Performance Evaluation of Liquid-Cooled Open Stand-Alone Refrigerated Cases

Alex Bulk, Grant Wheeler, and Ramin Faramarzi

Produced under direction of ComEd by the National Renewable Energy Laboratory (NREL) under Technical Services Agreement TSA-19-01159

NREL is a national laboratory of the U.S. Department of Energy
Office of Energy Efficiency & Renewable Energy
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Contract No. DE-AC36-08GO28308

Strategic Partnership Project Report
NREL/TP-5500-81945
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Suggested Citation
Bulk, Alex, Grant Wheeler, and Ramin Faramarzi. 2023. *Performance Evaluation of Liquid-Cooled Open Stand-Alone Refrigerated Cases.* Golden, CO: National Renewable Energy Laboratory. NREL/TP-5500-81945.https://www.nrel.gov/docs/fy22osti/81945.pdf.

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National Renewable Energy Laboratory
15013 Denver West Parkway
Golden, CO 80401
303-275-3000 • www.nrel.gov

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This work was authored by the National Renewable Energy Laboratory, operated by Alliance for Sustainable Energy, LLC, for the U.S. Department of Energy (DOE) under Contract No. DE-AC36-08GO28308. Support for the work was also provided by ComEd through CLEAResult under TSA-19-01159. The views expressed in the article do not necessarily represent the views of the DOE or the U.S. Government.

The ComEd Energy Efficiency Program is funded in compliance with state law.

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Acknowledgments

The authors would like to thank the ComEd Energy Efficiency Program and CLEAResult for supporting this research. This report was prepared by the National Renewable Energy Laboratory (NREL) Building Technologies and Science Center.

The authors would also like to thank Juan Catano Montoya, Robert Nash, Frank Wallis, Diane Patrizio, Carmem Valle-Pereira, Katrina Krites, and Suresh Shivashankar of Emerson for their detailed technical support and assistance in developing this experimental assessment in addition to their report review. Additionally, the authors would like to express their sincere gratitude to Gabriel Duarte and Brennan Karalus of CLEAResult and Steven LaBarge of ComEd for their in-depth report review.

The authors would also like to thank supporting members of NREL: Omkar Ghatpande for providing electrical engineering assistance, advice and expertise; to Erin Lewis for providing refrigeration assistance, advice, and expertise; as well as to Lauren Klun for her in-depth review and support as the project controller.

For more information, contact:
Alex Bulk
National Renewable Energy Laboratory
Email: alexander.bulk@nrel.gov

List of Acronyms

A	ampere: unit of current
AC	alternating current
AHRI	The Air Conditioning, Heating, and Refrigeration Institute: a North American trade association of air-conditioning, heating, and commercial refrigeration equipment manufacturers
ANSI	American National Standards Institute: a private nonprofit organization that oversees consensus standards for products, services, processes, systems, and personnel
ASHRAE	A professional association seeking to advance HVAC and refrigeration systems design and construction
CIMS	Copeland Indoor Modular Solution: line of Emerson-Copeland energy-efficient retrofit condensing unit models for commercial refrigerators
ComEd	The Commonwealth Edison Company: the sole electric utility provider for Chicago and Northern Illinois
DBT	dry-bulb temperature: temperature of air when shielded from radiation and moisture
DOE	U.S. Department of Energy
DPT	dew-point temperature: temperature of air to achieve saturation at constant pressure and constant water vapor content
FDA	U.S. Food and Drug Administration, a federal agency that develops and enforces food safety regulations
GWP	global warming potential: a measure of how much heat a greenhouse gas traps in the atmosphere relative to carbon dioxide
HFC	hydrofluorocarbon: a type of refrigerant composed of organic compounds that contain fluorine and hydrogen atoms with a GWP thousands of times greater than carbon dioxide
HFO	hydrofluoroolefin: a type of refrigerant composed of unsaturated organic compounds that contain hydrogen, fluorine, and carbon. They are chemically stable and offer lower GWP than HFC refrigerants
HP	horsepower: a unit of measurement of power
HVAC	heating, ventilating, and air conditioning
Hz	hertz: unit of electrical frequency
in	inch: a unit of measurement of length
kg	kilogram: a unit of measurement of mass
kW	kilowatt: a unit of measurement of power
kWh/day	kilowatt hours per day: a unit of measurement of energy consumed over one 24-h period
LAP	Laboratory Assessment Plan
LED	light-emitting diode: a semiconductor light source
mL	milliliter: a unit of measurement of volume
m/s	meters per second: a unit of measurement of velocity
NREL	National Renewable Energy Laboratory
OCL	Optimization and Control Laboratory: the NREL laboratory in which the refrigerated display cases are evaluated in an environmental chamber
Ph	phase: the number of distinct electrical wave cycles

