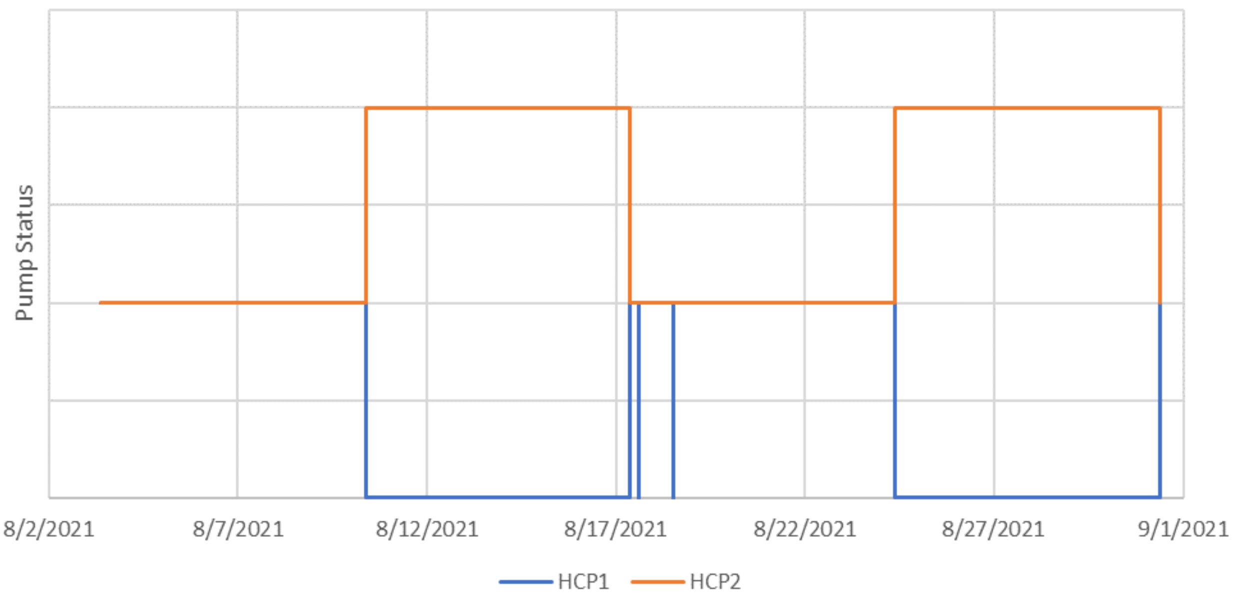
The chilled water temperature data shows that the temperature varies from 59°F to 52°F in the pre-installation time period. This temperature range indicates that the chiller cycles to maintain the average chilled water temperature shown in the previous figures. To estimate run time, the evaluator considers if the temperature for each line is less than 58°F as an indicator that the chiller operated a majority of the 10-minute interval. The same threshold and approach are used for the pre-install and post-install trend periods. A temperature bin model is used to calculate the average chiller runtime for 5°F temperature ranges for the pre-install and post-install trend periods. An occupied and unoccupied runtime profile is considered for the post-installation trend data based on the average chilled water temperature profile demonstrated in Figure 2-10.

Manufacturer performance data is used to estimate the operating chiller kW as a function of outside air temperature. The chiller demand is applied to the runtime profiles described above to calculate baseline and installed case average chiller demand kW.

The evaluated savings are calculated using an 8,760 hour bin model combining the runtime profiles for the existing baseline and installed conditions. The calculated runtime is applied to the estimated full load chiller demand based on the chiller manufacturer data. It is assumed that the chiller is enabled May through September based on feedback from the site. The hourly chiller demand is calculated as the factor of chiller demand and runtime.

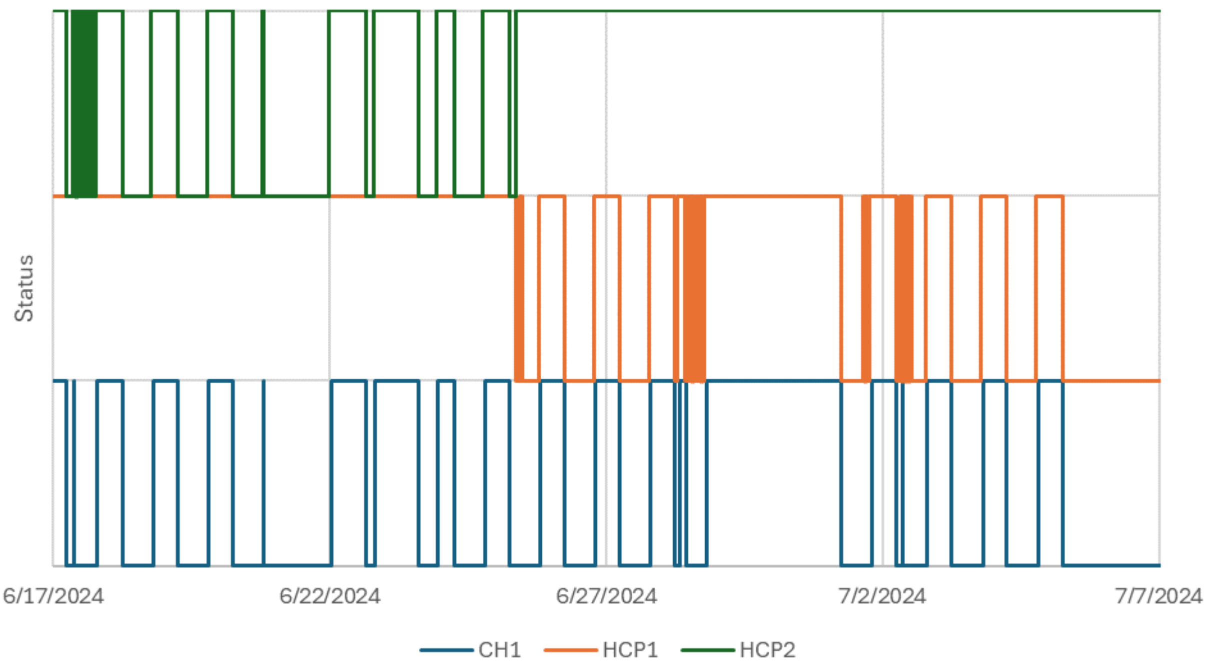
The evaluator found that this project resulted in a pump penalty. The pre-installation data demonstrates that the pump sequence maintained one pump operating at all times. The lead pump operates continuously and the lead pump assignment is changed every week between HCP1 and HCP2. Figure 2-12 demonstrates pre-installation pump operation.

Figure 2-12. Pre-Install Pump Operation



The stated intent of the measure is to limit pump operation during cooling mode to times when the chiller is running (i.e. there is a call for cooling). The trends demonstrate that the lead pump operates continuously and the lag pump stages on when the chiller is enabled resulting in a pumping penalty compared to the pre-installation pumping controls. Figure 2-13 demonstrates the evaluated pumping controls for M2.

Figure 2-13. Evaluated Pump Operation



A pumping penalty is evaluated for this measure based on the observation that two pumps run whenever the chiller is enabled in the post-installation case. The metered pump demand is 1.4 kW per pump.

The on-peak energy and peak demand savings were calculated with the same 8,760 spreadsheet model that was used to calculate annual savings.

M3 VAV Controller Replacements

This measure considers the replacement of failed VAV controllers. There are two sources of savings described by the applicant in the applicant documentation;

1. Replacing the VAV controllers resolves an issue with faulty return air temperature (RAT) readings that prevented the AHU discharge air temperature (DAT) reset sequence from functioning. With the new controllers, the DAT reset sequence operates as intended.
2. It was discovered that construction filters were still installed on return ducts during this project. The filters were removed and the pressure drop on the system decreased resulting in reduced average fan speed.

The documentation defining the baseline includes fan speed trends before and after the controllers were replaced. The applicant only considered fan savings associated with this measure.

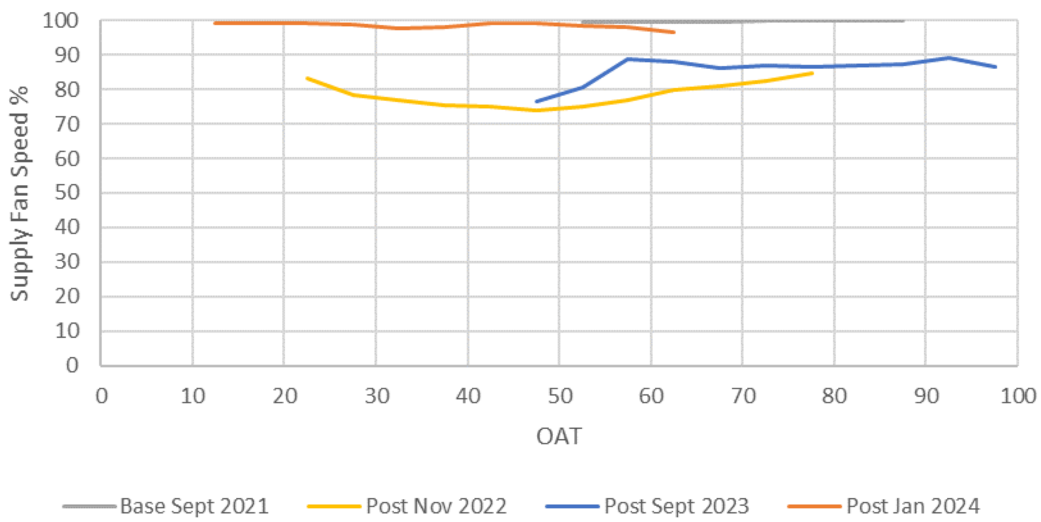
There are two factors that will impact fan energy that are under consideration for this project; changes in airflow, and changes in pressure drop.

The expected impact on average supply airflow associated with implementing discharge air temperature setback versus constant discharge air temperature is an increase in average airflow. A warmer discharge air temperature means that VAV boxes serving space cooling loads will require more airflow to serve the same space cooling load which results in a higher average airflow in the proposed case. The VAV box reheat load in space heating mode will be impacted by warmer supply air temperatures, but this change will not impact VAV airflow rates for VAV units in heating mode.

The removal of construction filters on some branches of the return ducts will impact the static pressure drop in the return air system. Trend data and site feedback indicates that return fan speeds are controlled to track supply fan speeds and supply fan speed is controlled to maintain static pressure setpoint. There is no indication that the control sequence was different prior to the implementation of this measure. Removing a pressure drop in the return duct will not directly impact return fan speed with this control sequence. A fan operating at the same speed with a lower pressure drop will operate at a different point on the fan curve providing more airflow at a higher fan power.

The evaluator collected supply and return fan speed trend data to determine if there was a change in fan speed from some action other than the removal of the construction filters. The applicant files for this project include fan speed trends during baseline / pre-installation (9/1/2021-9/30/2021) and post installation period (11/1/2022-11/30/2022). The evaluator collected more recent trends during the summer (9/1/2023-9/30/2023) and winter (1/1/2024-1/31/2024). The evaluator compared average fan speed per 5°F outside air temperature bin during occupied hours for these four periods to assess the impact of this measure on fan speed profile. This summary is presented in Figure 2-14.

Figure 2-14. AHU-1 Fan Speed Profile Comparison



The four time periods show different fan speed profiles. A challenge in analysing this data is that the applicant trends only include fan speed, i.e. there is no additional context to the fan speed trends. The most likely reason for the difference in fan speed profile is different load profiles during these different time periods. The site contact indicated that the space usage in this science building is variable and this likely accounts for the different load profiles during these different trend periods. Notably the evaluated fan profile for January 2024 shows that the fan runs at 100% almost all of the time it is on, which is the same operating profile as the baseline trend period from September 2021.

The impact of this change on the supply fan is more nuanced and depends on how the AHU and system pressurization is impacted by this change in return path static pressure drop. The supply fan speed modulates to maintain the supply duct static pressure and the supply duct static pressure is influenced by VAV boxes modulating airflows to satisfy zone conditioning loads. The impact of reducing pressure drop on the return side of the VAV system will not directly impact the supply fan control sequence and more baseline data would be required to document the indirect impact if any. The trend data available does not indicate a change in the operating profile of the air handler resulting from the removal of construction filters.

As discussed above discharge air temperature reset would not result in fan energy savings.

The source of energy savings associated with DAT reset would be cooling savings, not fan savings. Cooling savings would be achieved when a majority of VAV boxes are in re-heat mode. Increasing the discharge air temperature decreases the cooling load at the air handler and decreases the reheat load at the VAV box for this operating condition.

There is no baseline trend data supporting the claim that the discharge air temperature reset was not functioning prior to the VAV controller replacement.

The winter installed case trend data collected by the evaluator shows that the average supply fan speed increases at lower ambient temperature which does not indicate that a majority of VAV boxes are in heating mode (i.e. minimum airflow setting). No savings are attributed to changes in DAT reset based on this finding and lack of baseline data.

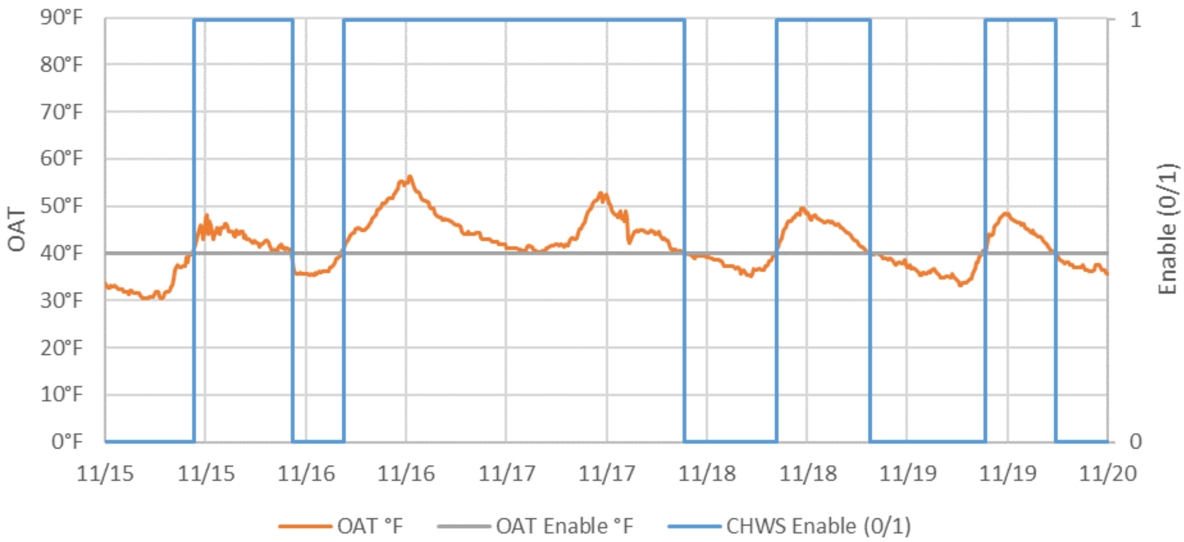
No energy savings are evaluated for this measure, so no peak demand savings are evaluated either.

M4 Chiller OAT Controls

Whole building 15-minute kW data is used to evaluate the energy savings for this measure and trends were used to confirm measure implementation. Site feedback indicates that the only change in building operation is the chilled water system control sequence so the change in energy use observed in the bill data can reasonably be attributed to the updated control sequence.

The trend data demonstrates that the chilled water system is enabled/disabled at 40°F outside air temperature. Figure 2-15 shows a time period during the shoulder cooling season where the outside air temperature is fluctuating above and below the outside air enable temperature and the chilled water system is being enabled and disabled automatically in response.

Figure 2-15. Chilled Water System Enable/Disable Profile



The evaluator calculates average hourly whole building demand when the outside air temperature is less than 40°F using a day-of-week/time-of-day matrix considering the time period before and after outside air temperature enable sequence was added to the chilled water system controls. Note that the correlation between outside air temperature and building demand for this data set is very poor with an

R^2 value less than 1 so outside air temperature is only used to consider the outside air enable controls in the evaluation analysis approach.

The trend period 1/1/2019-12/31/2021 is used to demonstrate building operation with no outside air temperature enable sequence for the cooling system, and the trend period 2/1/2022-11/7/2023 is used to demonstrate building operation with an outside air temperature enable sequence based on feedback from the site that the sequence was added in January 2022.

The baseline building demand profile calculated using this approach is presented in Table 2-8.

Table 2-8. Baseline Whole Building Demand Profile

Hour	Day of Week						
	Sunday	Monday	Tuesday	Wednesday	Thursday	Friday	Saturday
	1	2	3	4	5	6	7
0	44.5	43.4	43.9	43.7	43.9	45.3	44.9
1	43.9	43.9	44.0	43.7	45.3	44.1	43.9
2	44.4	43.5	44.7	43.8	44.0	44.4	45.1
3	43.6	44.4	43.6	42.9	44.4	45.2	44.6
4	43.4	43.4	44.2	44.0	44.5	45.3	44.5
5	44.1	43.3	43.3	43.7	44.8	44.8	44.0
6	43.8	43.8	44.5	43.2	45.0	44.9	44.2
7	43.9	43.0	44.4	44.0	44.9	44.8	43.5
8	44.5	45.5	46.9	47.2	46.6	48.5	44.6
9	44.6	51.0	52.8	51.8	53.8	53.5	45.5
10	45.2	52.1	54.0	52.8	55.7	54.4	45.4
11	43.8	52.4	56.7	54.5	56.2	53.7	44.2
12	44.0	51.0	55.5	55.1	56.6	55.8	46.1
13	44.0	51.8	56.3	56.6	55.5	55.2	43.7
14	44.6	51.7	57.4	57.6	55.1	56.9	43.5
15	44.5	53.4	56.4	54.2	55.8	55.3	44.2
16	43.1	50.4	55.4	55.4	55.3	53.9	45.9
17	42.8	48.3	48.1	50.5	54.4	49.7	43.1
18	43.0	45.2	43.8	44.9	52.3	46.7	42.6
19	43.7	43.4	45.1	44.9	46.4	45.8	43.3
20	44.3	44.4	44.2	44.5	43.9	43.3	44.6
21	44.7	44.4	44.2	44.9	44.9	42.8	43.8
22	44.2	44.2	44.1	43.7	44.6	43.8	43.7
23	44.5	44.4	43.7	44.3	44.3	44.1	45.9

The installed case building demand profile is presented in Table 2-9.

Table 2-9. Installed Whole Building Demand Profile

Hour	Day of Week						
	Sunday	Monday	Tuesday	Wednesday	Thursday	Friday	Saturday
	1	2	3	4	5	6	7
0	38.8	40.3	40.1	41.2	41.7	40.5	40.6
1	39.2	40.5	40.3	41.3	40.3	40.0	40.8
2	40.2	39.8	41.5	40.1	40.8	40.2	40.0

3	39.6	40.1	39.8	41.3	41.1	40.1	41.4
4	39.9	40.4	41.1	40.5	41.1	39.5	40.1
5	39.7	40.3	40.6	40.3	40.4	38.7	40.7
6	39.7	40.4	40.5	40.6	41.0	40.0	40.7
7	39.7	40.2	39.6	40.3	39.7	40.5	40.3
8	40.7	41.6	42.6	42.3	40.5	41.4	39.9
9	39.4	47.7	47.3	47.5	46.5	46.3	40.1
10	39.3	50.5	52.4	49.5	53.6	46.1	39.2
11	40.6	51.2	53.8	53.0	52.1	45.8	38.2
12	39.1	50.0	51.2	56.1	48.1	43.0	39.4
13	38.7	50.8	52.6	56.8	50.8	43.7	40.4
14	40.6	50.1	50.4	52.7	48.9	44.6	38.7
15	39.3	51.7	49.9	56.6	50.6	43.7	37.1
16	39.9	49.8	51.2	55.0	52.0	42.8	38.7
17	40.4	42.1	43.1	45.6	40.3	37.8	39.8
18	40.8	41.4	40.3	42.2	41.8	39.8	37.8
19	39.8	40.2	41.9	40.5	39.7	38.3	37.6
20	39.4	41.4	41.3	41.2	38.6	40.0	37.8
21	40.9	41.3	41.6	41.4	40.2	39.6	38.7
22	41.0	40.6	41.6	40.6	40.2	38.8	38.1
23	40.6	40.3	41.4	41.2	40.8	40.0	38.4

The difference in hourly demand as a function of weekday and hour is calculated and an 8,760 hour bin model using TMY3 weather data is used to calculate savings for the measure. Savings are only considered when the outside air temperature is less than 40°F (i.e. chilled water system is disabled per the control sequence).

The annual savings using this methodology is 12,094 kWh.

The on-peak energy and peak demand savings were calculated with the same 8,760 spreadsheet model that was used to calculate annual savings.

M5 Dining Area AHU Controls

Metered data was used to calculate the installed AHU operating schedule, fan power, exhaust fan power, and duct heater operating profile and evaluate energy savings for this measure. This measure considers a 24/7 operating schedule existing case baseline. The installed case is implementing scheduled occupancy controls via the BMS.

The fan operating profile for all four air handlers is calculated using a time of day/day of week matrix over the metering period. The metered data shows a strong correlation based on the operating schedule (i.e. time of day and day of week), but no correlation with outside air temperature. For this reason, the evaluator uses the observed operating schedule as the basis for projecting annual fan power for the four air handlers.