R448a	A type of HFO refrigerant blend composed of Solstice N40® that exhibits low GWP designed to replace R404a and R22 in certain types of refrigeration systems (GWP of 1273)
RH	relative humidity: the ratio of absolute water vapor concentration to the maximum water vapor concentration at a specific temperature
TC	thermocouple: a temperature transducer
V	volts: unit of measurement of voltage
WBT	wet-bulb temperature: temperature of air cooled to saturation by the evaporation of water

Executive Summary

This project is part of a joint effort between Commonwealth Edison Company (ComEd) and the National Renewable Energy Laboratory (NREL) to evaluate the energy and demand savings potential of emerging efficient buildings technologies. This project quantifies the energy efficiency benefits from retrofitting an air-cooled, constant-speed compression, self-contained medium-temperature open-vertical display case with a high efficiency water-cooled condensing unit. In addition to a water-cooled heat rejection mechanism, the high-efficiency condensing unit utilizes a variable-speed compressor, electronic expansion valve and advanced controls system. The focus of this project, however, was mainly on capturing the energy savings potentials of the water-cooled condenser, variable-speed compression and electronic expansion valve, and not on the advanced controls. The results of this evaluation will be considered by ComEd and CLEAResult for future energy efficiency rebate offerings.

The refrigerated display cases' performance was evaluated in a controlled environmental chamber at representative indoor dry-bulb and humidity conditions inspired by the ANSI/ASHRAE 72-2018 method of evaluation [2]; however, to better represent customer operation of the units, this method was slightly modified for this study. These methods were used to evaluate each technology under nearly equivalent conditions, and covered critical power, temperature, pressure, and condensate measurements in their respective instrumentation and monitoring procedures.

The energy efficient condensing unit was evaluated at the following three water inlet temperatures:

1. 55 °F – represents a closed-loop application where water is mechanically chilled prior to entering the condenser
2. 80 °F – represents scenarios including open loops with high ground water temperature or closed loops with cooling towers
3. 108 °F – represents an extreme scenario where the saturated condensing temperature of the water-cooled unit matches the air-cooled baseline unit.

The table below lists the case total and component energy savings at the midpoint supply temperature (80 °F) representative of a typical summer ground water or cooling tower temperature. Estimated condenser pump power is shown based on a low efficiency (40 %) and high efficiency (80 %) pump used to supply measured hydraulic power. The energy consumption was measured across 24-hour experiments initiated with a defrost cycle. The total and component power consumption averaged across the total compressor operation time is also provided.

The compressor consumed a majority of the energy in all assessments (89 % of the total energy in the baseline unit and case and 85 – 89 % of the total energy in the energy-conserving measure). The lower saturated condensing temperature of the water-cooled system resulted in a lower temperature lift and increased refrigeration effect. The reduced temperature lift resulted in a reduced compressor energy of 34.1 %. The variable speed compression contributed to these savings because as a result of the improved refrigeration effect, higher suction pressures allowed the compressor to operate at a lower RPM. The lighting and evaporator fan motors consumed an equivalent amount of energy in all tests. This comprised 2 % of the baseline unit and case energy and 4 – 5 % of the energy-efficient unit and case energy. The controller consumed little energy at less than 1 % in both the baseline and energy-efficient unit at all liquid temperatures. The air-cooled condenser fan motor contributed to 5 % of the baseline energy consumption. Energy consumed by the water-cooled pump was calculated from the hydraulic power using a range of estimated pump efficiencies between 40 and 80 %. This yielded an estimated condenser pump energy at only a fraction of that consumed by the baseline condenser fan (2 – 4 % of total energy at 40 – 80 % efficiency, respectively). Even if assuming a very low-efficiency pump, condenser pump energy would not contribute significantly to overall energy consumption.

Table ES-1. Total and Component Energy Savings

ECM @ 80°F Inlet Water Temperature	% Component Energy Savings
Total	**34.8 %**
Compressor	34.1 %
Evaporator Fans	1.03 %
Lighting	0.01 %
Controller	-50.2 %
Est. Supply Pump @ 40 % / 80 % Eff.	90.5 % / 95.2 %
Total w/ Est. Supply Pump	**34.5 % / 34.3 %**

Overall, the liquid-cooled condensing unit consumed 34.8 % less daily energy than the baseline when supplied with water at and below typical ground water, city main or water tower temperatures. Improved control strategies also reduced product temperature variation from 2.4 to 2.1 °F by limiting compressor cycling. Although refrigeration energy use was reduced through adoption of a liquid-cooled condensing technology here, total building or service territory-wide energy savings cannot be expected to be proportional to these savings due to building-specific requirements for supplying liquid to the condenser heat exchangers. This experimental evaluation only compared total refrigerator case energy use and does not account for total energy required to provide conditioned liquid coolant – instead, only estimating pump energy from hydraulic power.

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1 Introduction/Project Description

The purpose of this project is to evaluate the energy savings potential of a variable-speed, self-contained, open-vertical, medium-temperature commercial refrigerated display case utilizing liquid-cooled condensing technology. The power and daily energy consumption and refrigeration performance are compared between a baseline air-cooled case and an Emerson "Energy-Conserving Measure" (ECM), which includes a liquid-cooled condensing unit, variable speed compressor, electronic expansion valve and advanced controls. The following lists the specifications of the baseline case and ECM that will be evaluated in this project:

- **Baseline**: An 8ft, 5-deck/4-shelf case utilizing an air-cooled condenser, fixed speed compressor, thermal expansion valve (TXV), and R-448a, a drop-in refrigerant in the hydrofluoro-olefin family (GWP of 1273).
- **ECM**: A retrofit package for the baseline case using R-448a consists of a liquid-cooled condensing unit, a variable-speed scroll compressor, and an electronic expansion valve (EXV). An advanced controller operates the variable-speed compressor and EXV, and provides improved control.

The liquid-cooled system required a pump to circulate water to the condenser, which consumed energy and could adversely affect total energy savings. Furthermore, some setups for the liquid loop could require additional cooling to maintain the liquid temperature. The scope of this project did not include the power to condition the loop. However, the project did estimate pump power for the ECM based on the measured hydraulic power. Additional cooling of the condenser loop, while not needed in all scenarios, could affect net energy savings. A good future study would be to evaluate total building energy savings with various configurations compared to common baselines in supermarkets.

Recent studies have shown that using liquid-cooled condensing technology as a replacement for air-cooled condensers can reduce energy consumption if the condensing temperature is low enough [3-6]. In some water-cooled condensers, it has been shown that for every degree of inlet coolant temperature reduction, total energy consumption can be reduced by 2 % [3, 6]. Additionally, it has been shown that liquid-cooled cases can typically operate with less refrigerant [6].

These studies have primarily focused on modeling or experimental assessments of only residential refrigeration systems. Liquid-cooled self-contained cases have only recently become available on the market in the U.S. and there has been little publicly-available research for end-users. Open-vertical medium temperature merchandizers have a vast presence in the retail food industry across the U.S., and self-contained versions are popular in restaurants, convenience stores, and small supermarkets. Therefore, improving the energy efficiency of these cases compared to self-contained refrigerators with air-cooled condensers presents significant potential for energy savings.

The baseline and ECM condensing units were set to operate such that the temperature of case "product simulators" were maintained within acceptable ranges as defined by AHRI and the U.S. Food and Drug Administration [7, 8]. The display case was evaluated in an environmental control chamber set to maintain temperature and humidity conditions within a range as defined by ASHRAE [2]. The total daily energy use was calculated from metering the total case and component power and integrating power consumption over each 24-hour experiment.

The power measurements included the compressor, condenser fan motor, evaporator fan motors (quantity: 3), lighting, and controller. Since the pump at the laboratory fluid-conditioning module (FCM) was unable to be metered and is oversized for the ECM condenser, bypass lines and valves were used to reduce flow to the desired range at the condenser inlet, and an estimated pump power was included for the ECM. The mass of condensate was also measured over 24 hours by a weigh drum filled with condensate discharged through a lab-installed pump. Refrigerant temperatures and pressures were measured to provide an overview of thermodynamic performance to ensure normal operation and expected trends. These measurements are included in Appendix C. Product temperatures and air temperatures at the condenser, evaporator, and air curtain are also provided in the other sections of the Appendices.