As an example of the approach used, the time of day/ day of week matrix used to model the operation of air handler S3 is presented in Table 2-8. Similar data is collected for air handlers S4, S5, and S6 and the operating schedule demonstrated by the time of day/ day of week analysis is the same for the other air handlers, i.e. the fans run 5AM-6:30PM and cycle occasionally outside of these scheduled occupied hours. The systems are VAV, however all four air handlers operate at close to the same fan speed during all occupied hours throughout the evaluation period.

Table 2-10. Air Handler S3 Fan demand kW Profile

Hour	Day of Week						
	Sunday	Monday	Tuesday	Wednesday	Thursday	Friday	Saturday
	1	2	3	4	5	6	7
0	0.4	0.8	0.4	1.0	0.7	0.3	0.9
1	0.4	0.7	0.7	1.0	0.9	0.3	0.9
2	0.4	0.7	0.4	1.0	0.7	0.5	0.7
3	0.4	0.7	0.4	1.2	0.9	0.7	0.9
4	2.2	2.6	2.5	3.1	2.7	2.5	2.8
5	4.9	5.3	5.6	5.6	5.6	5.6	5.0
6	4.9	5.3	6.0	5.6	5.6	5.7	5.0
7	5.0	5.3	5.6	5.6	5.6	5.9	5.2
8	5.0	5.3	5.6	5.6	5.6	5.6	5.1
9	5.2	5.3	5.6	5.6	5.6	5.6	5.0
10	5.2	5.3	5.6	5.6	5.6	5.8	5.0
11	5.0	5.3	5.6	5.6	5.6	6.0	5.3
12	5.0	5.3	5.6	5.6	5.6	5.6	5.3
13	5.3	5.3	5.6	5.6	5.6	5.6	5.1
14	5.0	5.3	5.6	5.6	5.6	5.6	5.0
15	4.9	5.3	5.7	5.6	5.6	5.9	5.1
16	5.2	5.3	5.9	5.6	6.0	5.9	5.3
17	5.1	5.3	5.6	5.6	5.8	5.6	5.2
18	4.9	5.3	5.6	5.6	5.6	5.7	5.0
19	3.3	3.4	3.6	3.8	3.4	3.7	3.1
20	0.8	0.7	1.3	1.2	0.3	0.5	0.7
21	0.7	0.7	1.1	1.1	0.5	0.6	0.4
22	0.7	0.7	1.0	1.2	0.6	0.7	0.4
23	0.9	0.7	1.0	1.2	0.5	0.8	0.4

The evaluated baseline fan power is the average occupied operating demand for each fan 24/7.

The exhaust fan was spot metered with an amperage logger. The exhaust fan does not have a VFD and provides constant volume airflow at a constant fan speed. Exhaust fan power was calculated assuming a power factor of 0.9. V4 is identical to V3 and it is assumed that exhaust fan V4 operates at the same fan power as exhaust fan V3. The evaluated baseline AHU fan power is the average occupied operating demand for each fan 24/7.

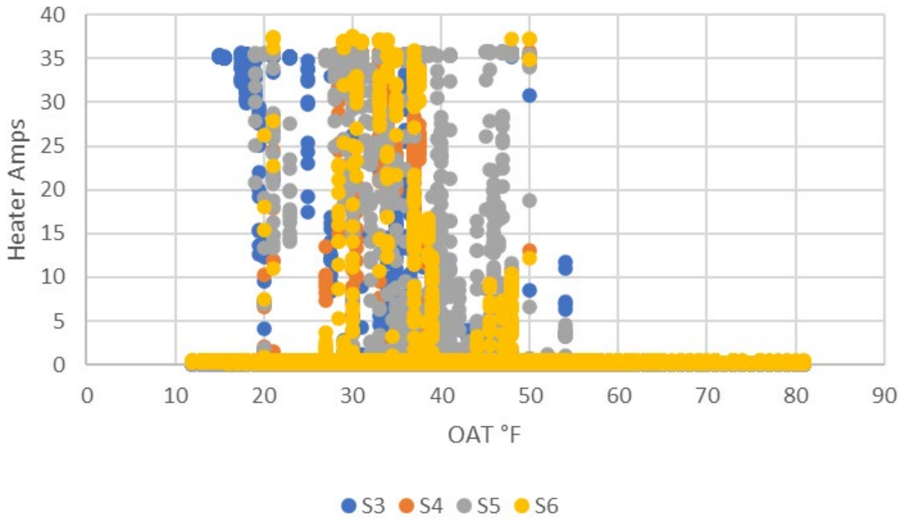
The average operating fan power for the four air handlers during occupied hours is presented in Table 2-11.

Table 2-11. Average Metered Operating Fan Power

Equipment	Average Operating Demand kW
AHU S3	5.9
AHU S4	3.7
AHU S5	4.0
AHU S6	3.6
EF V3	2.4
EF V4	2.4

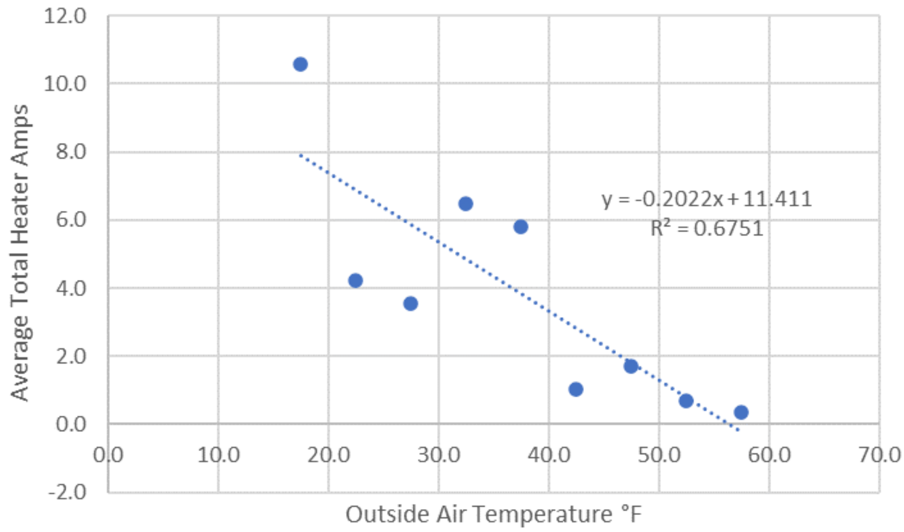
Each AHU has an electric resistance duct heater. Metered amperage data was collected for all four duct heaters during the same metering period as the metered kW data was collected for the AHU supply fans. The applicant assumed that the electric heaters would run when the building's dual temperature water loop is in cooling mode, however the data demonstrates that the electric heater only runs at cold ambient temperatures when the dual temperature loop is in heating/winter mode. Figure 2-16 shows the correlation between the raw heater amperage data for each AHU and outside air temperature.

Figure 2-16. Heater Amperage vs Outside Air Temperature



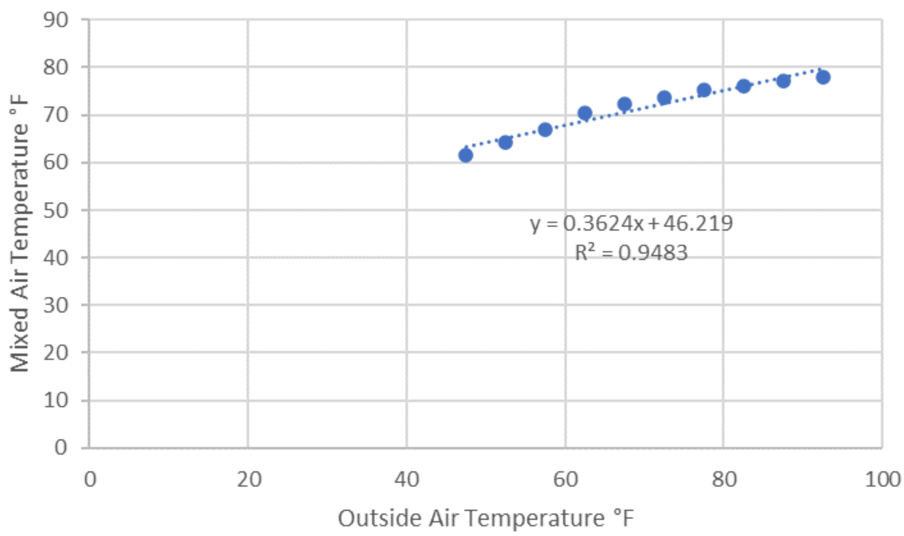
The duct heater operation is modelled using a 5°F temperate bin average of the total heater amperage (i.e. sum of the four heaters). This approach is used to account for the low runtime of the heaters. Figure 2-17 shows the resulting correlation between heater amperage and outside air temperature used in the energy savings analysis.

Figure 2-17. Average total Heater Amperage vs Outside Air Temperature



Trend data is used to calculate the cooling savings associated with this measure. Return air temperature, outside air temperature and mixed air temperature trends are used to calculate outside air percentage. The calculated outside air percentage is less than 5% outside air for all four air handlers indicating that the air handlers are controlled to operate in close to recirculation mode when in cooling. Return air humidity trends are not available so sensible cooling load is calculated only. Calculating sensible cooling load only is the same approach used by the applicant. The average discharge air temperature during cooling mode is 63.5°F. Mixed air temperature is calculated as a function of outside dry bulb temperature based on trend data. The relationship between mixed air temperature and outside air temperature is presented in Figure 2-18.

Figure 2-18. Average Mixed Air Temperature vs Outside Air Temperature



The total combined airflow for all four AHUs used for the cooling load calculation is the same as the applicant assumption which is based on a TAB data; 8,685 CFM. The sensible cooling load is calculated using the following formula.

$$Sensible\ Load\ \frac{BTU}{hr} = 8{,}685\ CFM * 1.08 * (MAT - 63.5°F)$$

Cooling energy is calculated assuming 1.1 kW/ton which is the same assumption used as the applicant and is a reasonable assumption for average cooling performance of an air-cooled chilled water system.

The evaluated energy savings for this measure are presented in Table 2-12.

Table 2-12. Evaluated Energy Savings for M-5

Equipment	Baseline kWh	Installed kWh	Savings kWh
Supply Fan	150,835	93,140	57,695
Exhaust Fan	42,388	28,259	14,129
Reheat	6,830	4,775	2,056
Chiller	22,229	16,389	5,840
Total			79,720

The on-peak energy and peak demand savings were calculated with the same 8,760 spreadsheet model that was used to calculate annual savings.

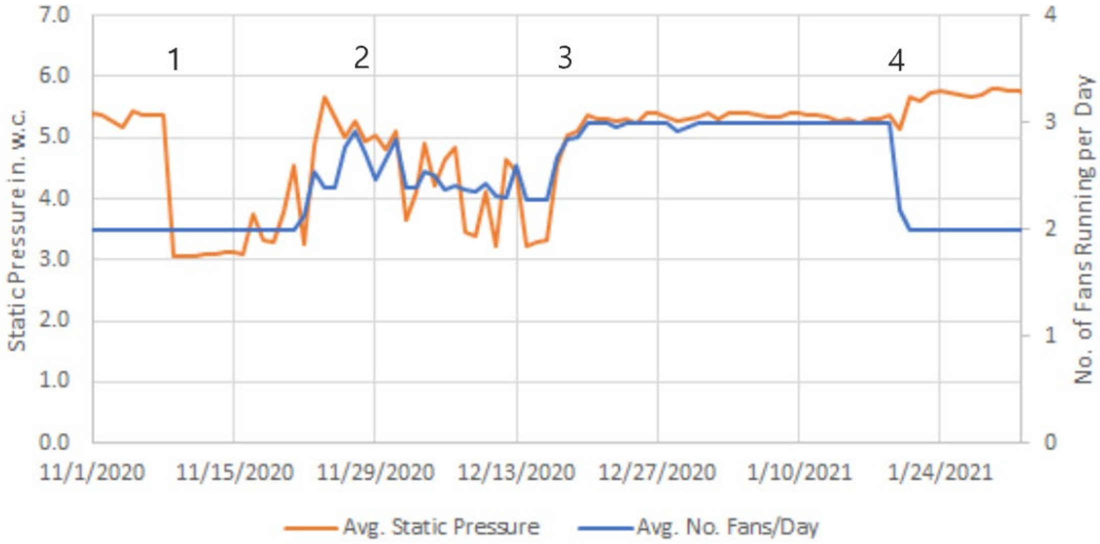
M6 Exhaust Fan Staging

The site provided a detailed timeline of the history of this project that is supported by the trend data that the evaluator used to adjust the baseline for this project. The timeline events are as follows.

1. November 10, 2020 there was a damper/actuator failure.
2. After this failure and until December 19, 2020 the fans were cycling between 2 and 3 fans and the system was unable to maintain static pressure at 5”. The 3rd fan would cycle on, the pressure would exceed the high pressure shutoff, then the 3rd fan would cycle off. The static pressure would drop below the low pressure limit with only 2 fans running causing the 3rd fan to cycle on and the cycle would repeat.
3. December 19, 2020 the static pressure control points were adjusted. The result of this change was 3 fans operating almost all of the time (3rd fan no longer cycles off on high static) and static pressure was maintained at or above ~5” with this adjustment.
4. January 20, 2021 the bypass damper was manually closed and the two fans with failed isolation dampers were set to run. This resulted in the system operating with 2 fans running and maintaining the static pressure at or above ~5”.
5. Work to replace dampers, actuators, etc. was completed over the course of the next year(s). This work improved automatic static pressure controls with proper bypass damper actuation, functional isolation dampers, etc.

The figure below shows the trend data that document this timeline of events.

Figure 2-19. Pre-Installation Operating Timeline



Based on this timeline, the time period between December 19, 2020 and January 20, 2021 is considered as the baseline operating condition. This time period represents the simple controls fix used to maintain health and safety requirements for the building and maintain the exhaust static pressure by adjusting the high pressure limit for the exhaust fan staging sequence to keep the 3rd fan running to compensate for the failed damper.

The bypass dampers are physically closed by facility staff on January 20,2021 and this represents the beginning of repairing the dampers for the fan bank. The damper repair work is ongoing from January 20, 2021 through July 2022 so this time period should not be considered pre-installation.

Table 2-13 shows the evaluated impact of damper repairs on exhaust fan staging with the adjusted baseline data period.

Table 2-13. Evaluated Damper Repair Impact on Exhaust Fan Staging

Number of fans	Runtime (Days)	
	2022 (pre-install)	2022 (post-install)
0	0.08	0.05
1	0.00	0.01
2	0.51	46.85
3	32.00	0.03
Total Runtime (Days)	32.59	46.94
Weighted Average*	2.98	2.00

**Average does not include runtime w/ 0 fans operating*

The spot metering results for the two operating exhaust fans at the time of the evaluation site visit was an average fan power of 24.8 kW per fan. The evaluated energy savings for this measure are calculated using the formula below.

$$Annual\ Energy\ Savings\ kWh = 24.8\ \frac{kW}{fan} * (2.98\ fans - 2.00\ fans) * 8{,}760\ hours = 214{,}015\ kWh$$

Peak demand savings for this measure are calculated assuming that the energy savings for this measure are evenly distributed across all operating hours and the exhaust fan system operates continuously all year.

3 FINAL RESULTS

This section summarizes the evaluation results determined in the analysis above. This section includes a summary table of savings by major end-use and application.

Table 3-1. Summary of Key Parameters

		BASELINE		PROPOSED / INSTALLED	
Measure	**Parameter**	**Tracking Value(s)**	**Evaluation Value(s)**	**Tracking Value(s)**	**Evaluation Value(s)**
M1	Conditioned Building Area	10,446 ft²	10,446 ft²	10,446 ft²	10,446 ft²
M1	FCU Controls	Manual dial	Manual dial	Programmable T-stats w/ BMS communication	Programmable T-stats w/ BMS communication
M1	Savings adjustment factor	0.93	N/A	0.93	N/A
M2	Chiller Occupancy Schedule	24/7	24/7	Monday-Friday 6AM-6PM	Summer of 2022 Monday-Friday 6AM-6PM Otherwise; 24/7
M2	Pumps running	1 at all times	1 at all times	0 when chiller off, 1 when chiller enabled	1 at all times, 2 when chiller enabled
M3	Fan Control Sequence	100% fan speed, sequence not described	SF modulates to maintain SP, RF tracks SF speed	90% fan speed, sequence not described	SF modulates to maintain SP, RF tracks SF speed
M4	Avg. Winter Cooling System Demand	Chiller: 10 kW Pumps: 3.0 kW	4.36 kW	Chiller: 0 kW Pumps: 0 kW	0 kW
M4	Cooling System Enable OAT	None	None	50°F	40°F
M4	Winter CHW System Run Hours	3,600 hours	3,317 hours	3,600 hours	3,317 hours
M5	AHU Fan Power	16 kW (S3-S5 combined)	17.2 kW (S3-S5 combined)	16 kW (S3-S5 combined)	17.2 kW (S3-S5 combined)
M5	Exhaust Fan Power	7.6 kW (V3&V4 combined)	4.8 kW (V3&V4 combined)	7.6 kW (V3&V4 combined)	4.8 kW (V3&V4 combined)
M5	Heater Power	21 kW	0.68 kW (S3-S5 combined)	21 kW	0.68 kW (S3-S5 combined)
M5	Combined AHU Supply Airflow	8,685 CFM	8,685 CFM	8,685 CFM	8,685 CFM
M5	Cooling DAT setpoint	55°F	63.5°F	55°F	63.5°F
M5	AHU Operating Hours	8,760	8,760	5,749	5,840
M6	Average Number of Fans running	2.05 fans	2.98 fans	1.40 fans	2.00 fans
M6	Average Fan Power	22.4 kW/fan	24.8 kW/fan	22.4 kW/fan	24.8 kW/fan

3.4 Explanation of Differences

This section describes the key drivers behind any difference in the application and evaluation estimates, annual kWh savings. The following table summarizes these differences. The purpose of this table is to describe how changes to the key parameters influenced the final project savings through the end-use summary analysis. Table 3-2 provides a summary of the differences between tracking and evaluated values.

Table 3-2. Summary of Deviations

Measure	Discrepancy	Parameter	Impact of Deviation	Discussion of Deviations
M1	Applicant Calculation Methodology - Analysis methodology	Applicant correction scaling factor	0.04%	**Increased savings –** The applicant applied a scaling factor to the calculated savings for this measure based on the result of a review process sampling measures and calculating a scaling factor based on the original estimated savings and the updated calculated savings. The evaluator does not use this scaling factor in the evaluation analysis resulting in an increase in evaluated savings.
M2	Applicant Calculation Methodology - Analysis methodology	TMY3 8,760 hour bin model vs. one line calculation	-2.5%	**Decreased savings –** The evaluator used an 8,760 hour bin model to calculate resulting in lower savings result than applicant's one line calculation method and accounted for chiller cycling.
M2	Controls	2nd pump cycles on with chiller	-1.1%	**Decreased savings –** Instead of only one pump running when the chiller runs, one pump runs all the time and the second pump cycles on with the chiller.
M2	Applicant Calculation Methodology - Analysis methodology	Applicant correction scaling factor	0.4%	**Increased savings –** The evaluator does not use the applicant's scaling factor in the evaluation analysis resulting in an increase in evaluated savings.
M3	Applicant Calculation Methodology - Unknown application algorithm or assumptions	No Evaluated Savings	-4.4%	**Decreased savings –** The evaluator does not calculate energy savings because there is no evidence of energy savings associated with this measure.
M3	Applicant Calculation Methodology - Analysis methodology	Applicant correction scaling factor	0.3%	**Increased savings –** The evaluator does not use the applicant's scaling factor in the evaluation analysis resulting in an increase in evaluated savings.
M4	Load shape	Billing analysis	-10.6%	**Decreased savings –** The evaluator calculates energy savings using whole building demand trend data that indicates the applicant overestimated the chiller load at low ambient temperatures.

M4	Applicant Calculation Methodology - Analysis methodology	Applicant correction scaling factor	1.2%	**Increased savings –** The evaluator does not use the applicant's scaling factor in the evaluation analysis resulting in an increase in evaluated savings.
M5	Load shape	Fan savings	3.3%	**Increased savings –** Evaluator's metered data indicates total average AHU fan power is greater than assumed by applicant resulting in increased savings
M5	Load shape	Duct heaters	-5.4%	**Decreased savings –** The evaluator's metered data shows much fewer heater run hours than assumed by the applicant resulting in a decrease in savings.
M5	Load shape	Exhaust Fan	-3.0%	**Decreased savings –** The evaluator collected metered data indicating lower exhaust fan demand than assumed by applicant resulting in reduced exhaust fan savings.
M5	Load shape	AHU Cooling Load	-2.8%	**Decreased savings –** Evaluator trend data shows higher average cooling DAT than assumed by applicant resulting in decrease in cooling savings
M6	Applicant Calculation Methodology - Unknown application algorithm or assumptions	number of fans	30.5%	**Increased savings –** The evaluator revised the methodology and time period considered for the baseline and resulted in more fans operating on average in the baseline for this measure which increased savings
Total				**6.0%**

3.5 Lifetime Savings

The evaluators calculated applicant and evaluated lifetime savings values using the following formula:

$$\text{Lifetime Savings kWh} = \text{Annual Savings kWh} * \text{Measure Lifetime Years}$$

The evaluated lifetime savings are greater than the tracking lifetime savings because the evaluated first-year savings are greater than the tracking first-year savings. The evaluated realization rate from the sampled measures is applied to the tracking savings for the application to calculate the evaluated application savings. Table 3-3 provides a summary of key factors that influence the lifetime savings for application 13839869.