The following section contains a detailed description of the refrigerated case technologies evaluated. This is followed by a detailed overview of the experimentation design, set up, data collection, and analysis.

1.1 Technology Description

The same open-vertical, self-contained, medium temperature display case was used for both the baseline and ECM. Therefore, the case dimensions, shelving, evaporator (heat exchanger and fans), interior lights and refrigerant piping were the same for both the baseline and ECM. As a self-contained case, all components of the vapor-compression cycle are contained within the display case assembly. Therefore, the baseline air-cooled condensing unit rejected the total heat of refrigeration to the small volume of the environmental control chamber, which affected the environmental conditions in front of the case. In order to maintain comparable conditions between the baseline and ECM, a baffling system and booster fan was constructed around the baseline condensing unit to divert the rejected heat to the chamber return grille. This would prevent warm ejected air from mixing with the environmental chamber and potentially entraining warm air into the case's air curtain.

A liquid-cooled condenser requires a system established to provide liquid coolant at a temperature typically ranging from 45 – 105 °F. For convenience stores, supermarkets, and restaurants, this could involve implementing a pumping system to supply coolant to one or multiple condensing units. Liquid coolant could be supplied from various sources in either an open loop using ground/city main water, or in a closed loop using a cooling tower or a chiller. An example of one of these configurations is shown in Figure 1. For this project, a fluid conditioning module (FCM) in NREL's laboratory was used to supply liquid coolant at a controlled temperature and flowrate. The energy use of this FCM, however, was unable to be evaluated. The unit was also oversized for this project's purpose and so would not exemplify the actual energy of a pump skid used in a typical scenario. For the purposes of this study, the pump power consumption was estimated based on the hydraulic power calculated from pressure and flowrate measurements made across the condenser at a range of typical pump efficiencies.

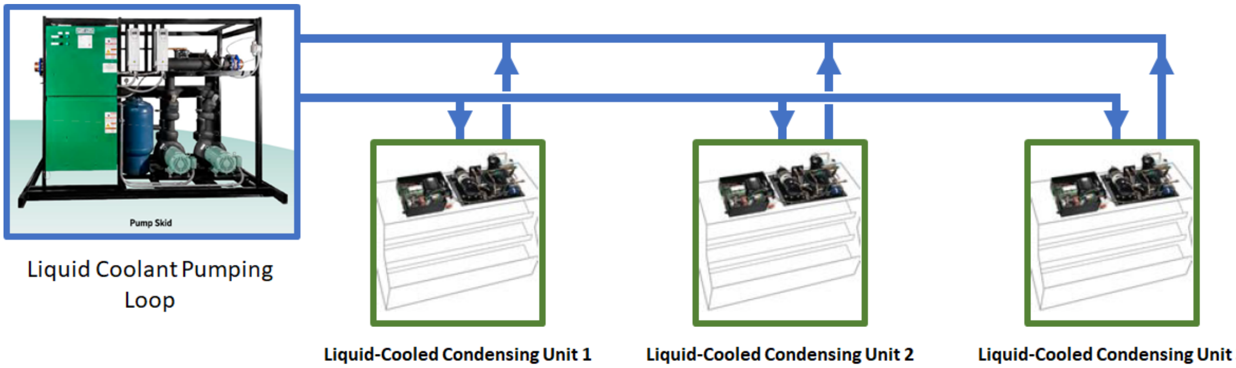


Figure 1. Example Liquid-Cooled Loop for Commercial Refrigeration

1.1.1 Baseline Display Case and Condensing Unit

The catalogue image and drawing of the refrigerator case used with both the baseline and ECM condensing unit is shown in Figure 2. The display case is a commercially available five-deck fixture with a stainless-steel interior that runs on R448a refrigerant. A mass of refrigerant was charged so that the system maintained 0 °C subcooling or greater at steady-state operation, as specified by the manufacturer. The unit is 99.25 inches in length, 45 inches deep, and 83.375 inches tall (89.375” with casters installed). The interior volumetric capacity between shelves, product load lines, and the air curtain is 92.66 cubic feet based on the ASHRAE 72 calculation [2]. The case has four stacks of two shelves that are each 48” wide and 22” deep. The bottom deck is 96.25” wide and 30.5” deep. The rear of each shelf marks the product load line, which sits a few centimeters from the rear wall. Air travels through the rear wall to the upper air curtain discharge, and gratings through the rear wall allow cooled evaporator discharge air to be ejected into the case product. The condensing unit is top-mounted with an optional enclosure. Optional accessories were not used for the project, and therefore the top enclosure was not included.

Figure 2. Refrigerator Display Case Catalogue Image* (Top) and Drawing (Bottom)

(Courtesy: Zero Zone, Inc. 2020), *12' model shown

The case has a single condensing unit with a 4-wired 208-230V/1-Ph/60Hz hardwire connection rated at 16.8 A. An image of the baseline condensing unit is shown in **Figure 3**. The compressor is a 2.5 HP Emerson Copeland Scroll model. The case's air-cooled condenser is a 16"x16" fin-and-tube heat exchanger 5" deep, mounted to a 15.25"-diameter fan. The fan motor is a 1/6 HP, 1550-rated RPM Emerson model induction motor. The lighting and three evaporator fans run off of a 115 V line and are rated at 0.38, and 0.9 A, respectively. The condensing unit contains a refrigerant accumulator and receiver.

Figure 3. Baseline Condensing Unit Pre-Installed

The baseline system cycled continuously to control case air discharge temperature. The control monitored an internal temperature sensor to “cut-out” the compressor when internal temperatures dropped 2.22 °C (4 °F) below a default 2.22 °C (36 °F) setpoint, and “cut-in” when internal temperatures rose 2.22 °C (4 °F) above the setpoint. Cut-in and cut-out temperatures are modifiable via the controller that can be adjusted on a digital Dixell-Emerson custom controller interface and were adjusted to maintain product temperature with the FDA limits. Condenser fans operate concurrently with fixed-speed compressor cycling, and evaporator fans operate continuously. Defrost cycles occur every six hours and terminate after the evaporator coil reaches 50 °F or after 30 minutes. This case contains no electrical defrost heater and allows natural convection to melt the frost on the evaporator coil.

1.1.2 ECM Condensing Unit

The catalogue image of the ECM condensing unit and control module that was retrofitted in the baseline display case is shown in Figure 4 below. The retrofit ECM unit is an Emerson brand concept model dubbed the “Copeland Indoor Modular Solution,” or “CIMS” unit. The entire air-cooled baseline condensing unit was replaced with the water-cooled ECM condensing unit equipped with a variable-speed scroll compressor with an advanced control module. The ECM also comes with an electronic expansion valve (EXV) that replaced the baseline case’s TXV. The evaporator is the only original vapor-compression cycle component that remained unchanged. The ECM used R448-a refrigerant, which was charged for 5 °C subcooling, according to the manufacturer’s instructions. This subcooling was monitored using the CIMS unit controller’s embedded measurements, instead of calculating from NREL-instrumented refrigerant temperatures and pressures as with the baseline.