Table 3-3. Application 13839869 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	2,947,806	2,947,806	3,123,266
First year savings	586,045	586,045	620,928

Measure lifetime*	5.03 years	5.03 years	5.03 years
Baseline classification	Retrofit	Retrofit	Retrofit

**Weighted average of all measure lifetimes included in application*

Table 3-4 provides a summary of key factors that influence lifetime savings for M1.

Table 3-4. Measure M1 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	7,460	7,460	8,043
First year savings	1,492	1,492	1,609
Measure lifetime	5 years	5 years	5 years
Baseline classification	Retrofit	Retrofit	Retrofit

Table 3-5 provides a summary of key factors that influence lifetime savings for M2.

Table 3-5. Measure M2 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	80,280	80,280	28,891
First year savings	16,056	16,056	5,778
Measure lifetime	5 years	5 years	5 years
Baseline classification	Retrofit	Retrofit	Retrofit

Table 3-6 provides a summary of key factors that influence lifetime savings for M3.

Table 3-6. Measure M3 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	64,535	64,535	0
First year savings	12,907	12,907	0
Measure lifetime	5 years	5 years	5 years
Baseline classification	Retrofit	Retrofit	Retrofit

Table 3-7 provides a summary of key factors that influence lifetime savings for M4.

Table 3-7. Measure M4 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	216,645	216,645	64,150
First year savings	43,329	43,329	12,830
Measure lifetime	5 years	5 years	5 years
Baseline classification	Retrofit	Retrofit	Retrofit

Table 3-8 provides a summary of key factors that influence lifetime savings for M5.

Table 3-8. Measure M5 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	479,535	479,535	398,598
First year savings	95,907	95,907	79,720
Measure lifetime	5 years	5 years	5 years
Baseline classification	Retrofit	Retrofit	Retrofit

Table 3-9 provides a summary of key factors that influence lifetime savings for M6.

Table 3-9. Measure M6 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	633,115	633,115	1,070,073
First year savings	126,623	126,623	214,015
Measure lifetime	5 years	5 years	5 years
Baseline classification	Retrofit	Retrofit	Retrofit

3.5.1 Ancillary impacts

There are gas savings associated with evaluated measures M1 and M5. The gas savings associated with M1 are not impacted by evaluation findings. The impact on gas savings for M5 is not expected to be significant because the realization rate for the measure is 83%.

RI CUSTOM ELECTRIC EVALUATION SITE-SPECIFIC REPORT

DNV SITE ID: RICE22S068

Report Date: 3/7/2024

Application ID(s)	12449900	
Project Type	C&I Retrofit	
Program Year	2022	
Evaluation Firm	DMI	
Evaluation Engineer	Bennett Rose	
Senior Engineer	Mickey Bush	

1 EVALUATED SITE SUMMARY AND RESULTS

This site is a fast food restaurant that is open 24/7. The site consists of a kitchen area with multiple grills and frying vats, a dining area, and a play area for children. The energy savings measures installed were as follows:

M1 Demand Control Kitchen Ventilation - Install sensors on grill and two fryer vat hoods to monitor cooking activity and modulate associated exhaust and makeup airflow accordingly. There are three separate exhaust fans; one per hood. New exhaust fans have EC fan motors and were installed outside of the scope of the project. Savings are provided by reduced average exhaust fan speed. The evaluator found that the implemented controls do no communicate with the RTUs so no ventilation cooling savings were evaluated for this measure.

M2 HVAC Rooftop Unit Blower Motor VFD (Catalyst) - This measure considers the installation of an add-on controller and VFD to modify the 5 constant volume RTUs serving the building to function as variable air volume. The evaluator found that this measure was installed, but the variable speed controls have been disabled due to comfort issues experienced with the modified RTU controls.

M3 Walk-In Refrigeration Controls and Fan Motor Retrofit - Savings were claimed for retrofitting two (2) evaporator fan motors in the cooler with EC motors and adding anti-sweat door heater controls. Savings result from reduced fan power and reduced anti-sweat heater runtime. These reductions also decrease the cooling load (fan heat and heater output) on the refrigeration system. The evaluator found these measures were implemented as expected.

Savings claimed for retrofitting three (3) evaporator fan motors in freezer with EC motors and adding anti-sweat door heater controls. The evaluator found that the freezer does not have an antisweat door heaters but that the evaporator fan motors were retrofitted.

The evaluation results are presented in Table 1-1.

Table 1-1. Evaluation Results Summary

PA Application ID	Measure Name		Annual Electric Energy (kWh)	% of Energy Savings On-Peak	Summer On-Peak Demand (kW)	Winter On-Peak Demand (kW)
12449900	M1-Demand Control Kitchen Ventilation	Tracked	4,229	46.0%	1.16	0.50
		Evaluated - ops	2,774	46.0%	0.29	0.25
		Realization Rate	65.6%	100.0%	24.9%	49.4%
	M2-HVAC Rooftop Unit Blower Motor VFD	Tracked	53,354	46.0%	1.16	0.85
		Evaluated - ops	0	0.0%	0.00	0.00
		Realization Rate	0.0%	0.0%	0.0%	0.0%
	M3-Walk-in Refrigeration Controls	Tracked	4,519	46.0%	0.43	0.43
		Evaluated - ops	28,985	46.0%	4.21	3.28
		Realization Rate	641.4%	100.0%	978.2%	230.0%
Total		**Tracked**	**62,102**	**46.0%**	**2.75**	**1.78**
		Evaluated - ops	**31,758**	**46.0%**	**4.50**	**3.53**
		Realization Rate	**51.1%**	**100.0%**	**163.5%**	**127.1%**

N/A = Not applicable

1.1 Explanation of Deviations from Tracking

The evaluated savings are 49% less than the applicant-reported savings. The reduction in savings is primarily due to the evaluation finding that the variable speed RTU controls have been disabled by the site. This measure accounted for 86% of the savings associated with the application.

1.2 Recommendations for Program Designers & Implementers

The site disabled the variable speed controls for the RTUs due to comfort issues reported shortly after the installation of the measure. Catalyst RTU controllers have been installed as an energy efficiency measure for many years at many sites. It may be useful to investigate if this specific measure has poor persistence generally or if this is a site-specific finding with no broader relevance to the energy efficiency program.

1.3 Customer Alert

None.

2 EVALUATED MEASURES

The following sections present the evaluation procedure, including the findings from an in-depth review of the supplied applicant calculations and the evaluation methodology determined to be the best fit for the site and the information available.

2.1 Application Information and Applicant Savings Methodology

This section describes the applicant's application information, savings methodology, and the evaluation assessment of the savings calculation algorithm used by the applicant.

2.2 Applicant Description of Baseline

The applicant measure event type is retrofit. The baseline considered by the applicant is the existing systems and controls. These are as follows:

M1 – Recently installed kitchen hood exhaust fans with EC motors operating at constant speed.

M2 – Five RTUs serving the building providing constant volume airflow and operating continuously.

M3 – Cooler evaporator fans with PSC motors and no cycling controls for the cooler door heater. Freezer evaporator fans with PSC motors and no cycling controls for the freezer door heater.

The baseline values used in the applicant savings analysis are presented in Table 2-1.

Table 2-1. Applicant baseline key parameters

Measure	Parameter	BASELINE		
		Value(s)	Source of Parameter Value	Note
M1	Exhaust Fan power	0.23 kW (grill) 0.19 kW (french fryer) 0.19 kW (protein fryer)	Applicant assumption, source unclear	
	Exhaust Fan Operating Hours	8,736 hours	Applicant assumption	
	RTU Cooling Performance	0.923 kW/ton (same for all 5 RTUs)	Applicant assumption	
	Minimum Outside Air Percentage	27.5% (same for all 5 RTUs)	Applicant assumption	
M2	RTU Fan Power	RTU-D1: 1.8 kW RTU-D2: 1.8 kW	Nameplate HP	

		RTU-K1: 3.0 kW RTU-K2: 0.9 kW RTU-P1: 1.8 kW		
	RTU Operating Hours	8,736 hours	Applicant assumption	
M3	Cooler Evaporator kW	0.075 kW	1/10-HP converted to kW	
	Cooler Evaporator Run Hours	8,760 hours	Applicant assumption	
	Cooler Door Heater kW	0.23 kW	Applicant assumption	
	Cooler Door Heater Run Hours	8,760 hours	Applicant assumption	
	Freezer Evaporator kW	0.075 kW	1/10-HP converted to kW	
	Freezer Evaporator Run Hours	8,760 hours	Applicant assumption	
	Freezer Door Heater kW	0.23 kW	Applicant assumption	
	Freezer Door Heater Run Hours	8,760 hours	Applicant assumption	

2.2.1 Applicant Description of Installed Equipment and Operation

The applicant assumes the following retrofit measures were implemented:

M1 – Variable exhaust controller and sensors on the kitchen hoods are installed to monitor cooking activity by sensing the temperature of the exhaust air and modulate exhaust accordingly. The applicant assumed that makeup air is reduced as part of this measure.

M2 – A catalyst controller and VFD are installed on all five RTUs serving the building. The catalyst controller monitors outside air temperature and modulates the RTU supply fan speed according to a reset schedule designed to mimic a VAV system.

M3 – Cooler evaporator fans are upgraded to EC motors and cycling controls are installed for the cooler door heater and cycling is controlled based on temperature and RH readings to prevent the door seal from freezing shut.

Freezer evaporator fans are upgraded to EC motors and cycling controls are installed for the freezer door heater and cycling is controlled based on temperature and RH readings to prevent the door seal from freezing shut.

The proposed system values used in the applicant savings analysis are presented in Table 2-2.

Table 2-2: Application proposed key parameters

Measure	Parameter	PROPOSED		
		Value(s)	Source of Parameter Value	Note
M1	Exhaust Fan power	0.11 kW (grill) 0.09 kW (french fryer) 0.09 kW (protein fryer)	Baseline kW @ 74.66% speed	
	Exhaust Fan Operating Hours	8,736 hours	Applicant assumption	
	RTU Cooling Performance	0.923 kW/ton (same for all 5 RTUs)	Applicant assumption	
	Minimum Outside Air Percentage	21% (same for all 5 RTUs)	Applicant assumption	
M2	RTU Average Fan Power	RTU-D1: 0.6 kW RTU-D2: 0.6 kW RTU-K1: 1.0 kW RTU-K2: 0.3 kW RTU-P1: 0.6 kW	Calculated using airflow bin model and fan affinity law	
	RTU Operating Hours	8,736 hours	Applicant assumption	
M3	Cooler Evaporator kW	0.026 kW	TRM	
	Cooler Evaporator Run Hours	8,760 hours	Applicant assumption	
	Cooler Door Heater kW	0.06 kW	Assumes 74% reduction in runtime	Applicant models cycling as a reduction in average kW.
	Cooler Door Heater Run Hours	8,760 hours	Applicant assumption	
	Freezer Evaporator kW	0.039 kW	TRM	
	Freezer Evaporator Run Hours	8,760 hours	Applicant assumption	
	Freezer Door Heater kW	0.06 kW	Assumes 74% reduction in runtime	
	Freezer Door Heater Run Hours	8,760 hours	Applicant assumption	

2.2.2 Applicant Energy Savings Algorithm

The applicant uses a calculation spreadsheet designed to calculate a specific set of measures on a large scale for multiple sites that references nameplate data for some equipment assumptions, TRM savings calculations, and some generic non-site-specific load assumptions.

M1 Demand Control Kitchen Ventilation

The savings for this measure are calculated using a bin model spreadsheet.

Reduce Outside Air – Savings are calculated using a 1°F OAT temperature bin model. Savings consider difference in cooling load due to change in minimum OA%. The formulas are as follows.

$$Mixed\ Air\ Temperature = OA\% * OAT + (1 - OA\%) * RAT$$

$$Mixed\ Air\ Enthalpy = OA\% * OAH\ + (1 - OA\%) * RAH$$

where:

RAT = 73°F

RAH = assumes 73°F and humidity of saturated air at cooling coil

$OA\%$ = 27.5% (Baseline) / 21% (Proposed) both with dry bulb economizing

Cooling load is calculated using the following formula comparing mixed air enthalpy/temperature to supply enthalpy/temperature assuming that each RTU provides 100% design airflow 24/7.

$$Cooling\ Load\frac{kBTU}{hr} = \frac{CFM * MAX\big(4.5 * (MAH - SAH), 1.08 * (MAT - SAT)\big)}{1{,}000}$$

Mechanical cooling savings are calculated using the following formula. Savings result from cooler mixed air temperature/lower mixed air enthalpy during the cooling season resulting from reduced outside airflow percentage.

$$Cooling\ kW = DX\frac{kW}{Ton} * \frac{Cooling\ Load\ kBTU}{12\frac{kBTU}{ton}}$$

Table 2-3 summarizes the results of this analysis approach.

Table 2-3. RTU cooling savings for M1

RTU	RTU Design CFM	DX kW/Ton	Baseline kWh	Proposed kWh	Savings kWh
RTU-P1	4,000	0.923	15,086	14,762	323
RTU-K1	5,000	0.923	18,857	18,453	404
RTU-D2	3,000	0.923	11,314	11,071	242
RTU-D1	4,000	0.923	15,086	14,762	323
RTU-K2	2,000	0.923	7,543	7,381	162
Total			67,884	66,429	1,455

The applicant calculated exhaust fan savings using the same oneline calculation for three fans. Existing fan power kW is a value entered in the spreadsheet and its source is unclear.

$$Proposed\ Fan\ Power = Existing\ Fan\ Power\ kW * 74.66\%^{2.5}$$

where:

74.66% = Average expected fan speed with variable speed control installed.

$Existing\ Fan\ Power$ = 0.23 kW (grill), 0.19 kW (french fryer), 0.19 kW (protein fryer)

Annual fan savings are calculated using the following formula.

$$Energy\ Savings\ kWh = (Existing\ Fan\ Power\ kW - Proposed\ Fan\ Power\ kW) * 8{,}736\ hours$$

Exhaust fan savings = 2,773.7 kWh

Total Savings = 4,229 kWh

M2 HVAC Rooftop Unit Blower Motor VFD (Catalyst)

Applicant savings are calculated using a 1°F OAT temperature bin model.

Baseline fan power is calculated using nameplate fan HP, 70% load and using ASHRAE 90.1 value for the fan type power adjustment associated with forward-curved, damper control system.

$$Baseline\ Fan\ Power\ BHP = Fan\ HP * 70\%\ Load\ Factor * 100.07\%\ Fan\ Type\ Adjustment\ Factor$$

Proposed case VAV flow schedule is assumed to be 90% flow at 95°F, 50% flow at 55°F and 90% flow at 5 °F varying linearly between these temperatures ("V" shaped airflow profile).

$$Proposed\ Case\ Fan\ Power\ \%\ BHP\ =\ A\ +\ B\ *\ \%\ Flow\ +\ C\ *\ \%\ Flow^2$$

Coefficients for this equation are sourced from AHRAE 90.1 and are associated with Variable Speed Drive control:

$$A = 0.219762$$

$$B = -0.874784$$

$$C = 1.652597$$

Fan kW is calculated by converting BHP to kW and applying the NEMA premium efficiency motor rating based on the nominal HP of the fan motor assuming the motor is 1800 rpm.

Energy savings are calculated assuming 8,760 annual operating hours for all of the RTUs.

Table 2-4 summarizes the results of this analysis approach.

Table 2-4. RTU Savings for M3

RTU	Fan HP	Baseline kWh	Proposed kWh	Savings kWh
RTU-P1	3	15,694	5,409	10,285
RTU-K1	5	26,157	9,016	17,142
RTU-D2	3	15,694	5,409	10,285
RTU-D1	3	15,694	5,409	10,285
RTU-K2	1.5	8,174	2,817	5,357
Total		81,415	28,061	53,354

M3 Walk-In Refrigeration Controls

The savings for this measure are calculated using one line calculations from the TRM.

$$Existing\ Motor\ kW\ =\ 0.05\frac{HP}{motor} * 0.746\frac{kW}{HP} * 2\ Motors\ =\ 0.075\ kW$$

$$Annual\ Fan\ Savings\ =\ 0.075\ kW\ *\ 65\%\ demand\ reduction\ *\ 8{,}760\ hours\ =\ 424.6\ kWh$$

$$Annual\ Refrigeration\ Load\ Savings\ = Annual\ Fan\ Savings * \frac{1.6\ kW}{ton}\ =\ 190.2\ kWh$$

Cooler anti sweat heater savings are calculated using the following formula.

$$(230\ V\ *\ 1\ amp\ /\ 1000)\ *\ 0.74\ \ *\ 8{,}760\ hours\ =\ 1{,}491\ kWh$$

The savings for the freezer control upgrades are calculated using one line calculations from the TRM.

$$Existing\ Motor\ kW\ =\ 0.05\ HP/motor\ *\ 0.746\ kW/HP\ *\ 3\ Motors\ =\ 0.112\ kW$$

$$Annual\ Savings\ =\ 0.112\ kW\ *\ 65\%\ demand\ reduction\ *\ 8{,}760\ hours\ =\ 636.9\ kWh$$

$$Annual\ Refrigeration\ Load\ Savings\ = Annual\ Fan\ Savings\ * \frac{3.412\frac{kBTU}{hr}}{\frac{kWh}{12\frac{kBTU}{ton}}} * 1.6\frac{kW}{ton}\ =\ 285.3\ kWh$$

Freezer anti sweat heater savings are calculated using the following formula from the TRM.

$$\left(\frac{230V * 1\ amp}{1{,}000}\right)\ *\ 0.74\ *\ 8760\ hours\ =\ 1{,}491\ kWh$$

2.2.3 Evaluation Assessment of Applicant Methodology

M1 – It appears that the applicant's assumption for average exhaust fan speed with variable speed controls is informed by a previous project. The ventilation cooling savings are calculated assuming that the RTUs will operate with a different minimum outside airflow percentage. Based on the MRD the applicant assumes that the controller will communicate with the RTU economizer module and adjust OA damper position in response to exhaust fan speed.

M2 – The catalyst RTU controls saving calculation is consistent with the methodology used in the National Grid RTU optimizer calculation tool which is a vetted tool used for the same measure type with a reasonable analysis approach referencing ASHRAE 90.1 for performance assumptions.

M3 – The applicant calculates fan savings and door heater savings using TRM calculations which is reasonable. The evaluator found that the refrigeration contractor noted two details that were misrepresented in the applicant analysis. The refrigerator contractor installed 1/15 HP EC motors, but the applicant analysis considered 1/10 HP motors which decreases savings for the evaporator fan retrofits. The refrigeration contractor also noted that there is no freezer door heater at this site however the applicant claims savings for freezer door heater cycling controls.

2.3 On-site Inspection and Metering

This section provides details on the work performed during the on-site inspection.

2.3.1 Summary of Site Visit

This section summarizes the site visit.

- The evaluator visited the site on August 24, 2023.
- The evaluator installed kW meters on the 3-phase 208V feeds for RTU-D1, RTU-D2, RTU-K1, RTU-K2, and RTU-P1. The metered loads include supply fan, compressor, and auxiliary RTU power.
- The evaluator installed amperage meters on the 240V split pole circuit that includes the cooler and freezer evaporator fans as well as the cooler door heater. The evaluator found that the freezer does not have a door heater or door heater controls. The freezer door is inside of the cooler so humidity is low and condensate building up and freezing in the door seal is not a concern.
- The evaluator installed amperage meters on the single phase 120V circuits serving the Grill, Fryer Vat, and the Protein Vat. The kitchen equipment is wired in series with its associated exhaust fan as a safety measure so that if the exhaust fan does not turn on, the kitchen equipment will not have power. The meters were installed in the electrical panel. Each exhaust fan has a switch box located on the roof however the evaluator found that the switch box was not big enough to install metering equipment so metering in the panel was the only feasible option.
- The evaluator was informed that the catalyst RTU controls have been disabled because the site experienced comfort issues during warm weather following the implementation of the catalyst controls. The evaluator found that the catalyst controllers are still installed as well as the associated VFDs. Based on visual inspection it is not clear if the VFDs are set to a fixed speed, or are bypassed and abandoned in place. The metered data shows that the fans are running at nameplate BHP indicating that the VFDs are likely bypassed.
- The evaluator returned to the site on January 18, 2023 to retrieve the meters.

- The project engineer for the TA vendor who processed the application confirmed that the kitchen hood exhaust controls are not connected to the RTUs to reduce make-up air when the exhaust fans run less.
- Trend data is not available for this site.

Table 2-5. Measure Verification

Measure Name	Verification Method	Verification Result
M1 – Demand Control Kitchen Ventilation	Metering and onsite verification	Evaluator confirmed that controls have been installed and control the exhaust fans but do not communicate with the RTUs to control make up air flows.
M2 – HVAC Rooftop Unit Blower Motor VFD (Catalyst)	Metering and onsite verification	Metering and site feedback confirm that catalyst controls were installed but have been disabled by the site due to comfort complaints after implementation.
M3 – Walk-In Refrigeration Controls	Metering and onsite verification, refrigeration vendor invoices	Evaluator confirmed EC motor retrofit and door heater controls for cooler. The freezer does not have door heater.

2.3.2 Measured and Logged Data

Table 2-6 summarizes the metered data collected at this site by the evaluator.

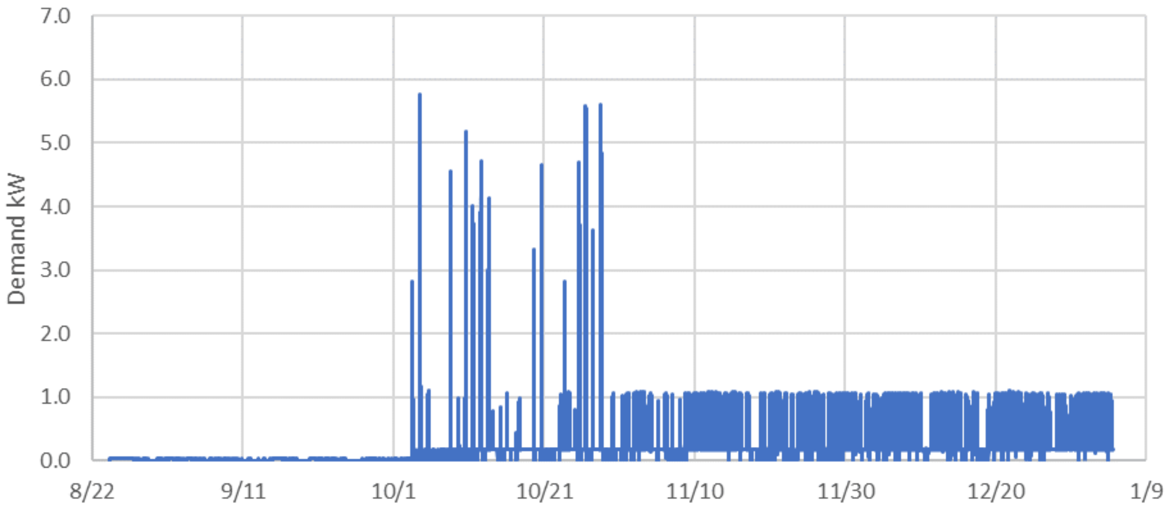
Table 2-6. Metered Data Summary

Equipment	Meter	Metering Period
RTU kW Data (RTU D1, D2, K1, K2, P1)	eGauge 15004	133 Days (8/24/2023-1/4/2024)
Grill, fryer vat and protein vat circuit amperage (kitchen equipment and dedicated exhaust fan amps combined)	HOBO UX120	107 Days (8/24/2023-12/9/2023)
Evaporator circuit amps (includes cooler and freezer evaporator fans and cooler door heater)	HOBO UX120	133 Days (8/24/2023-1/4/2024)

All five RTUs are served from one electrical panel and one three-phase kW meter was used to monitor all five RTUs separately. The electric meter installed used a single voltage connection and separate amperage channels for each RTU feed.

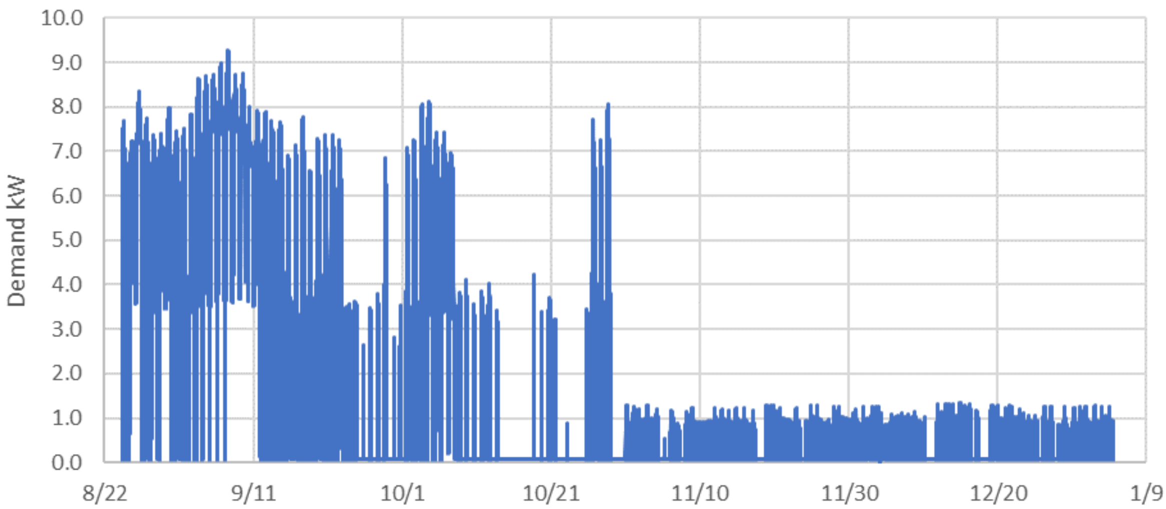
The metered kW data for RTU-D1 is presented in Figure 2-1.

Figure 2-1. RTU-D1 Raw kW Data



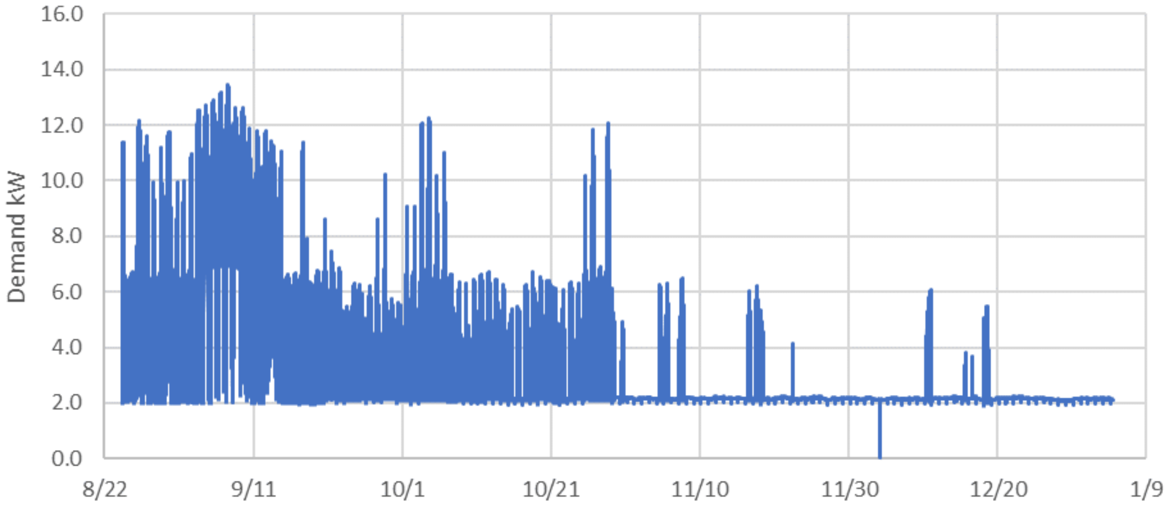
The metered kW data for RTU-D2 is presented in Figure 2-2.

Figure 2-2. RTU-D2 Raw kW Data



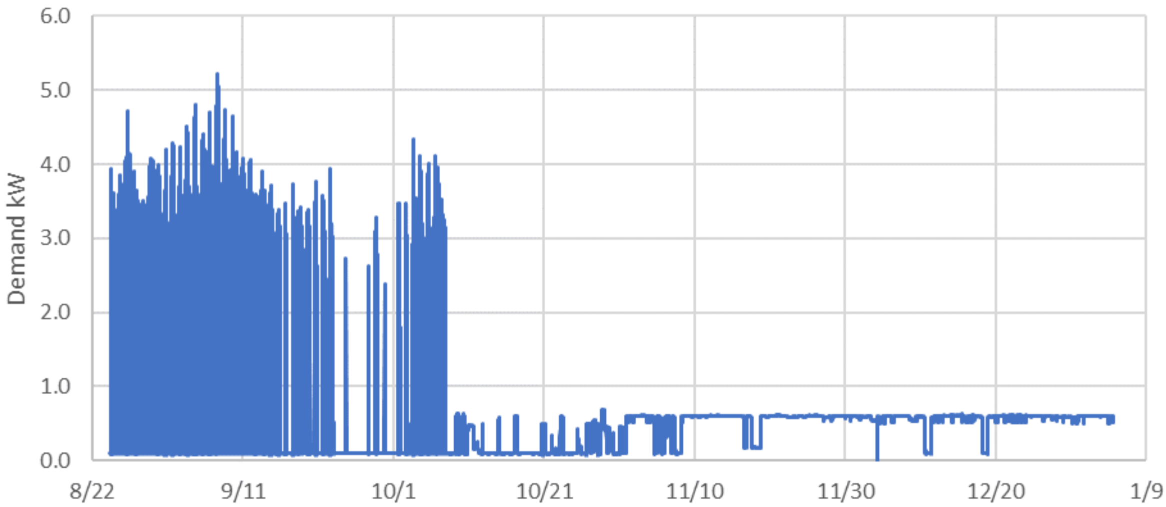
The metered kW data for RTU-K1 is presented in Figure 2-3.

Figure 2-3. RTU-K1 Raw kW Data



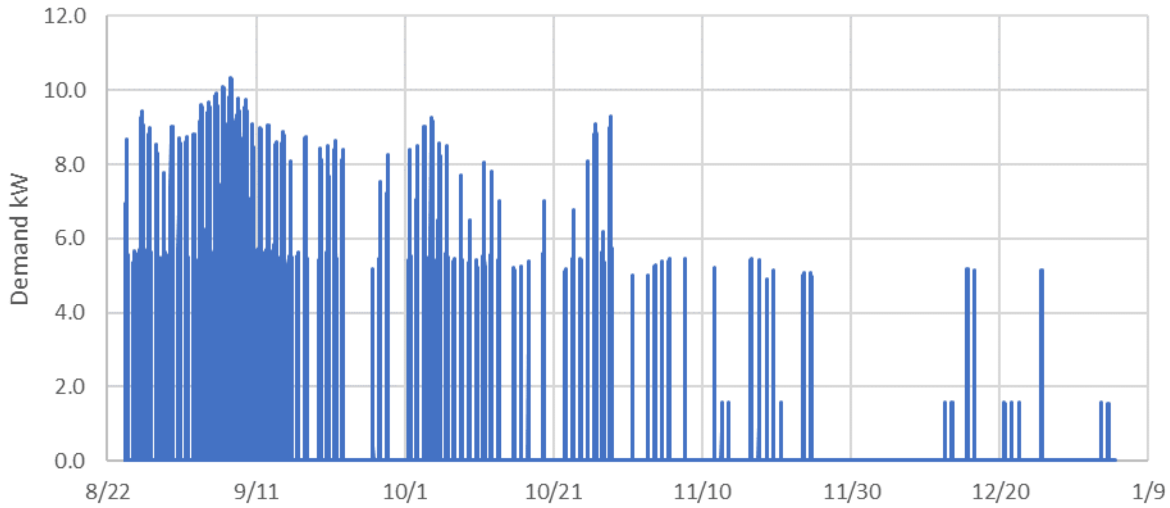
The metered kW data for RTU-K2 is presented in Figure 2-4.

Figure 2-4. RTU-K2 Raw kW Data



The metered kW data for RTU-P1 is presented in Figure 2-5.

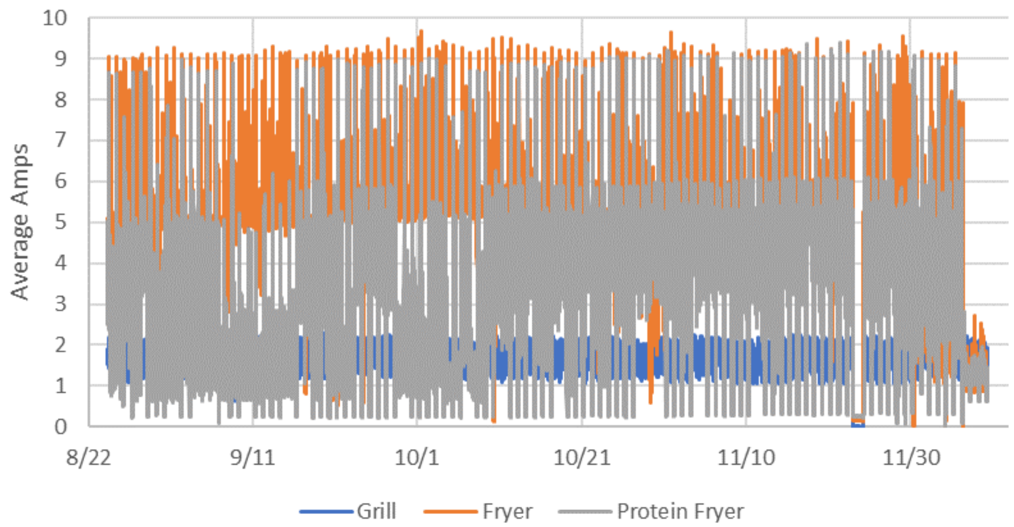
Figure 2-5. RTU-P1 Raw kW Data



The kitchen cooking equipment is on the same circuit as the associated dedicated exhaust fan. The exhaust fan is wired in series with the kitchen equipment as a hard-wired safety to ensure that ventilation is available for the kitchen equipment to operate, i.e. if the exhaust fan fails, the circuit is not complete and the kitchen equipment does not have power.

The metered amperage data for the three kitchen equipment circuits is presented in Figure 2-6.

Figure 2-6. Kitchen Equipment Circuit Amperage Data



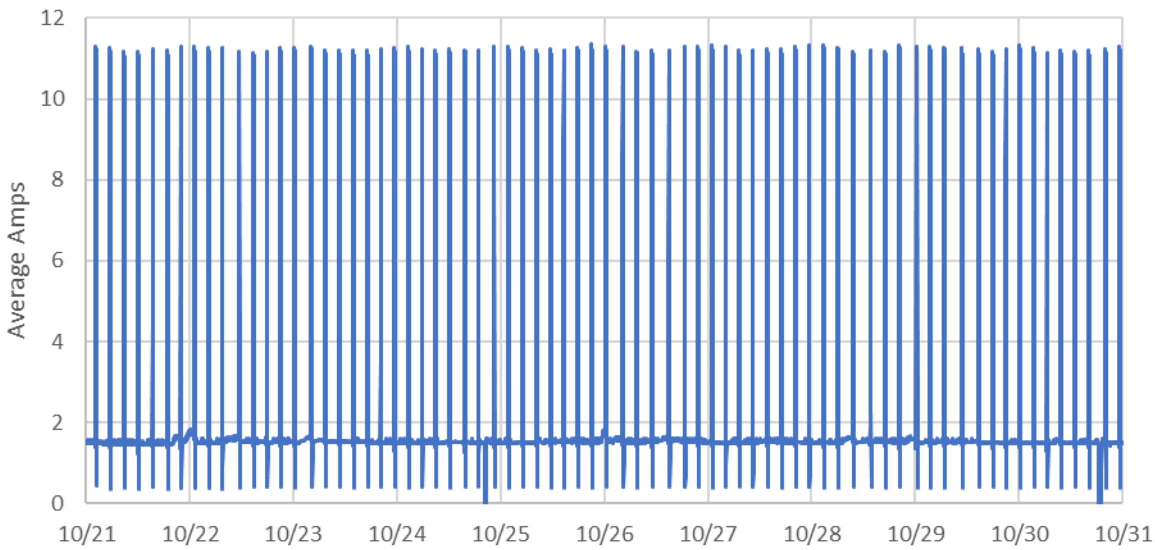
The evaluator confirmed that the controller for variable speed exhaust fan control is installed and controlling exhaust fan speed. The controller is presented in Figure 2-7.

Figure 2-7. Kitchen Exhaust Fan Controller

The evaluator and electrician identified the evaporator circuit that is fed from the refrigeration system condensing unit on the roof. The circuit is a 240V split pole circuit that feeds the cooler evaporator, freezer evaporator, and cooler door heater. The evaluator found that the freezer door is located inside the cooler and does not have a door heater because relative humidity inside the cooler is low, i.e. the freezer door freezing shut is not a concern.

A sample period of the metered amperage data for the 240V split phase circuit serving the cooler evaporator, freezer evaporator and cooler door heater is presented in Figure 2-8. The amperage profile shows cooler door heater cycling on and off at ~11 amps and consistent combined evaporator fan amperage draw at ~1.5 amps.

Figure 2-8. Cooler/Freezer Evaporator and Door Heater Amperage Data



The evaluator confirmed that the cooler and freezer controllers are installed. The controller is presented in Figure 2-9.

Figure 2-9. Walk-in Controllers

Trend data is not available for this site.

2.4 Evaluation Methods and Findings

This section describes the evaluator methods and findings.

2.4.1 Evaluation Description of Baseline

The evaluator classifies the measure event type for each of the measures as add-on. The evaluator generally agrees with the applicant baselines.

M1 Demand Control Kitchen Ventilation

The evaluator agrees that the existing case baseline is appropriate for the kitchen hood exhaust fans operating at constant volume prior to the installation of variable speed controls.

M2 HVAC Rooftop Unit Blower Motor VFD (Catalyst)

The evaluator observed the installation of VFDs and catalyst controllers in each of the five RTUs. In the absence of VFDs the RTU supply fans would operate at fixed speed and constant volume, as assumed.

M3 Walk-In Refrigeration Controls

The evaluator considers the existing case baseline of PSC cooler evaporator fans and cooler door heaters without controls.

The evaluator considers the existing case baseline of PSC freezer evaporator fans and found that the freezer door is located inside of the cooler and does not have a door heater.

2.4.2 Evaluation Calculation Method

Exhaust Fan Variable Speed Controls

The evaluator collected metered amperage data for the three kitchen equipment circuits with dedicated exhaust. Each circuit includes the amperage draw of both the piece of cooking equipment and the associated exhaust fan. The evaluator confirmed that the controller for variable speed exhaust has been installed. Figure 2-7 demonstrates that the controller is modulating exhaust fan speeds as EF-1 and EF-3 are both shown to be at less than 100% speed during the site visit.

Figure 2-10 compares the applicant's assumptions for average installed case exhaust fan power to the load profile of the grill circuit that includes the grill exhaust fan calculated for the single phase 120V circuit using the evaluators metered amperage data.

Figure 2-10. Evaluated Grill + Exhaust kW vs Applicant Grill Exhaust kW

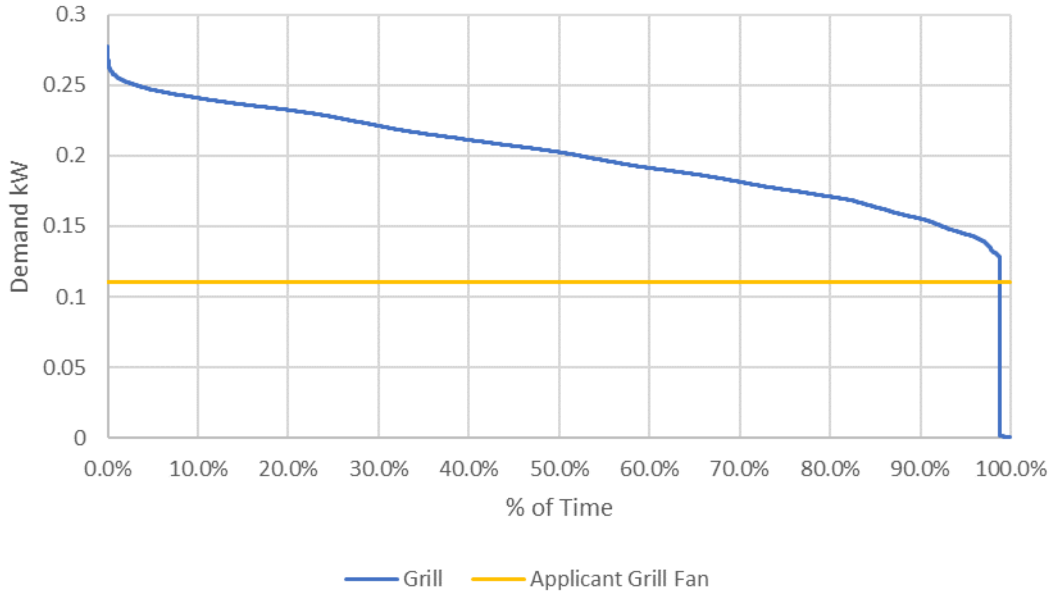


Figure 2-11 compares the applicant's assumptions for average installed case exhaust fan power to the load profile of the fryer vat circuit that includes the fryer vat exhaust fan calculated for the single phase 120V circuit using the evaluators metered amperage data.