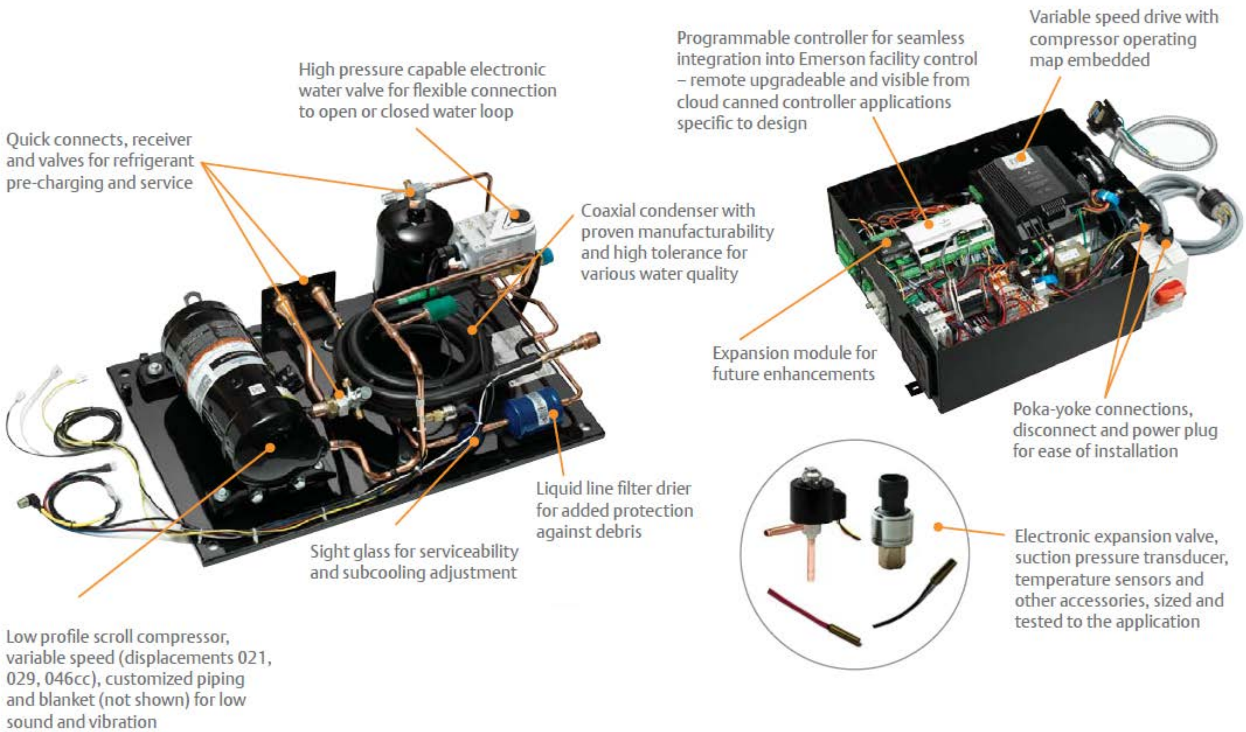


Figure 4. ECM Condensing Unit and Control Module

(Courtesy: Emerson Climate Technologies, Inc. 2020)

The ECM operates on a 230V/3-Ph/60 Hz connection which will be used to distribute the 115V lines to the baseline case lighting and evaporator fans. The compressor is a Copeland variable speed scroll compressor with a 0.75 – 2.5 HP operating range and speed ranging between 1,500 – 5,000 RPM. The condenser utilizes a counter-flow co-axial heat exchanger and a liquid control valve to regulate temperature differential across the condenser. The liquid condenser operates under a wide range of inlet temperatures, between 4.44 °C (40.0 °F) and 42.2 °C (108.0 °F). Lower temperatures are expected to provide additional reduction in compressor energy consumption. The condensing unit contains a refrigerant receiver but no accumulator. An image of the condensing unit pre-installed to the case is shown in Figure 5.

With the exception of defrost periods, the ECM compressor runs continuously at an optimized speed controlled by the VFD, and thereby, a lower peak power. Heat rejection at the condenser is regulated automatically by adjusting the coolant flow rate to maintain a target ΔT of 13.9 °C (25 °F). This target ΔT is the temperature difference between the refrigerant saturated condensing temperature and coolant inlet temperature. When the compressor shuts off, including during defrost, the liquid control valve automatically closes. The ECM controller did not affect evaporator fan operation, which ran continuously.

Figure 5. ECM Condensing Unit Before Install

There are many options for managing defrost within the ECM's embedded controls. Defrost cycles were configured to terminate when a sensor on the evaporator fins reached 8.89 °C (48.0 °F), or after 32 minutes of compressor off-cycle. The user is able to input the number of defrost cycles per 24 hours. Initially, the team attempted to match the baseline defrost schedule of four defrost cycles per 24 hour test. However, case air discharge and product temperatures started increasing before each defrost due to ice accumulation on the evaporator inhibiting air flow and the heat transfer effectiveness of the coil. Therefore, the frequency of defrosts had to be increased to six defrosts per 24 hour test only for the ECM. This is an important distinction between the baseline and ECM unit controls. End-users should consider increasing defrost frequency compared to constant-speed compressor systems that cycle frequently. All defrost cycles were time-initiated and set to terminate at an evaporator fin temperature of 48.0 °F to match the mean baseline defrost duration as close as possible, however defrost would also terminate on a 32 minute backup timer. This control scheme caused errors in the controller when a maximum liquid coolant inlet temperature of 108 °F was used due to issues with a fan restart delay continuously being tripped. Therefore, for this evaluation only, the manufacturer suggested to shut off temperature-terminated controls so that all defrost cycles terminated after 32 minutes.