Figure 2-11. Evaluated Fryer + Exhaust kW vs Applicant Fryer Exhaust kW

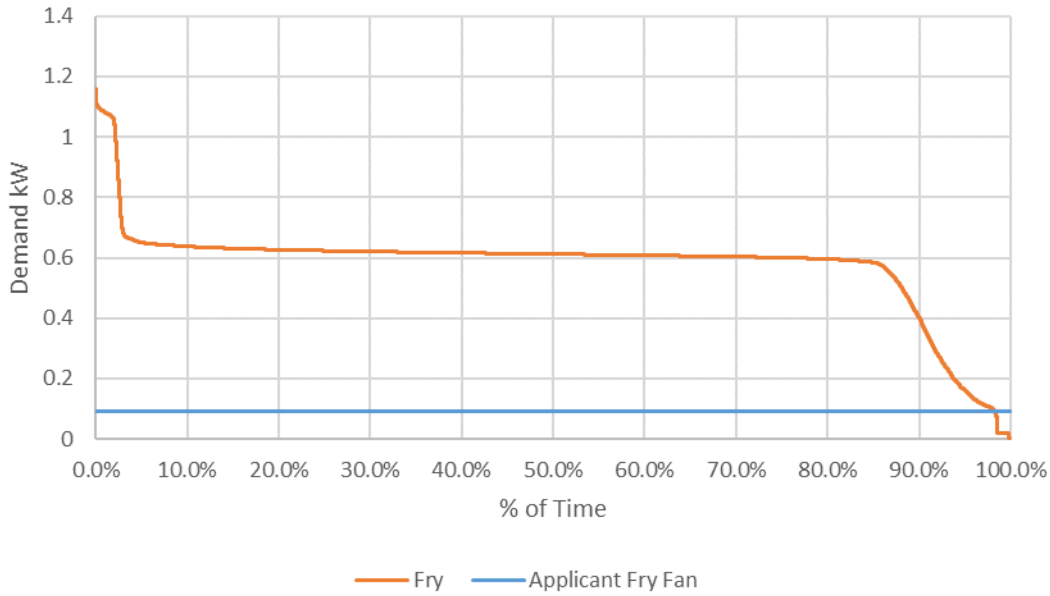
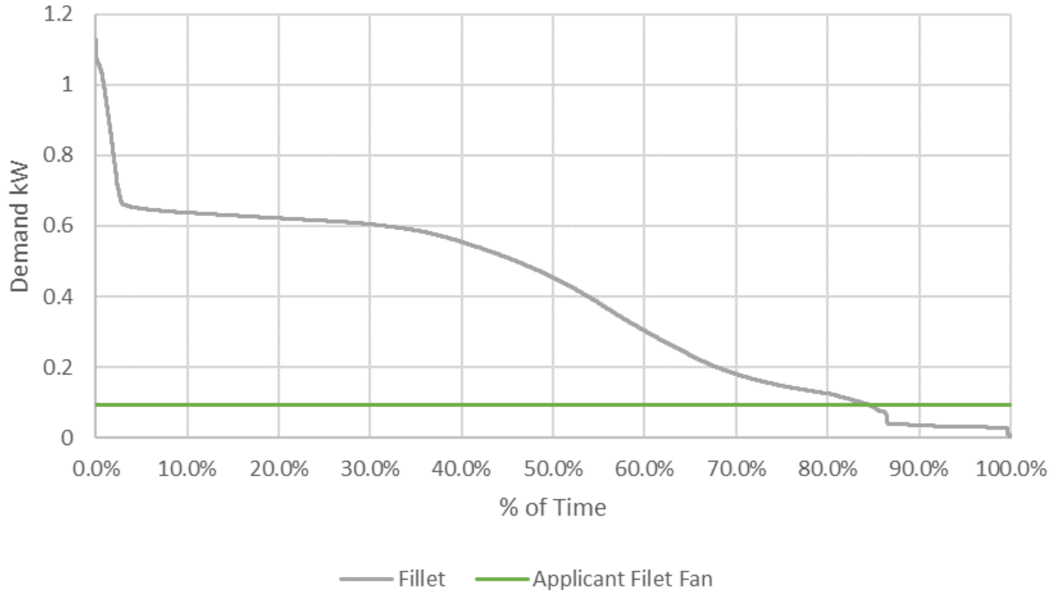


Figure 2-12 compares the applicant's assumptions for average installed case exhaust fan power to the load profile of the protein fryer vat circuit that includes the protein fryer vat exhaust fan calculated for the single phase 120V circuit using the evaluators metered amperage data.

Figure 2-12. Evaluated Protein Fryer + Exhaust kW vs Applicant Protein Fryer Exhaust kW



The evaluator is not able to confirm average exhaust fan power using the metered data. The evaluator is able to verify that the measure has been implemented and the metered data indicates that the applicant's estimate for average installed case fan power is reasonable because the total metered kW (Kitchen equipment and fan) is greater than the applicant fan kW. Based on this finding, the evaluator makes no changes to the applicant's exhaust fan savings calculations.

The project engineer who processed the applicant confirmed with the kitchen hood controls vendor that the variable speed exhaust controller does not communicate to the RTUs. No makeup air cooling savings are considered by the evaluator.

Roof Top Units

The site contact stated that catalyst controls had been installed at multiple fast food restaurant locations and shortly after had been disabled. The site contact could not confirm definitively if the evaluation site was one of the sites that had the controller disabled. By reviewing the RTU operating profile based on the evaluator's metered data the evaluator was able to confirm that the variable fan speed controls have been disabled at this site.

The RTUs serving the dining area; RTU-D1 and RTU-D2, cycle on as required to satisfy the space heating and cooling loads. Figures 2-13 and 2-14 demonstrate that during the heating season when the kW is just for the fans, the units have a consistent electrical demand indicating that fan speed does not vary with outside air temperature. This finding indicates that the units are not used to provide continuous ventilation and that variable fan speed controls are not being used for these units.

Figure 2-13 demonstrates the average demand when the unit is operating and the percentage of time that the unit runs as a function of outside air temperature for RTU-D1.

Figure 2-13. RTU-D1 kW and Runtime % vs Outside Air Temperature

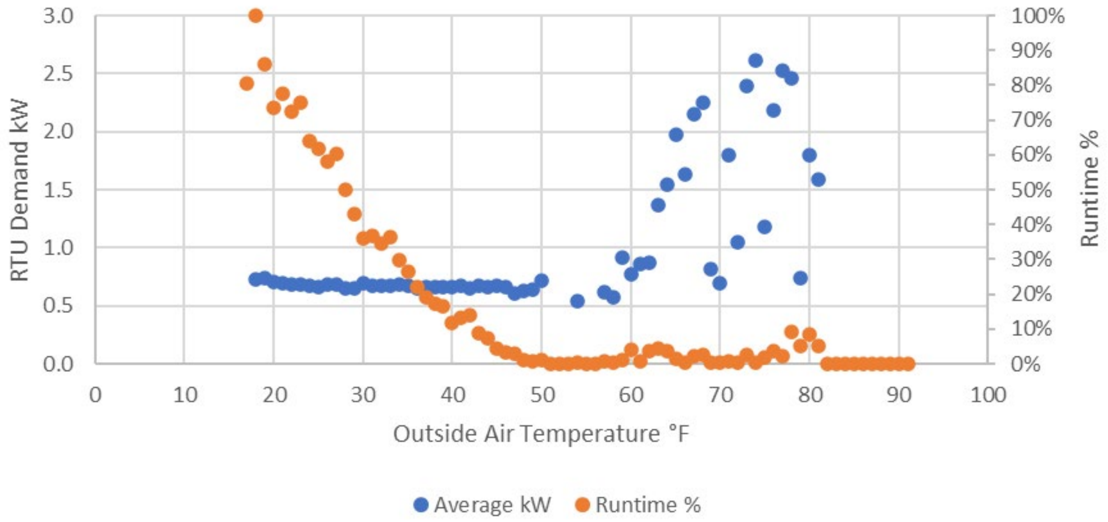
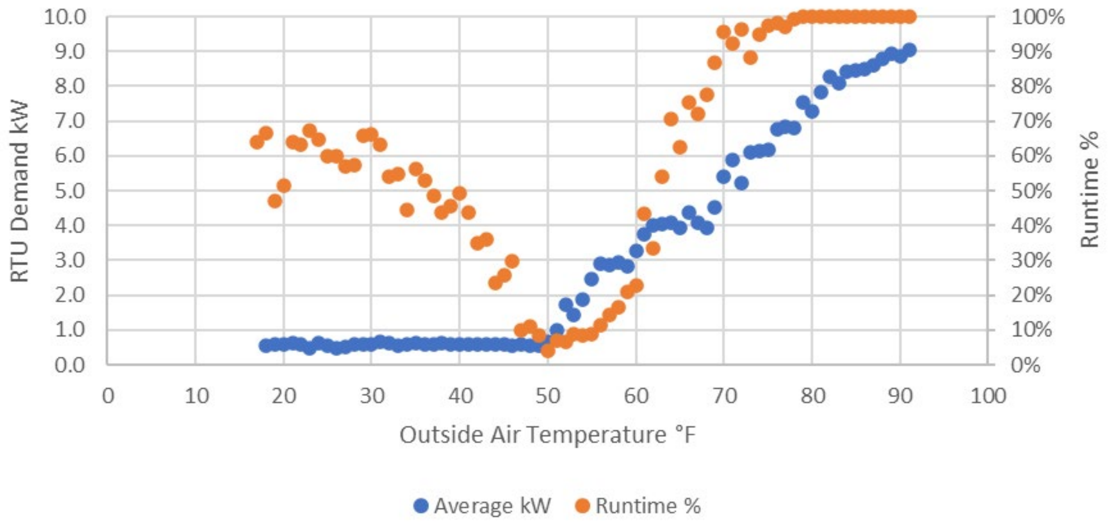


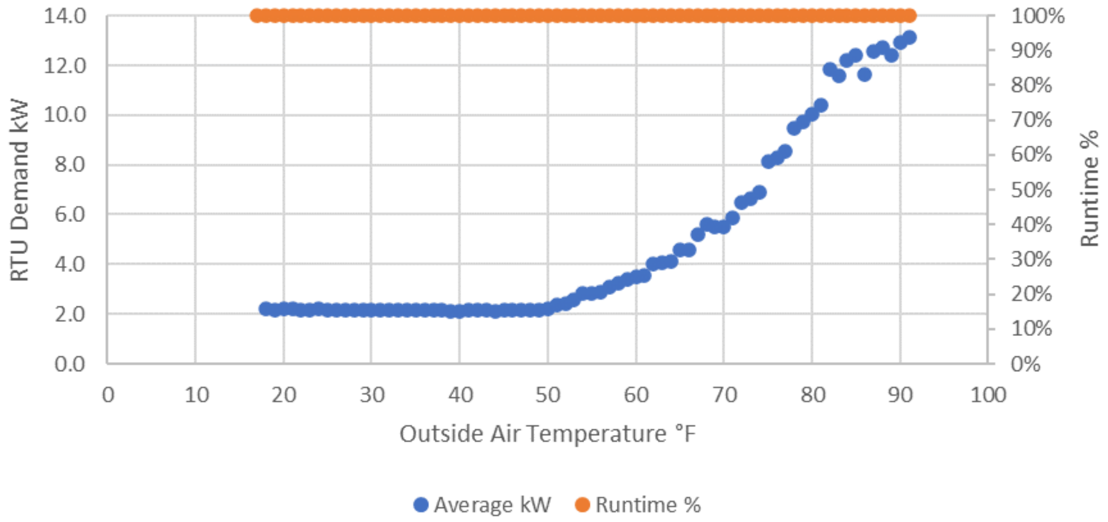
Figure 2-14 demonstrates the average demand when the unit is operating and the percentage of time that the unit runs as a function of outside air temperature for RTU-D2.

Figure 2-14. RTU-D2 kW and Runtime % vs Outside Air Temperature



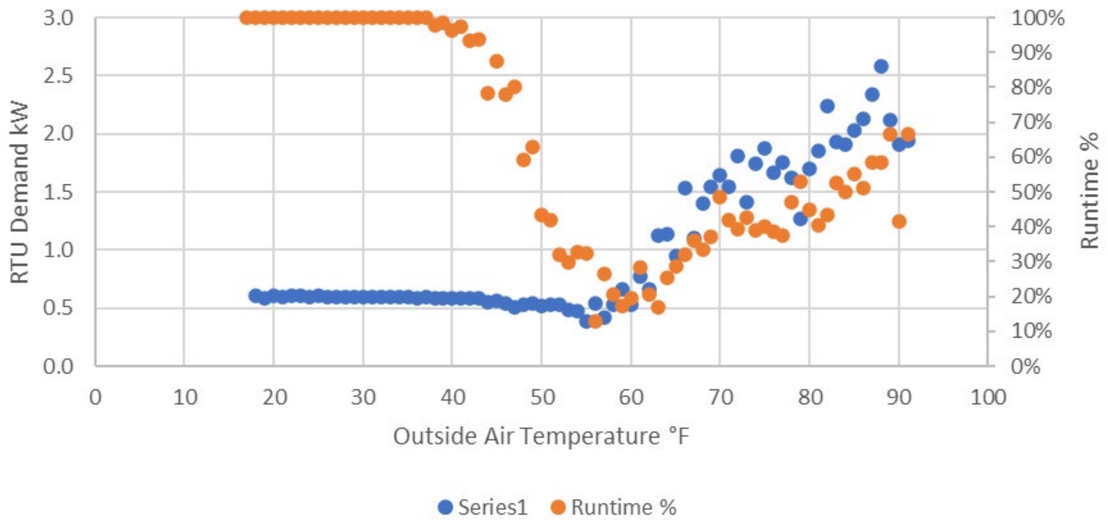
RTU-K1 serves the kitchen and operates continuously during the metering period. This demonstrates that RTU-K1 is used to provide ventilation to the kitchen. The demand profile for the cooler months when no cooling is required and the kW is just for the fans, demonstrates that the fan power does not vary and that variable speed fan controls have been disabled for the unit. Figure 2-15 demonstrates the relationship between RTU demand and the percentage of time that the unit runs as a function of outside air temperature for RTU-K1. The controls vendor confirmed that the kitchen exhaust controller does not communicate the RTUs.

Figure 2-15. RTU-K1 kW and Runtime % vs Outside Air Temperature



RTU-K2 also serves the kitchen. Unlike RTU-K1 it cycles and does not operate continuously indicating that it is controlled to cycle on and off as required to serve space heating and cooling loads. The demand profile demonstrates that the fan power does not vary and that variable speed fan controls have been disabled for the unit. Figure 2-16 demonstrates the relationship between RTU demand when the unit is operating and the percentage of time that the unit runs as a function of outside air temperature for RTU-K2.

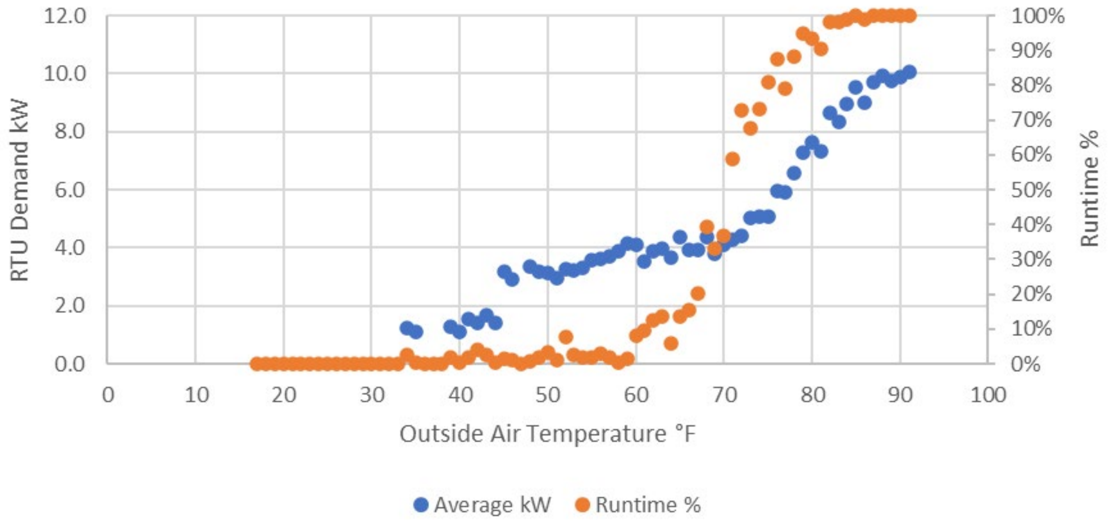
Figure 2-16. RTU-K2 kW and Runtime % vs Outside Air Temperature



RTU-P1 serves the play area. The play area has a large south facing glazing area. The metered data shows that RTU-P1 cycles to serve the cooling load and that there is very little need for the unit to provide heating as RTU-P1 does not cycle on at low outside air temperatures.

Figure 2-17 demonstrates the relationship between RTU demand when the unit is operating and the percentage of time that the unit runs as a function of outside air temperature for RTU-P1.

Figure 2-17. RTU-P1 kW and Runtime % vs Outside Air Temperature

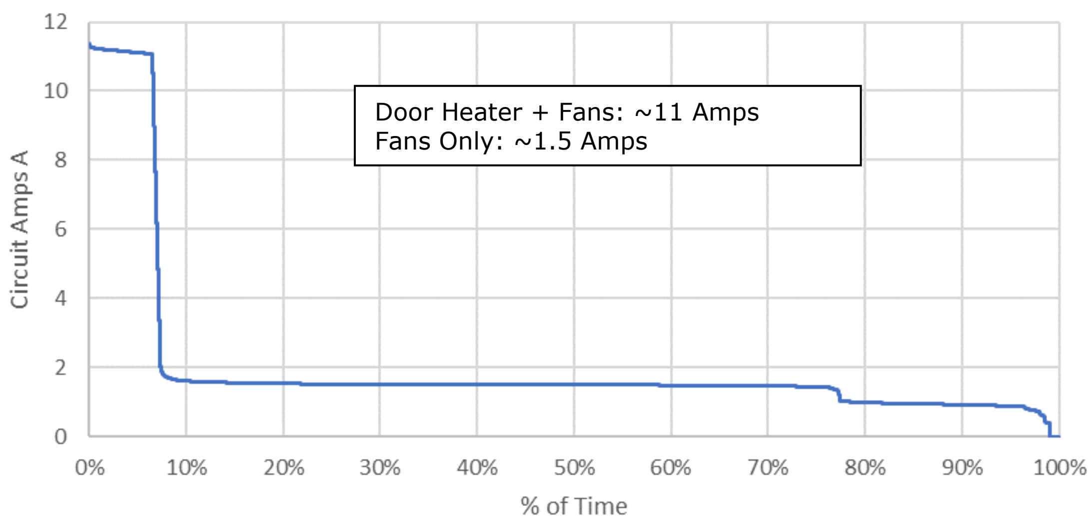


The metered data shows that the RTUs serving the Dining area and Play area RTUs cycle to serve heating and cooling loads. The applicant analysis assumes that these units operate continuously. The fact that they are used only for space conditioning and not for ventilation indicates that they are not considered for the ventilation load reductions considered for the variable kitchen hood ventilation controls. The data shows that variable speed fan controls are not implemented and confirms that the catalyst controllers are not being used so no savings are evaluated for these RTUs.

Refrigeration

The evaluator confirmed that five evaporator fans have been retrofit with 1/15 HP EC motors and that door heater controls have been installed for the cooler door. The evaluator metered the amperage on the split phase 240V circuit that provides power for the cooler door heater, the cooler evaporator fans and the freezer evaporator fans. This amperage data is presented in the load shape duration curve in Figure 2-18 and demonstrates the distinct amperage amplitudes associated with the loads on the circuit.

Figure 2-18. Evaporator Circuit Amps



The metered data shows two demand levels associated with evaporator fan operation, one at ~1.5 amps, and one level at ~1 amp. It is not known if the decrease in amperage draw from 1.5 A to 1 A is due to one of the evaporator fan sets cycling off or if the fan speeds were manually manipulated. The evaluator confirmed on the cooler and evaporator controller interfaces that automatic evaporator fan controls are not being implemented. The time period at the lower amperage level is 8/24-9/21. This period includes the first site visit and all evaporator fans were operating at the time of the install. Without more granular metered data it is assumed that average evaporator fan amperage includes both ranges of operation.

The evaluator calculates operating demand assuming a power factor of 1 for the door heater electric resistance heating element and 0.85 for the evaporator fans. Table 2-7 shows the evaluated savings for the door heater and all (cooler and freezer) evaporator fans.

Table 2-7. Refrigeration installed case refrigeration system analysis

Load	Avg. Operating Amps	Runtime %	Power Factor	Calculated kW	Installed Annual Energy kWh Usage
Fans and Door Heater	10.6	7.4%	-	-	-
Evap Fans Only (all fans)	1.4	99.1%	0.85	0.3	2,457
Door Heater Only (calc)	9.2	7.4%	1	2.2	1,430

The evaluator uses the same TRM methodology as the applicant to calculate EC motor retrofit savings, but using installed kW as the starting point for the calculation and applying the TRM savings factor of 65% to calculated baseline fan power. The evaluated energy savings for total evaporator fan savings, i.e. for cooler and freezer evaporators combined, is presented in Table 2-8 assuming 8,760 annual run hours.

Table 2-8. Evaluator evaporator fan savings

Parameter	Avg. Operating Amps	Calculated kW	Annual Energy kWh	Savings Factor	Baseline		Savings	
					kW	kWh	kW	kWh
Evap Fans Only	1.4	0.3	2,457	65%	0.8	7,019	0.5	4,563

Evaporator fan savings are attributed to two cooler evaporator motors and three freezer evaporator motors proportionally based on motor count. Refrigeration load savings are calculated assuming the same 1.6 kW/ton performance assumed by the applicant. The savings for M3 are presented in Table 2-9.

Table 2-9. Evaluator evaporator fan and refrigeration load savings

Parameter	Cooler Evaporator			Freezer Evaporator		
	Fan	Refrigeration	Total	Fan	Refrigeration	Total
Savings kWh	1,825	830	2,655	2,738	1,245	3,983
Savings kW	0.21	0.10	0.31	0.32	0.14	0.46

The evaluator calculates baseline door heater energy assuming that the heater operates continuously without cycling controls. Demand savings for the door heater cycling controls assume that the installed case runtime is evenly distributed across each hour of the day and each day of the week. The evaluated energy savings for door heater cycling controls is presented in Table 2-10 and apply runtime percentage to 8,760 total annual operating hours.

Table 2-10. Door heater savings summary

Parameter	Avg. Amps	Installed Case			Baseline			Savings	
		Runtime %	Calc. kW	Annual Energy kWh	Runtime %	Avg. kW	Annual Energy kWh	Demand kW	Heater Energy kWh
Door Heater Only (Calc)	9.2	7.4%	2.2	1,430	100%	2.2	19,404	2.1	17,974

The evaluation savings also consider the cooling savings associated with the reduction in door heater energy. It is assumed that half of the door heater energy is a load on the refrigeration system and half is a cooling load in the space. Savings are calculated assuming refrigeration system performance of 1.6 KW/ton and DX cooling performance of 0.91 kW/ton. The door heater savings are assumed to be evenly distributed year-round and DX cooling savings associated with door heater cycling controls are only considered for the 1,053 hours per year that the outside air temperature is greater than 75°F.

Table 2-11. Door Heater Cooling Load Savings

Cooling Load Savings										
Heater Energy Savings kWh	Refrigeration			DX Cooling				Savings		
	Heat to Refrig.	Ton-hrs	kW/ton	Heat to DX	Annual Cooling Hours	DX Ton-hrs	DX kW/ton	Annual Energy kWh	Summer kW	Winter kW
17,974	50%	2,555	1.6	50%	1,053	307	0.92	4,372	1.39	0.47

3 FINAL RESULTS

This section summarizes the evaluation results determined in the analysis above. This section includes a summary table of savings by major end-use and application.

Table 3-1. Summary of Key Parameters

Measure	Parameter	BASELINE		PROPOSED / INSTALLED	
		Tracking Value(s)	Evaluation Value(s)	Tracking Value(s)	Evaluation Value(s)
M1	Exhaust Fan Power	0.23 kW (grill) 0.19 kW (french fryer) 0.19 kW (protein fryer)	0.23 kW (grill) 0.19 kW (french fryer) 0.19 kW (protein fryer)	0.11 kW (grill) 0.09 kW (french fryer) 0.09 kW (protein fryer)	0.11 kW (grill) 0.09 kW (french fryer) 0.09 kW (protein fryer)
M1	Average RTU Min OA	27.5% (all RTUs)	27.5% (all RTUs)	21% (all RTUs)	27.5% (all RTUs)
M2	RTU Control Method	Constant Volume	Constant Volume	VFD retrofit for VAV operation	Constant Volume
M3	Cooler Door Heater Amperage	1 Amp	9.2 Amps	1 Amp	9.2 Amps
M3	Cooler Door Heater Runtime	100%	100%	26%	7.4%
M3	Annual Door Heater Cooling Load	Refrigeration: 286 Ton-hr DX: 34 Ton-hr	Refrigeration: 2,759 Ton-hr DX: 332 Ton-hr	Refrigeration: 74 Ton-hr DX: 9 Ton-hr	Refrigeration: 203 Ton-hr DX: 24 Ton-hr
M3	Evaporator Fan kW	0.075 kW	0.32 kW	0.026 kW	0.11 kW
M3	Freezer Door Heater Amperage	1 Amp	0 amps (no door heater)	1 Amp	0 amps (no door heater)
M3	Freezer Door Heater Runtime	100%	0% (no door heater)	26%	0% (no door heater)
M3	Evaporator Fan kW	0.112 kW	0.49 kW	0.039 kW	0.17 kW

3.4 Explanation of Differences

This section describes the key drivers behind any difference in the application and evaluation estimates of annual kWh savings. The following table summarizes these differences. The purpose of this table is to describe how changes to the key parameters influenced the final project savings through the end-use summary analysis. Table 3-2 provides a summary of the differences between tracking and evaluated values.

Table 3-2. Summary of Deviations

Measure	Discrepancy	Parameter	Impact of Deviation	Discussion of Deviations
M1	Operating Load	Kitchen Ventilation Cooling Savings	-2.1%	**Decreased savings –** The evaluator confirmed that the RTUs do not communicate with the variable kitchen exhaust controller so RTU cooling savings were not evaluated for M1
M2	Controls	Catalyst Controllers Disabled	-75.9%	**Decreased savings –** Variable speed fan controllers have been disabled and RTUs are operating as constant volume. Fan savings are not being achieved for this measure.
M3	Operating Load	Cooler Door Heater Amperage	23.4%	**Increased savings –** Evaluator updated door heater savings calculation based on metered door heater amperage and runtime which increased savings.
M3	Measure Interactivity	Cooler Door Heater Cooling Load Savings	0.5%	**Increased savings –** Cooling load savings associated with decreasing cooler door heater energy are considered by the evaluator which increases energy savings.
M3	Operating Load	Evaporator Fan Amperage	7.3%	**Increased savings –** Evaluator updated evaporator fan savings calculation based on metered fan amperage which increased savings.
M3	Baseline	No Freezer Door Heater	-2.1%	**Decreased savings –** The freezer does not have a door heater so there are no evaluated savings for freezer door heater controls.
Total				**-48.9%**

3.5 Lifetime Savings

The evaluators calculated applicant and evaluated lifetime savings values using the following formula:

$$\text{Lifetime Savings kWh} = \text{Annual Savings kWh} * \text{Measure Lifetime Years}$$

The evaluated lifetime savings for M1 are less than the tracking lifetime savings because the evaluated first-year savings are smaller than the tracking first-year savings. Table 3-3 provides a summary of key factors that influence lifetime savings for M1. The evaluator assumes that the tracking lifetime savings match the lifetime savings from the BCR.

Table 3-3. Measure M1 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	42,285 kWh	42,285 kWh	27,737 kWh
First year savings	4,228 kWh	4,228 kWh	2,774 kWh
Measure lifetime	10 years	10 years (project BCR)	10 years (TRM)
Baseline classification	Retrofit	Retrofit	Retrofit

The evaluated lifetime savings for M2 are less than the tracking lifetime savings because the evaluated first-year savings are smaller than the tracking first-year savings. Table 3-4 provides a summary of key factors that influence lifetime savings for M2. The evaluator assumes that the tracking lifetime savings match the lifetime savings from the BCR.

Table 3-4. Measure M2 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	533,538 kWh	533,538 kWh	0 kWh
First year savings	53,354 kWh	53,354 kWh	0 kWh
Measure lifetime	10 years	10 years (project BCR)	10 years (TRM)
Baseline classification	Retrofit	Retrofit	Retrofit

The evaluated lifetime savings for M3 are greater than the tracking lifetime savings because the evaluated first-year savings are greater than the tracking first-year savings. Table 3-5 provides a summary of key factors that influence lifetime savings for M3. The evaluator assumes that the tracking lifetime savings match the lifetime savings from the BCR.

Table 3-5. Measure M3 - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	63,265 kWh	67,784 kWh	289,846 kWh
First year savings	4,519 kWh	4,519 kWh	28,985 kWh
Measure lifetime	14 years	15 years (project BCR)	10 years (TRM)
Baseline classification	Retrofit	Retrofit	Retrofit

The evaluated lifetime savings for application 12449900 (all measures) are less than the tracking lifetime savings because the evaluated first-year savings are smaller than the tracking first-year savings. Table 3-6 provides a summary of key factors that influence lifetime savings for all measures combined.

Table 3-6. Application 12449900 (all Measures) - Lifetime Savings Summary

Factor	Tracking	Application	Evaluator
Lifetime savings	639,088 kWh	643,607 kWh	317,583 kWh
First year savings	62,101 kWh	62,101 kWh	31,758 kWh
Measure lifetime	10.3 Years	10.4 Years	10.0 Years
Baseline classification	Retrofit	Retrofit	Retrofit

3.5.1 Ancillary impacts

This measure includes gas savings associated with reducing makeup air with variable speed kitchen ventilation controls. The evaluation finding that the variable speed kitchen exhaust controller does not communicate to the RTUs indicates that there are likely no gas savings associated with the project. If implementing variable exhaust changes the building pressurization such that infiltration decreases there may be some gas savings however these savings would be significantly less than claimed for the project.

RHODE ISLAND CUSTOM ELECTRIC SITE-SPECIFIC REPORT
SITE ID: RICE22N070

Report Date: March 25, 2024

Program Administrator	Rhode Island Energy	
Application ID(s)	13741512, 13710972, 13249513, 13475450	
Project Type	C&I Existing Building Retrofit	
Evaluation Type	Full M&V	
Program Year	PY2022	
Evaluation Firm	DNV	
Evaluation Engineer	Joe St. John	
Senior Engineer	Olav Hegland	

1 EVALUATED SITE SUMMARY AND RESULTS

The evaluated project was implemented at a supermarket through the GrocerSmart initiative and consisted of the measures shown in Table 1-1.

Table 1-1. Measure list

Identifier	Project ID	kWh savings	% of total project savings	Measure description
1	13741512	145,459	77.7%	Anti-sweat heater controls of low and medium temp cases. 472 linear feet total.
2	13710972	17,482	9.3%	Install 54 linear feet of doors on coolers that had no doors previously.
3	13249513	14,908	8.0%	Clean condenser coil on 16 tons of low-temp and 39 tons of medium-temp refrigeration systems in a grocery store.
4		5,337	2.9%	Clean evaporator and condenser coils on 107.5 tons of packaged RTU equipment.
5		171	0.09%	Clean condenser coil on 0.80 tons of low-temp and 0.86 tons of medium-temp refrigeration systems in a grocery store.
6	13475450	3,840	2.1%	Install gaskets on 61 doors on low-temperature reach-in freezers, and gaskets on 35 doors on medium-temperature freezers.
Total		**187,197**	**100%**	

During the initial interview with the site contact, evaluators learned the following:

- The site contact is present on-site and agreed to accommodate an on-site evaluation.
- It is safe to visit the facility and inspect the measure.

Based on the information gathered during the initial interview with the site contact, the evaluator proposed this site be evaluated using on-site verification with full M&V, where an on-site audit was used to verify measure installation and operation and install loggers to capture the key parameters for the largest measure, the anti-sweat heater controls to ascertain any necessary operational adjustments. For the remaining, non-anti-sweat heater measures, on-site verification consisted of visual inspection, interviewing the site-contact and review of the calculations. The evaluation results are presented in Table 1-2.

Table 1-2. Evaluation results summary

PA application ID	Measure name		Annual electric energy (kWh)	% of energy savings on-peak	Summer on-peak demand (kW)	Winter on-peak demand (kW)
13741512	Anti-sweat heater controls	Tracked	145,459	46.0%	13.04	21.04
		Evaluated	124,245	45.8%	9.06	19.79
		Realization Rate	85.4%	99.6%	69.5%	94.0%
13710972	Install 54 linear feet of doors	Tracked	17,482	41.0%	5.57	3.52
		Evaluated	17,461	41.0%	5.57	3.52
		Realization Rate	99.9%	100.0%	100.0%	100.0%
13249513	Clean condenser and evaporator coils	Tracked	20,416	60.5%	5.75	0.83
		Evaluated	9,761	53.2%	2.86	0.99
		Realization Rate	47.8%	100.0%	49.7%	119.3%
13475450	Install door gaskets on 61 reach-in freezer doors	Tracked	3,840	100.0%	0.44	0.44
		Evaluated	3,840	100.0%	0.44	0.44
		Realization Rate	100.0%	100.0%	100.0%	100.0%
Total		Tracked	187,197	48.2%	24.80	25.83
		Evaluated	155,306	47.1%	17.93	24.73
		Realization Rate	83.0%	97.6%	72.3%	95.8%

1.1 Explanation of deviations from tracking

This section provides an overview of the evaluation deviations from the tracking savings for each of the measures. Further details regarding deviations from the tracked savings are presented in Section 3.1.

13741512 / Anti-sweat heater controls

The evaluation savings for the anti-sweat heater measure were 15% lower than the tracking energy savings, primarily due to the evaluation finding that the wattage per door for the freezers was 202.6 Watts, rather than 240.9 Watts.

13710972 / Install 54 linear feet of doors

The evaluated savings for the installation of 54′ of doors measure was 0.1% lower than the tracking energy savings because of a slight discrepancy between the observed case temperatures, and the case temperatures used in the tracking calculations.

13249513 / Clean condenser and evaporator coils

The evaluation savings for this measure, which involved cleaning condenser and evaporator coils, was 47.8% of the tracked savings because the evaluator found two studies collected empirical data on the impact that cleaning coils has on the efficiency of space-conditioning equipment. The average of those studies found that the efficiency improvement was about 1.48%, while the tracking calculations estimated that the efficiency improvement would be about 2.9%. In addition, in several of the calculations the tracking calculations used an 88%, 99%, and 100% diversity factor for the refrigeration system, whereas the evaluator used a diversity factor of 85%. The diversity factor for HVAC equipment was 26% in the tracking calculations and was unchanged by the evaluator.

13475450 / Install door gaskets on 61 reach-in freezer doors

For this measure, there were no differences between the tracking savings and the evaluation savings.

1.2 Recommendations for program designers and implementers

The evaluators have three recommendations for program designers and implementers.

1. Consider updating the savings estimates for the condenser and evaporator coil cleaning measure.
2. Consider updating the measure life for the door gasket measure from 5 years to 1 year.
3. Consider investigating enhanced grocery-store programs, around continuous maintenance and/or natural refrigerants.
4. Consider investigating typical diversity factors of grocery store refrigeration systems.

Consider updating condenser and evaporator coil cleaning savings estimates.

The evaluators recommend that program designers consider updating the savings estimates for the measure for cleaning condenser and evaporator coils. Based on the review of the two available studies[1] that collected empirical data on the impact of these two measures, the evaluators believe the impact may be about 0% to 51% of the claimed 3% energy savings for this measure. One of the studies cited, ASHRAE RP-1705, found that cleaning the condenser coils produces no significant effect on performance. The CPUC study found approximately 4.7% savings for cleaning condenser coils, and 0.15% savings for cleaning evaporator coils. The savings estimates used in the tracking estimates are built on a non-empirically backed estimate that cleaning the condenser coils would restore a condenser's cooling capacity from a degraded value (caused by fouling) of 85% back up to 100%. The ASHRAE project found that cleaning coils had no significant effect on a unit's cooling capacity. As quoted by the author in a summary of the report:[2]

> *"This project yielded an astonishing result that initially none of us believed. It showed that fouling can increase the heat capacity of a condenser, even though it reduced the airflow rate. In fact, our small sample of coils, on average, there was no improvement in heat capacity from cleaning the coils. I doubt anybody reading this will believe that this result could be correct, but we repeated our experiments several times, and exhaustively switched out sensors before we finally started to understand how these results could happen… We believe that the fouling material, which almost always all collects on the upstream face of the coil, increases turbulence on the coil's heat transfer surfaces, improving the convective heat transfer, even though airflow is reduced (just like fin enhancements do)."*

Consider updating measure life for door gasket measure.

[1] "Investigation of Airside Fouling on Condenser Heat Exchangers (RP-1705)". ASHRAE 1705

[2] "Impact Evaluation of 2013-13 Commercial Quality Maintenance Programs (HVAC3)" Prepared for the California Public Utilities Commission, 2016, DNV. CALMAC Study ID CPU117.

The evaluators recommend updating the measure life from 5 years to 1 year, based on the documentation on the Regional Technical Forum (RTF) presentation[3] where this adjustment was made to this measure in efficiency programs in the Pacific Northwest that are under the RTF's jurisdiction. While the energy savings used for this measure cite the RTF documentation, the measure life for this measure does not cite the RTF documentation. The reason cited by the RTF for the 1-year measure life is based on an ADM report[4] on door gaskets completed for the CPUC. This report found:

- A high replacement rate [of door gaskets] even without maintenance contracts (12 out of 71 had contracts)
- For stores without maintenance contracts, 69% replaced gaskets more frequently than every 3 years.
- For stores with maintenance contracts, half maintain gaskets at least annually, and 64% replace them within 2 years.
- We do not know how much more frequently the gaskets will be checked by program participants.
- Measure life is the average difference between when the gasket is replaced and when it would otherwise have been replaced.

<u>Consider grocery store programs around continuous maintenance and/or natural refrigerants (recommendation from site contact).</u>

The site contact for this facility – the refrigeration engineer for this store as well as several others that are part of the same grocery store chain at nearby locations – stated that he would like to see additional programs that address refrigeration system continual maintenance, or potentially around incentives for switching to natural refrigerants that have a lower global warming potential (GWP). While a natural refrigerant program may not significantly impact energy use, and may not fit into RI statewide policy framework, it is something that some program administrators[5] in other jurisdictions have begun to explore.

<u>Consider investigating typical diversity factors of grocery store refrigeration systems</u>

For the coil cleaning measures that took place on grocery store refrigeration system, the tracking calculations used diversity factors of 88%, 99%, and 100% on the refrigeration equipment, and 26% on the HVAC equipment that had the evaporator coils cleaned. The original tool recommends a value of 85%, but no source is cited. During the evaluation, the evaluators attempted to collect data on typical diversity factors, but no reliable data could be found. The evaluators thus used 85% for this evaluation but recommend that if this measure is expected to become common, that additional investigation be performed to inform this estimate.

1.3 Customer alert

There are no relevant customer alerts for this site.

[3] 20190618DoorGasketPres.pptx | Powered by Box
[4] Microsoft Word - ComFac Evaluation V3 HIM Appendices _02-18-2010_.doc (calmac.org)
[5] SMUD Launches Natural Refrigerant Incentive Program — North American Sustainable Refrigeration Council (nasrc.org)

2 EVALUATED MEASURES

The following sections present the evaluation procedure, including the findings from an in-depth review of the supplied applicant calculations and the evaluation methodology determined to be the best fit for the site and the information available.

The evaluated measures for this site are summarized in Table 2-1.

Table 2-1. Evaluated measures

Measure	Project ID	kWh savings	% of total project savings	Parameter
1	13741512	145,459	77.7%	Anti-sweat heater controls of low and medium temp cases. 472 linear feet total.
2	13710972	17,482	9.3%	Install 54 linear feet of doors.
3		14,908	8.0%	Clean condenser coil on 16 tons of low-temp and 39 tons of medium-temp refrigeration system in grocery store.
4	13249513	5,337	2.9%	Clean evaporator and condenser coils on 107.5 tons of packaged RTU equipment.
5		171	0.09%	Clean condenser coil on 0.80 tons of low-temp and 0.86 tons of medium-temp refrigeration system in grocery store.
6	13475450	3,840	2.1%	Install gaskets on 61 doors on low-temperature reach-in freezers, and gaskets on 35 doors on medium-temperature freezers.
Total		**187,197**	**100%**	

2.1 Application information and applicant savings methodology

This section describes the savings methodology, and the evaluation assessment of the savings calculation algorithms used by the applicant.

2.1.1 Applicant energy savings algorithm and applicant key parameters

13741512 / Anti-sweat heater controls

The applicant used eQUEST modeling software to calculate the savings, following the GrocerSmart program guidelines. In addition, the applicant used the on-site findings to determine a variety of building inputs, including refrigeration system type and efficiency, complex building geometry, lighting systems, and HVAC systems to estimate the energy savings.

The refrigeration specs for the eQUEST model were generated using a proprietary audit tool that maps on-site data collection to a proprietary database. The database is a collection of refrigeration equipment data and specification sheets that have been modified such that the database outputs are compatible with eQUEST parameter keywords. The database outputs shape the eQUEST model to match observed equipment specifications (e.g., number of refrigeration fixtures, case heat conduction rate, case lighting power, suction groups) that were collected through the on-site audit.

However, to be able to compare the key parameters between the tracking calculations and the evaluator findings, the evaluator developed an Excel spreadsheet with simplified parameters that reproduced the eQUEST results. That simplified calculation is listed below.