Table 1 summarizes relevant specifications for the baseline and ECM condensing units.

Table 1. General Condensing Unit Specifications

(*) Manufacturer Recommendation, (**) Manufacturer Suggestion

Measure	Cond. HX Type	Comp. Type	Refrigerant, Charge Mass	V/Hz/Ph	Defrost Cycle Frequency	Rated Cooling (Btu/h)	Valve
Baseline	Air-Cooled Fin-and-Tube	Fixed Speed Scroll	R448a, 8 lb to reach ideal 0 deg subcooling*	208-230/ 60/1	6 hours, Terminated at 50 °F or 30 min	5,393	TXV
ECM	Liquid-Cooled Coaxial Tube	Variable Speed Scroll	R448a, 7.5 lb to reach ideal 5 deg subcooling*	208-230/ 60/3	4 hours, Terminated at 48 °F or 32 min**	2,010 – 6,700	EXV

Please note, that for commercial availability of the CIMS unit, please contact your Emerson Sales representative. The specific model that was used in this evaluation may or may not be production-released. Although this assessment was conducted to understand the impact of this technology in any air-cooled, medium temperature refrigerator case, the CIMS unit might not be compatible with every type of case. Therefore, availability, compatibility, and lead times will need to be discussed depending on customer requirements.

2 Laboratory Assessment Procedure

This project is meant to provide tailored performance data for these technologies under specific environmental and condenser inlet conditions. This project is therefore not intended to replicate any tests performed by rating entities for medium-temperature refrigerated cases. However, where applicable, the laboratory assessment procedure (LAP) is based on relevant rating standards: ANSI/ASHRAE 72-2018 and ANSI/AHRI 1200-2013, which prescribe key parameters and conditions under which performance assessments should be conducted, and the range under which product temperatures must be maintained to satisfy FDA requirements [2, 7, 8].

Case total power, as well as the power consumed by each major case component was collected across a 24-hour test period initiated by a defrost cycle. Repeated 24-hour assessments were collected until data was stable and repeatable for at least two consistent assessments at each condition following a period of steady-state monitoring of at least 48 hours. Daily energy consumption was calculated by integrating power data over each 24-hour test period. Measurements were collected every 200 milliseconds, but the data was averaged to minute time intervals for analysis. Temperature measurements were gathered from air probes and from within product simulators placed at various locations within each case to monitor performance. Additionally, refrigeration temperatures and pressures, environmental chamber temperatures, condenser air/water inlet/outlet temperatures, and FCM flowrates/pressures were monitored. The flowrate and pressure from the FCM at the condenser was used to calculate estimated pump power across a range of estimated pump efficiencies.

Assessments were performed within an environmental chamber at controlled indoor environmental conditions according to ASHRAE 72 [2]. A baffling system was instrumented around the top of the case which enclosed the condensing unit and removed warm air rejected from the baseline condenser to the chamber return via a centrifugal blower and VFD. This was done to prevent recirculated warm air from entering the condenser or infiltrating the air curtain. The baffling system was used during both baseline and ECM experiments to maintain equivalent airflow conditions within the chamber. Air density and pressure within the environmental chamber was unable to be corrected to altitude but did not change between the baseline and ECM evaluations. Condensate mass was also measured after each 24-hour experiment to quantify each case's moisture removal and verify similar chamber humidity conditions between experiments.

Details pertaining to the experimental setup, steady state testing, and environmental chamber evaluations are provided in the following sections.

2.1 Instrumentation/Setup

The condensing units were each installed onto the case using a forklift and were mounted to the case roof using self-tapping screws. Certified refrigerant technicians performed hot work to install the condensing unit compressor suction and condenser outlet lines to the case and then charge with R-448a refrigerant to an initial charge mass of 7 lb. The refrigerant lines were purged with nitrogen for at least 48 hours prior to charging with refrigerant. The 4-wire hardwire connections were installed to an L21-30 plug by electricians to easily connect to power at 200 – 230V, L21-30 receptacles set up outside the environmental chamber. The refrigerator case was modified to remove any optional equipment that could increase total energy consumption including evaporator condensate pans. Instead, condensate lines were connected to a laboratory condensate pumping system. When in the chamber, the case condensate was pumped to a weigh bucket for analysis of condensate mass using the case's provided condensate pump (Little Giant model 553206, 270 gph condensate pump). The condensate pump power was however not monitored since it is an optional accessory. The weigh bucket consisted of a 50-gallon drum atop a 0 – 500 lb floor scale (SellEton model SL7510).