The low-temp freezers used the following formula:

$$kWh\ Savings\ = 146\ Doors\ \times 2\frac{Amps}{Door} \times 120\ Volts\ \times 1.0\ P.F. \times (8{,}760\ Hours\ Baseline - 4{,}730\ Hours\ Post)$$

= 141,723.82 kWh

The medium temp freezers use the following formula:

$$kWh\ Savings\ = 12\ Doors\ \times 0.40\ \frac{Amps}{Door} \times 120\ Volts\ \times 1.0\ P.F. \times (8{,}760\ Hours\ Baseline - 2{,}277.6\ Hours\ Post)$$

= 3,735 kWh

13710972 / Install 54 linear feet of doors

The kWh savings per foot of door shown above was deduced from the calculation sheet. The calculations were based on e modeling software.

The addition of doors to reach-in coolers is composed of 17,482 kWh of energy savings.

This measure uses the following simplified formula to estimate savings:

$$kWh\ Savings\ = 18\ Doors\ \times 3\ \frac{Ft}{Door} \times 323.75\ \frac{\text{kWh Saved}}{\text{ft}}\ = 17{,}482\ \text{kWh}$$

13249513 / Clean condenser and evaporator coils

Clean Condenser Coils

There are three calculation sheets that all use the same template which describes cleaning condenser coils at this grocery store. The first spreadsheet shows how 16 tons of low-temperature and 39 tons of medium-temperature refrigeration condenser coils were cleaned, which resulted in an energy savings claim of 14,908 kWh of savings.

The low-temperature calculations use the following (simplified) calculations. The full calculations are a more complex bin calculation where the EER changes based on the outdoor air temperature, but they reduce to the equation below with the key input variables. The EER values shown are the weighted average annual values.

$$kWh\ Savings_{LowTemp1}$$

$$= 18.2\ \text{tons operating}\ \times 88\%\ \text{Diversity Factor}\ \times \left(\frac{1}{7.83\ \text{EER}_{\text{baseline}}} - \frac{1}{7.96\ EER_{post}}\right)$$

$$\times \frac{12{,}000\,\frac{\text{Btuh}}{\text{ton}}}{1{,}000\,\frac{Watts}{kW}} \times 8{,}760\,\frac{\text{hours}}{\text{year}}\ =$$

$$kWh\ Savings_{LowTemp1} = 3{,}511\ kWh$$

$$kWh\ Savings_{LowTemp2}$$

$$= 0.86 \text{ tons design } \times 100\% \text{ Diversity Factor } \times \left(\frac{1}{9.88 \text{ EER}_{\text{baseline}}} - \frac{1}{10.07\ EER_{post}}\right) \times \frac{12{,}000 \frac{\text{Btuh}}{\text{ton}}}{1{,}000 \frac{Watts}{kW}}$$

$$\times 8{,}760 \frac{\text{hours}}{\text{year}} =$$

$$kWh\ Savings_{LowTemp2} = 171\ kWh$$

The medium-temperature calculations use the following (simplified) calculations.

The full calculations are more complex, but they reduce to the equation below.

$$kWh\ Savings_{MediumTemp}$$

$$= 39.2 \text{ tons design} \times 99\% \text{ diversity factor } \times (\frac{1}{13.11 \text{ EER}_{\text{baseline}}}$$

$$- \frac{1}{13.61\ EER_{post}}) \times \frac{12{,}000 \text{ Btuh/ton}}{1{,}000\ Watts/kW} \times 8{,}760 \frac{\text{hours}}{\text{year}} =$$

$$kWh\ Savings_{MediumTemp} = 11{,}397\ kWh$$

Clean Evaporator Coils (and condenser coils)

The savings from cleaning of the evaporator and condenser coils uses the following (simplified) calculations.

The full calculations are more complex, but they reduce to the equation below.

$$kWh\ Savings_{CleanCoils}$$

$$= 107.5 \text{ tons design } \times 26\% \text{ Diversity Factor } \times (\frac{1}{11 \text{ EER}_{\text{baseline}}}$$

$$- \frac{Ft}{11.34\ EER_{post}}) \times \frac{12{,}000 \frac{\text{Btuh}}{\text{ton}}}{1{,}000 \frac{Watts}{kW}} \times 5{,}872 \frac{\text{hours}}{\text{year}} =$$

$$kWh\ Savings_{CleanCoils} = 5{,}337\ kWh$$

13475450 / Install door gaskets on 61 reach-in freezer doors

This measure is part of the ESPO program, which has prescription values for various measures, i.e., the savings for this project are treated more like a prescriptive measure with deemed values that were developed to be uniform across all installations. The savings calculations are as follows:

$$kWh\ Savings_{Install\ Gaskets} = 49.3 \frac{\text{kWh}}{\text{Door}_{\text{lowtemp}}} \times 61 \text{ Doors}_{\text{lowtemp}} + 23.8 \frac{\text{kWh}}{\text{Door}_{\text{mediumtemp}}} \times 35 \text{ Doors}_{\text{mediumtemp}} =$$

$$kWh\ Savings_{Install\ Gaskets} = 3{,}840\ kWh$$

The 49.3 kWh savings per low-temperature door and 23.8 kWh savings per medium-temperature door cite a currently deactivated Regional Technical Forum (RTF) Unit Energy Savings (UES) workbook. The RTF oversees the energy efficiency programs in the Pacific Northwest. This measure was de-activated due to an ADM study[6] that indicated a high replacement rate even without maintenance contracts. One slide stated that the average gasket fails halfway through the three-year period of maintenance, and the program could accelerate the replacement by approximately a year and a half, rounded down to a year to account for the stores that do maintenance more frequently. The measure life recommended was one year.

The original RTF savings calculations state that the average savings per door are 69 kWh per low-temp door and 39 kWh per medium-temp reach-in door. However, these calculations were adjusted to 49.3 and 23.8 kWh/door by eliminating the interactive electric heating savings associated the original RTF measure, as well as using a 6.3 EER and 9.1 EER respectively for the refrigeration system's efficiency, rather than the 5.4 EER and 7.8 EER efficiency used in the RTF UES workbook. Both adjustments seem appropriate.

2.1.2 Evaluation assessment of applicant methodology

13741512 / Anti-sweat heater controls

The evaluator determined that the applicant's use of eQUEST to estimate energy savings was appropriate. However, the evaluator found that the wattage per freezer door upgraded was only 202.6 Watts, as opposed to the 240.9 Watts used in the tracking calculations. Though the method was appropriate, the evaluators used actual metered result heater runtime to calculate the heater savings and CT TRM results to determine interactive savings.

13710972 / Install 54 linear feet of doors

The evaluator determined that the applicant's use of eQUEST to estimate the energy savings was appropriate. The eQUEST file used in the tracking calculations modeled 80′ of new doors added, and then scaled those savings by a ratio of 54′ / 80′, whereas the evaluator simply modeled 54′ of new doors in eQUEST.

13249513 / Clean condenser and evaporator coils

The tracking calculations assumed that cleaning the coils would improve the heat capacity of the coils from 85% of the nominal heat capacity back up to the original heat capacity of 100%, and this would result in an increase to the EER by 1.66% to 3.8%. The evaluator performed a brief literature review and found two studies that measured the change in EER caused by cleaning condenser and evaporator coils based on empirically collected data. The most recent study, ASHRAE RP-1705, only looked at condenser coils, and found that cleaning the condenser coils produces no significant effect on performance. The change to the EER in this study was -1.8% ±5.7% at 68% confidence. The reason cited for this finding was that fouling induced turbulence (like fins) and while pressure drop increased and airflow reduced with increased fouling, turbulence increased and thus the overall impact on heat capacity could in some cases be increased. The CPUC study found 4.7% ±1.8% savings at 68% confidence, but this study differed from the ASHRAE study in that it extrapolated laboratory results relating to reduced pressure drop to reduced airflow and capacity to field results, without fouling the condenser coils in the same manner used by the ASHRAE study. The CPUC study found that cleaning evaporator coils resulted in a change to the EER by 0.15% ±0.13% at 68% confidence. Additionally, for the condenser coil cleaning coil measures on refrigeration systems, the tracking calculations used diversity factors of 88%, 99%, and 100% for the refrigeration systems, and 26% for an RTU that had the evaporator coils cleaned. After a discussion with several parties, the evaluators settled on using a consistent diversity factor of 85% for refrigeration equipment. The 26% value for the RTU equipment is left unchanged by the evaluator. However, this diversity factor discussion, for refrigeration equipment and RTU equipment, warrants further investigation to potentially inform future similar projects.

[6] Microsoft Word - ComFac Evaluation V3 HIM Appendices _02-18-2010_.doc (calmac.org) and Microsoft Word - ComFac Evaluation V1 Final Report _02-18-2010_.doc (calmac.org) and 20190618DoorGasketPres.pptx | Powered by Box

13475450 / Install door gaskets on 61 reach-in freezer doors

The evaluator agrees with the energy savings estimates for this measure, which cite a very thorough analysis of this measure developed by the RTF in the Pacific Northwest, based on a 2006 ADM report performed for the CPUC. However, while the energy savings for this project reference the RTF's well-researched measure documentation, the measure life does not. This project claims five years of measure life savings, while the RTF's measure documentation only claims one year. It was due to this low measure life that the measure was not found to be cost-effective in the RTF's jurisdiction, and the measure was removed from the program offerings there.

2.2 On-site inspection and metering

The site contact indicated that it was safe to visit the site and preferred an on-site verification with spot measurement of the evaluated measure. The evaluator conducted the site visit on October 30, 2023, and was assisted by the store manager. Table 2-2 summarizes the findings from the installed measure verification.

Table 2-2. Measure verification

Measure name	Verification method	Verification result
13741512 / Anti-sweat heater controls	On-site inspection / Logger Installation	The evaluator verified the number of freezer and cooler doors that had anti-sweat heater controls installed. In addition, the evaluator installed Amp loggers on the anti-sweat heater heaters for 10 cases and took a spot kW measurement of the anti-sweat heater on (1) case. The amp loggers recorded at 1-minute intervals, from 10/30/23 until 1/11/24. In addition to the Amp loggers, the evaluator installed temperature and humidity loggers inside the store, as well as a temperature and relative humidity logger outside of the store, on the roof, to measure the indoor and outdoor dew point temperature. This was to be able to relate indoor dew-point temperature to anti-sweat heater on-time, as well as to relate outdoor dew point temperature to indoor dew point temperature.
13710972 / Install 54 linear feet of doors	On-site inspection / site-contact interview	The evaluator verified the number and length of all installed case doors and confirmed with the site-contact the location of the installed doors. In addition, the evaluator collected temperatures of the cases where the new doors were added, to be able to update the eQUEST model if necessary.
13249513 / Clean condenser and evaporator coils	Site-contact interview / visual inspection	The evaluator confirmed with the site-contact that the coil cleaning measure occurred, and visually verified the condenser coils on the refrigeration circuit.
13475450 / Install door gaskets on 61 reach-in freezer doors	On-site inspection / site-contact interview	Interviewed the site-contact to confirm that the measure occurred. The site-contact confirmed it occurred but could not confirm which cases received new door gaskets.

Table 2-3 shows the list of the amp and kW loggers installed on the (10) cases at this site. The Volts and Power Factor (PF) were measured with the DENT kW logger, and assumed to be the same for the other cases where only Amps were measured.

Table 2-3. Amp and kW loggers installed for anti-sweat heater measure

Case name	Cooler or freezer	Number of doors	Logger number	Volts	PF	Amps when on	Watts when on	Amps/ door	Watts/ door
17EC2	Freezer	2	DENT 8005 HOBO 7384	118	1.00	3.4	404.3	1.7	202.1
16EC2	Freezer	3	HOBO 4821 & HOBO 7413	*118*	*1.00*	4.7	552.3	1.6	184.1
16L5	Freezer	5	HOBO 4817	*118*	*1.00*	6.2	727.6	1.2	145.5
16L4	Freezer	5	HOBO 4818	*118*	*1.00*	7.3	861.8	1.5	172.4
16L2	Freezer	5	HOBO 7382	*118*	*1.00*	7.5	883.2	1.5	176.6
16L6	Freezer	5	HOBO 7409	*118*	*1.00*	7.7	905.2	1.5	181.0
16R6	Freezer	4	HOBO 7414	*118*	*1.00*	6.2	728.0	1.5	182.0
16R5	Freezer	4	HOBO 7420	*118*	*1.00*	5.7	671.3	1.4	167.8
16L3	Freezer	5	HOBO 7423	*118*	*1.00*	7.4	874.0	1.5	174.8
16L1	Freezer	5	HOBO 7425	*118*	*1.00*	7.4	873.0	1.5	174.6

Table 2-4 shows the average Amps per door and Watts per door for the 2-door, 3-door, 4-door, and 5-door freezer cases.

Table 2-4. Average Amps/door and Watts/door of 2-door, 3-door, 4-door, and 5-door freezer cases

Door quantity	Sample size	Average amps / door	Average watts / door
2-door	1	1.71	202.1
3-door	1	1.56	184.1
4-door	2	1.48	174.9
5-door	6	1.45	170.8
Total	10		

Figure 2-1 shows the evaluator measured daily anti-sweat heater % on-time versus the daily average indoor dew point temperature for the 10 Amp loggers deployed on the 10 cases between 10/30/2023 and 1/11/2024, as well as the linear regression line finding the least-squares residual relationship through all the data.

Figure 2-1. Daily average anti-sweat heater % on-time vs. daily average indoor dew point temperature for 10 loggers on 10 reach-in freezer cases

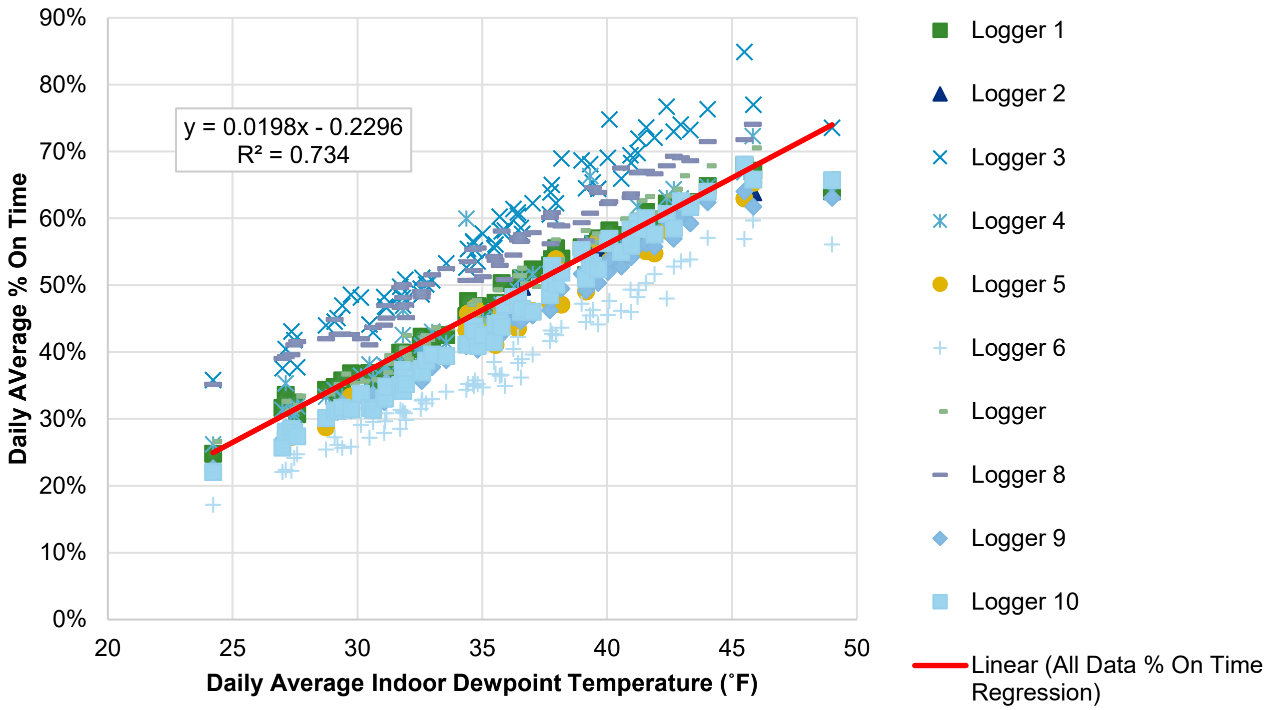
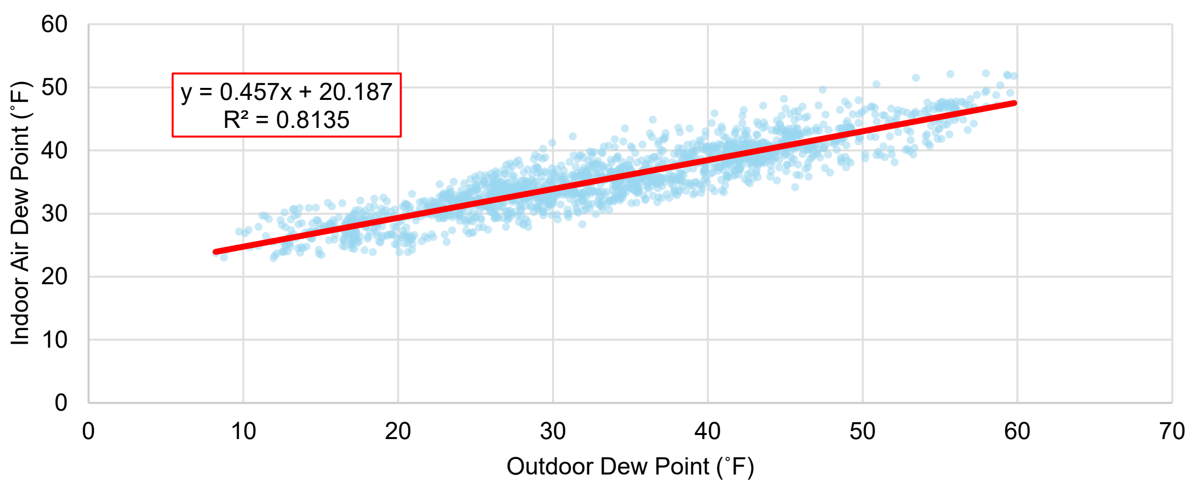


Figure 2-2 shows the evaluator measured hourly indoor dew point temperature plotted against the evaluator measured hourly outdoor dew point temperature.

Figure 2-2. Measured hourly indoor dew point temperature vs. measured hourly outdoor dew point temperature



2.3 Evaluation methods and findings

13741512 / Anti-sweat heater controls

The evaluators used the measured Amps and Watts shown in Table 2-3 and Table 2-4, and applied those measured results to the total number of cases that were found on-site to determine the total connect Wattage. During the site visit, the evaluator counted 145 freezer doors, and 12 cooler doors, whereas the tracking calculations showed 146 freezer doors, and 12 cooler doors. The tracking calculations showed 366 linear feet of freezer doors and 30 linear feet of cooler doors, but with each door being 2.5′ wide, this translates to 146 freezer doors and 12 cooler doors. Applying the Watts/door by case in Table 2-4 to the number of cases (145) results in 25,633 connected Watts, not including the additional load on the refrigeration system required to remove the heat from the door heaters. Additionally, since the evaluators did not measure any of the coolers, the evaluators used the tracking value of 48 Watts/door for the coolers (including compressor load interactive effect) and used the post-case cooler hours from the tracking calculations of 2,278 hours per year. Referencing a SDGE workpaper[7] that estimates that 35% of the connected door heater Wattage ends up as load on the refrigeration system, and the CT PSD, the total connected wattage of the door heaters and the additional load on the refrigeration system is 29,953.5 Watts. The baseline operation consists of running that 29,953.5 Watts continuously, whereas the post-case consists of cycling the door heaters on and off to ensure that the glass temperature stays above the store dew point temperature to ensure that the glass does not fog up.

TMY3 weather data from the nearest weather station was used to supply the hourly outdoor dew point temperature for each hour of the year. Using the outdoor dew point temperature vs. indoor dew point temperature data shown in Figure 2-2, the indoor air dew point temperature was estimated for each hour of the year. With an estimate for the indoor dew point temperature for each hour of the year, the relationship between indoor dew point temperature and post-case door heater % on time from the data shown in Figure 2-1, the post-case door heater % on time could be estimated for each hour of the year. With the baseline % on-time being 100% for all 8,760 hours per year, and the post-case % on-time calculated as described previously, and with the total connected Wattage, the energy savings could be calculated.

The evaluator savings algorithm is as follows:

Freezer kWh Savings = Freezer Door Quantity x
Volts x
Measured Freezer Door ASH amps per door x
power factor x
(8760 baseline hours – post-case hours) x
1 kW / 1,000 Watts x
(1 + 35% of load from heater that ends up as load on case[8])/ 2.03 ACOP[9]

Cooler kWh Savings = Cooler Door Quantity x
Tracking Watts/Door Inclusive of Interactive Effects x
(8,760 baseline hours – Post Case Hours) x

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7 https://view.officeapps.live.com/op/view.aspx?src=https%3A%2F%2Fwww.sdge.com%2Fsites%2Fdefault%2Ffiles%2FWPSDGENRRN0009%252520Rev%2525200%252520Anti-Sweat%252520Heat%252520%252528ASH%252529%252520Controls%252520_0.doc&wdOrigin=BROWSELINK

8 From SDG&E workpaper https://www.sdge.com/sites/default/files/WPSDGENRRN0009%2520Rev%25200%2520Anti-Sweat%2520Heat%2520%2528ASH%2529%2520Controls%2520_0.doc

9 From CT PSD

13710972 / Install 54 linear feet of doors

The evaluator used the same methodology that was used in the tracking calculations, which was the use of an eQUEST model. The evaluator updated the case temperatures in the eQUEST model, and modeled 54′ of new doors, rather than modeling 80′ of new doors, and then pro-rating that by 54/80. The realization rate for this measure ended up being 100%.

13249513 / Clean condenser and evaporator coils'

The evaluator estimated the energy savings for these two measures by first simplifying the tracking bin calculations into simplified calculations which include only the key parameters.

For example, the tracking calculations are simplified to the following equations for the medium temperature refrigeration cases:

$$kWh\ Savings_{MediumTemp} = 18.2 \text{ tons Design } \times 88\% \text{ Diversity Factor } \times \left(\frac{1}{13.11 \text{ EER}_{\text{baseline}}} - \frac{1}{13.61\ EER_{post}}\right) \times \frac{12{,}000 \text{ Btuh/ton}}{1{,}000\ Watts/kW} \times 8{,}760 \frac{\text{hours}}{\text{year}} =$$

$$kWh\ Savings_{MediumTemp} = 11{,}397\ kWh$$

The above tracking calculations save energy because of the improvement in EER by 3.80%. The evaluator performed a brief literature review and found two studies which measured the change in EER caused by cleaning condenser and evaporator coils based on empirically collected data. The most recent study, ASHRAE RP-1705,[10] only looked at condenser coils in small residential HVAC applications and found that that cleaning the condenser coils produces no significant effect on performance. The change to the EER in this study was -1.75% ±5.7% at 68% confidence. The reason cited for this finding was that fouling induced turbulence (like fins) and while pressure drop increased and airflow reduced with increased fouling, turbulence increased and thus the overall impact on heat capacity could in some cases be increased. The CPUC study[11] found 4.7% ±1.8% savings at 68% confidence but this study differed from the ASHRAE study in that it extrapolated laboratory results relating to reduced pressure drop to reduced airflow and capacity to field results, without fouling the condenser coils in the same manner used by the ASHRAE study. The average of these two studies was a change to the EER by 1.475%. While neither of these studies address large refrigeration coils like those addressed in this project, there are no other known studies that have produced empirical data on the effect that coil cleaning has on a unit's coefficient of performance, so the values from these studies were used as the best available. So, for this measure the evaluator calculations were calculated as follows:

$$kWh\ Savings_{MediumTemp} = 18.2 \text{ tons Design } \times 85\% \text{ Diversity Factor } \times \left(\frac{1}{13.11 \text{ EER}_{\text{baseline}}} - \frac{1}{(13.11\ EER_{baseline}) \times (1 + 0.01475)}\right) \times \frac{12{,}000 \text{ Btuh/ton}}{1{,}000\ Watts/kW} \times 8{,}760 \frac{\text{hours}}{\text{year}} =$$

$$kWh\ Savings_{MediumTemp} = 4{,}422\ kWh$$

Note that the evaluator calculations also updated the Diversity Factor from 88% to 85%. The tracking calculations used 88%, 99%, and 100% diversity factors for coil cleaning measures on grocery store refrigeration equipment. The evaluators could not find a reliable source for typical diversity factors on grocery store equipment, so used 85%, which is the recommended value in the original tool used in the tracking calculations. This refrigeration coil cleaning measure, as well as all the

[10] "Investigation of Airside Fouling on Condenser Heat Exchangers" Yuill, David and Mehrabi, Mehdi, University of Nebraska, ASHRAE, 2019, RP-1705

[11] "Impact Evaluation of 2013-14 Commercial Quality Maintenance Programs (HVAC 3)", California Public Utilities Commission, 2016, Calmac Study ID CPU0117

refrigeration coil cleaning measures at this site, only included cleaning the refrigeration condenser coils. However, the evaluators recommend that this factor be investigated further for future savings claims of similar projects.

The same approach was used for the other measures that involved cleaning condenser coils.

Additionally, the same approach was used for calculating the evaluator savings for cleaning the evaporator coils, except that the improvement to the EER was found to be 0.15% based on the referenced CPUC study.[12]

For the cleaning the evaporator and condenser coils on the HVAC unit, the tracking savings are:

$$kWh\ Savings_{CleanCoils} = 107.5 \text{ tons design } \times 26\% \text{ Diversity Factor } \times \left(\frac{1}{11 \text{ EER}_{\text{baseline}}} - \frac{Ft}{11.34\ EER_{post}}\right) \times \frac{12{,}000\,\frac{\text{Btuh}}{\text{ton}}}{1{,}000\,\frac{Watts}{kW}} \times 5{,}872\,\frac{\text{hours}}{\text{year}} =$$

$$kWh\ Savings_{CleanCoils} = 5{,}337\ kWh$$

The tracking calculations claimed a 3.1% EER improvement for cleaning both coils (11.0 EER to 11.3 EER).

The evaluators referenced the two previously mentioned studies, to claim 1.475% EER improvement due to the condenser coil cleaning, and 0.15% for the evaporator coil cleaning, for a total of 1.62%. 1 - (100 * (1- 0.01475))* (1-0.0015) / 100 = 1.62%.

So, the evaluator savings are:

$$kWh\ Savings_{CleanCoils} = 107.5 \text{ tons design } \times 26\% \text{ Diversity Factor } \times \left(\frac{1}{11 \text{ EER}_{\text{baseline}}} - \frac{Ft}{11.18\ EER_{post}}\right) \times \frac{12{,}000\,\frac{\text{Btuh}}{\text{ton}}}{1{,}000\,\frac{Watts}{kW}} \times 5{,}872\,\frac{\text{hours}}{\text{year}} =$$

$$kWh\ Savings_{CleanCoils} = 2{,}841\ kWh$$

13475450 / Install door gaskets on 61 reach-in freezer doors

The evaluator agrees with the tracking calculation methodology for estimating the first-year energy savings for this measure, which referenced the calculation workbooks developed by the RTF for energy efficiency programs in the Pacific Northwest. The calculations reference an ADM evaluation report created for the CPUC in 2006.[13] While the evaluators agree with the tracking calculation methodology for the first-year savings estimates, the tracking calculations claim a measure life of 5 years, whereas the original RTF source documentation references a measure life of 1 year for this measure. The evaluators recommend updating the measure life from 5 years to 1 year based on the research and analysis that went into developing this measure by the RTF in the Pacific Northwest, which the tracking savings estimates are based on. The evaluation did not perform any M&V on this measure apart from confirming with the site-contact that damaged gaskets were repaired as part of this project and collecting case temperatures. The first-year savings were used as, after making slight adjustments to the eQUEST model, which ended up having no significant impact on the energy savings.

[12] "Impact Evaluation of 2013-14 Commercial Quality Maintenance Programs (HVAC 3)", California Public Utilities Commission, 2016, Calmac Study ID CPU0117

[13] Microsoft Word - ComFac Evaluation V1 Final Report _02-18-2010_.doc (calmac.org) and Microsoft Word - ComFac Evaluation V3 HIM Appendices _02-18-2010_.doc (calmac.org)

3 FINAL RESULTS

The final evaluation results are summarized in Table 3-1.

Table 3-1. Evaluation results summary

PA application ID	Measure name		Annual electric energy (kWh)	% of energy savings on-peak	Summer on-peak demand (kW)	Winter on-peak demand (kW)
13741512	Anti-sweat heater controls	Tracked	145,459	46.0%	13.04	21.04
		Evaluated	124,245	45.8%	9.06	19.79
		Realization Rate	85.4%	99.6%	69.5%	94.0%
13710972	Install 54 linear feet of doors	Tracked	17,482	41.0%	5.57	3.52
		Evaluated	17,461	41.0%	5.57	3.52
		Realization Rate	99.9%	100.0%	100.0%	100.0%
13249513	Clean condenser and evaporator coils	Tracked	20,416	60.5%	5.75	0.83
		Evaluated	9,761	53.2%	2.86	0.99
		Realization Rate	47.8%	100.0%	49.7%	119.3%
13475450	Install door gaskets on 61 reach-in freezer doors	Tracked	3,840	100.0%	0.44	0.44
		Evaluated	3,840	100.0%	0.44	0.44
		Realization Rate	100.0%	100.0%	100.0%	100.0%
Total		Tracked	187,197	48.2%	24.80	25.83
		Evaluated	155,306	47.1%	17.93	24.73
		Realization Rate	83.0%	97.6%	72.3%	95.8%

The comparison between the key parameters and the tracking and evaluator analyses are provided below.

13741512 / Anti-sweat heater controls

Table 3-2 shows the summary of the key parameters for the anti-sweat heater control measure between tracking and evaluation. The reduced evaluator energy savings are primarily due to the evaluator finding that the freezer door heat power use is 240.9 Watts, rather than 202.6 Watts as estimated in the tracking calculations.

Table 3-2. Summary of key parameters for anti-sweat heater control measure

Parameter	BASELINE		PROPOSED / INSTALLED	
	Tracking Value(s)	Evaluation Value(s)	Tracking Value(s)	Evaluation Value(s)
Freezer door quantity with ASH controls	146	145	146	145
Cooler door quantity with ASH controls	12	12	12	12
ASH freezer door heater annual operating hours	8,760	8,760	4,730	4,658
ASH cooler door heater annual operating hours	8,760	8,760	2,278	2,278
Freezer Watts per door (includes load on compressor)	240.9	202.6	240.9	202.6
Cooler Watts per door (includes load on compressor)	48.0	48.0	48.0	48.0
Freezer ASH energy use	308,088	257,345	166,364	136,836
Cooler ASH energy use	5,048	5,048	1,312	1,312
Total energy use	313,135	262,393	167,676	138,148
Energy savings			145,459	124,245

13710972 / Install 54 linear feet of doors

Table 3-4 shows the key parameters used in the eQUEST model for both the tracking and evaluator calculations. The tracking calculations modeled 80′ of cases with new doors, then multiplied the result by a pro-rating factor of 54/80, whereas the evaluator simply modeled 54′ of cases with new doors. The evaluator case temperatures are based on data collected on site.

Table 3-3. Summary of key parameters for the install of 54 linear feet of doors measure

Case #	Tracking		Evaluator	
	Length	Temp (˚F)	Length	Temp (˚F)
C3982B72E47E6	8	34	8	32
C3B82B72E47E6	20	35		
C3F82B72E47E6	4	33	4	32
C4082B72E47E6	6	33	6	32
C4A82B72E47E6	32	40	32	36
C4B82B72E47E6	6	36		
C4E82B72E47E6	4	36	4	36
Total	80		54	

13249513 / Clean condenser and evaporator coils

Project 13249513 consisted of three tracking calculation workbooks. Table 3-4 shows the comparison of key parameters between the tracking analysis and evaluator analysis for refrigeration condenser coil cleaning measure found in the first workbook. The primary driver of the discrepancy is due to the evaluator finding that the % EER improvement is only 1.475%, rather than the 1.66% and 3.80% found in the tracking analysis. The tracking analysis based its EER improvement on an assumption that cleaning the coils would increase the condenser capacity from 85% rated capacity to 100% rated capacity, without providing supporting evidence for this claim. The evaluator methodology referred to two studies,[14] which, when averaged together, find that the average EER % improvement for cleaning condenser coils is 1.475%.

Table 3-4. Summary of key parameters for condenser coil cleaning measure – Refrigeration Equipment

Parameter	BASELINE		PROPOSED / INSTALLED	
	Tracking Value	Evaluation Value	Tracking Value	Evaluation Value
Low Temp Tons Design	18.17	18.17	18.17	18.17
Low Temp Tons Operating	15.99	15.44	15.99	15.44
Low Temp Diversity Factor	88%	85%	88%	85%
Low Temp EER	7.8	7.8	8.0	8.0
EER Improvement			1.66%	1.475%
Low Temp Hours	8,760	8,760	8,760	8,760
Low Temp kWh	214,722	207,020	211,210	204,010
Med Temp Tons Design	39.2	39.2	39.2	39.2
Med Temp Tons Operating	38.8	33.3	38.8	33.3
Low Temp Diversity Factor	99%	85%	99%	85%
Med Temp EER	13.1	13.4	13.6	13.6
EER Improvement			3.80%	1.475%
Med Temp Hours	8760.0	8760.0	8760.0	8760.0
Med Temp kWh	311,114	261,130	299,718	257,333
Total kWh	525,836	468,150	510,928	461,343
kWh Savings			14,908	6,807

[14] "Investigation of Airside Fouling on Condenser Heat Exchangers" Yuill, David and Mehrabi, Mehdi, University of Nebraska, ASHRAE, 2019, RP-1705 and "Impact Evaluation of 2013-14 Commercial Quality Maintenance Programs (HVAC 3)", California Public Utilities Commission, 2016, Calmac Study ID CPU0117

Table 3-5 shows the comparison of key parameters between the tracking analysis and evaluator analysis for the condenser and evaporator coil cleaning measure found in the second workbook. The second workbook involves cleaning both the evaporator coils and condenser coils on a 27.8-ton unit that is used for space-conditioning. For this reason, the evaluator EER % improvement of 1.62% is the sum of the EER% improvement from cleaning condenser coils (1.475%), and cleaning evaporator coils (0.15%). The operating hours in Table 3-5 come from the weather-bin analysis for this HVAC equipment, and the evaluator believes these hours are reasonably, although the evaluator did not measure them to confirm.

Table 3-5. Summary of key parameters for condenser and evaporator coil cleaning measure – HVAC Equipment

Parameter	BASELINE		PROPOSED / INSTALLED	
	Tracking Value	Evaluation Value	Tracking Value	Evaluation Value
Tons Design	107.50	107.50	107.50	107.50
Tons Operating	27.77	27.77	27.77	27.77
Diversity Factor	26%	26%	26%	26%
EER	11.0	11.0	11.3	11.18
EER Improvement			3.09%	1.62%
Hours	5,872	5,872	5,872	5,872
kWh	177,901	177,901	172,564	175,060
kWh Savings			5,337	2,841

Table 3-6 shows the comparison of key parameters between the tracking analysis and evaluator analysis for the condenser coil cleaning measure on HVAC equipment found in the third workbook.

Table 3-6. Summary of key parameters for condenser coil cleaning measure – HVAC Equipment

Parameter	BASELINE		PROPOSED / INSTALLED	
	Tracking Value	Evaluation Value	Tracking Value	Evaluation Value
Tons Design	0.86	0.86	0.86	0.86
Tons Operating	0.86	0.73	0.86	0.73
Diversity Factor	1.00	0.85	1.00	0.85
EER	9.9	9.9	10.1	10.02
EER Improvement			1.91%	1.48%
Hours	8,760	8,760	8,760	8,760
kWh	9,152	7,779	8,980	7,666
kWh Savings			171	113

13475450 / Install door gaskets on 61 reach-in freezer doors

Table 3-7 shows the comparison of key parameters between the tracking analysis and evaluator analysis for the measure that involves installing door gaskets on 61 reach-in freezer doors. The evaluator accepted the first-year savings estimates that were developed in the tracking analysis, as they reference the RTF methodology used in the Pacific Northwest. For this reason, the evaluator and tracking analysis result in the same first-year energy savings. However, the tracking analysis claims a measure life of 5 years, even though the RTF only claims 1 year for this measure. For this reason, the evaluator recommends that the RI program consider updating the measure life from 5 years to 1 year.

Table 3-7. Summary of key parameters for adding gaskets to reach-in freezer doors.

Parameter	BASELINE		PROPOSED / INSTALLED	
	Tracking Value	Evaluation Value	Tracking Value	Evaluation Value
Door Quantity Low Temp	6	61	61	61
kWh Savings/Door Medium Temp			49.3	49.3
kWh Savings Medium Temp			3008.4	3008.4
Door Quantity Medium Temp	35	35	35	35
kWh Savings/Door Medium Temp			24	24
Total kWh Savings			832	832
kWh Savings			3,840	3,840

3.1 Explanation of differences

Table 3-8 shows the list of discrepancies for each measure, the impact the discrepancy had on the overall evaluation results, as well as the reason for the discrepancy.

Table 3-8. Discrepancy analysis

Measure	Discrepancy	Parameter	Impact of deviation	Discussion of deviations
13741512 / Anti-sweat heater controls	Operation	Freezer Wattage	-14.5%	Decreased savings – Evaluator used the measured Wattage per door of 202.6 Watts, rather than the 240.9 Watts used in the tracking calculations.
13741512 / Anti-sweat heater controls	Operation	Freezer Post Hours	1.4%	Increased savings – The evaluator found that the freezers operated 4,658 hours in the post-case, rather than 4,730 hours which were used in the tracking calculations

Measure	Discrepancy	Parameter	Impact of deviation	Discussion of deviations
13741512 / Anti-sweat heater controls	Quantity	Freezer Door Count	-0.18%	Decreased savings – The evaluator found 145 freezer doors with new ASH controls, whereas the tracking calculations stated 146.
13710972 / Install 54 linear feet of doors	Operation	Operating Temperature	-0.013%	Decreased savings – slight discrepancy between the observed case temperatures, and the case temperatures used in the tracking calculations
13741512 / Clean condenser and evaporator coils	Operation	EER % Improvement	-3.5%	Decreased savings – Evaluator used an EER % improvement of 1.475% for condenser coil cleaning, 0.15% for evaporator coil cleaning, and 1.62% for cleaning both evaporator and condenser coils by referencing two studies [1], whereas the tracking calculations claimed between 1.66% and 3.80%, by making an engineering estimate that a unit's capacity would increase from 85% to 100% from cleaning the coil.
13741512 / Clean condenser and evaporator coils	Operation	Diversity Factor	-0.26%	Decreased savings – Evaluator used a diversity factor of 85%, whereas the tracking calculations used a diversity factor of 88%, 99%, and 100% for coil-cleaning measures that took place on grocery store refrigeration equipment.
13475450 / Install door gaskets on 61 reach-in freezer doors	N/A	N/A	0.0%	No discrepancies

13475450 / Install door gaskets on 61 reach-in freezer doors

There were no deviations between the evaluator first-year savings and the tracking first-year savings for the door gasket measure. While the evaluators agree with the tracking calculation methodology for the first-year savings estimates, the tracking calculations claim a measure life of 5 years, whereas the original RTF source documentation references a measure life of 1 year for this measure. The evaluators recommend updating the measure life from 5 years to 1 year based on the research and analysis that went into developing this measure by the RTF in the Pacific Northwest, which the tracking savings estimates are based on.

3.2 Lifetime savings

The evaluator calculated applicant and evaluated lifetime savings values using the following formula:

$$\mathrm{LAGI} = \mathrm{FYS} \times [\,\mathrm{RUL} + \mathrm{outyear}\,\% \times (\mathrm{EUL} - \mathrm{RUL})]$$

where:

LAGI =	lifetime adjusted gross impact (therms)
FYS =	first year savings (therms)
EUL =	measure life (years)
RUL =	1/3 of EUL (years)
outyear % =	100% for this single baseline measure

The evaluated lifetime savings are lower than the tracking lifetime savings because the evaluated first year savings are lower than the tracking first year savings for the anti-sweat heater measure, the measure that involves cleaning the evaporator and condenser coils, and because the evaluated EUL is lower for the door gasket measure than the EUL used in the tracking calculation. Table 3-9 provides a summary of key factors that influence the lifetime savings for the four measures.

Table 3-9. Lifetime savings summary

Measure	Factor	Tracking	Application	Evaluator
13741512 / Anti-sweat heater controls	Lifetime savings (kWh)	1,454,590	1,454,590	1,242,445
	First-year savings (kWh)	145,459	145,459	124,245
	Measure lifetime (years)	10	10	10
	Baseline classification	Retrofit	Retrofit	Add-on retrofit
13710972 / Install 54 linear feet of doors	Lifetime savings (kWh)	227,266	227,266	226,993
	First-year savings (kWh)	17,482	17,482	17,461
	Measure lifetime (years)	13	13	13
	Baseline classification	Retrofit	Retrofit	Add-on retrofit
13249513 / Clean condenser and evaporator coils	Lifetime savings (kWh)	40,832	40,832	19,521
	First-year savings (kWh)	20,416	20,416	9,761
	Measure lifetime (years)	2	2	2
	Baseline classification	O&M	O&M	O&M
13475450 / Install door gaskets on 61 reach-in freezer doors	Lifetime savings (kWh)	19,200	19,200	3,840
	First-year savings (kWh)	3,840	3,840	3,840
	Measure lifetime (years)	5	5	1
	Baseline classification	O&M	O&M	O&M
Total	Lifetime savings (kWh)	1,741,888	1,741,888	1,492,800

Measure	Factor	Tracking	Application	Evaluator
	First-year savings (kWh)	187,197	187,197	155,306
	Measure lifetime (years)	9.3	9.3	9.6

3.3 Ancillary impacts

A total of 136.74 MMBtu of increased gas usage was claimed in the tracking database for the ant-sweat heater measure. This measure had a realization rate of 85.4%. The evaluators estimate that the gas usage increased by 116.80 MMBtu because of the anti-sweat heater measure.

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