A list of instruments used, model numbers, and accuracies are listed below in Table 2. Descriptions of the installation procedure, measurement purpose, and any use in feedback controls will be discussed in further detail in the following subsections. Only measurements used to evaluate the case are listed below. Measurements used to control coolant to the ECM condenser by the FCM is listed in Table 5.

Table 2. List of Measurement Sensors Used to Evaluate the Refrigerator Display Case and Accuracies

MEASUREMENT:	Brand/Model	Type	Accuracy
Product Simulator, Internal Air, and Chamber Dry-Bulb Temperatures	Omega/TMQSS-062U-6	1/16” Type-T thermocouple probes	± 0.50 °C (± 0.90 °F)
Chamber Dew-Point Temperature	EdgeTech/DewTrak II DPS3	chilled-mirror dew-point hygrometer	± 0.22 °C (± 0.4 °F)
Refrigerant Piping Surface Temperatures	Omega/SA1-T-SRTC	Type-T surface temperature thermocouple	± 0.50 °C (± 0.90 °F)
Case Total Plug and Compressor Power	Continental Control Systems/WMC-3Y-208-MB, Accu-CT ACTL-0750	Wattnode power meter, 50 A current transformer	± 0.50 %
Condenser Fan Power		Wattnode power meter, 20 A current transformer	
Evaporator Fans, Lighting and Baseline Controller Power		Wattnode power meter, 5 A current transformer	
Refrigerant Pressure	Omega/PX309-1KG5V	0 – 1,000 PSIG Multimedia pressure transducer	± 0.25 %
Condensate Mass	SellEton/SL7510	24”x24”, 500 lb-capacity Floor Scale	± 0.05 lbs
Air Velocity in Chamber	Kestrel 2000/312153	Wind vane anemometer	± 3.00 %

Environmental Chamber

Each case was evaluated in the Espec walk-in environmental control chamber shown in Figure 6. The interior of the chamber is 85” wide x 142.5” in length, with a 144”-high ceiling. The cases were aligned parallel to the long end of the chamber facing away from the chamber’s discharge steam/air vent. The cases were oriented exactly 12” from the back wall according to ASHRAE 72 [2], and centered in the chamber 24” from each side wall and 24” behind the ambient measurement pole. With a 24”-tall baffling structure on top of the case, 20” of clearance was available above the top, with no floor perforations around the perimeter of the case.

Figure 6. Case Setup in the Environmental Chamber

<u>Product Simulators/Filler Material/Case Temperature Control</u>

For a medium temperature display case, FDA regulations require the average of product temperatures to be maintained within 3.33 ± 1.11 °C (38 ± 2 °F) [8]. Each condensing unit controller was adjusted in order to keep the average temperature of internal "product simulators" within these FDA limits. Product simulator temperatures were measured at the left, center, and right ends of the top shelf, middle shelf, and bottom deck. At each location, two simulators were placed at the front and rear of the shelf up to the product load line. The product simulators specifications (ASHRAE 72) are as follows [2]:

- Product simulators consisted of 3" x 3" (base) x 2.5" (height) plastic containers.
- Simulators were filled with grout sponges soaked in a 50/50 (± 2 %) mix of food-grade propylene glycol and deionized water.
- Simulators were inserted with Omega brand 1/16" type-T thermocouple probes (model number TMQSS-062U-6) with ± 0.9 °F accuracy.
- Thermocouple probes were inserted through a drillhole in the simulator lid such that the tip of each probe would rest 1.5" from the bottom.
- Thermocouple pre-calibration was verified using an ice bath.

Eighteen simulators were placed at the left wall, right wall, and geometric center of the cases on the bottom deck, the second shelf from the bottom, and the top shelf. Each of these locations had two simulators placed at both the front and rear product load limit lines on the shelf. Figure 7 shows the location of these product simulators throughout the case interior.

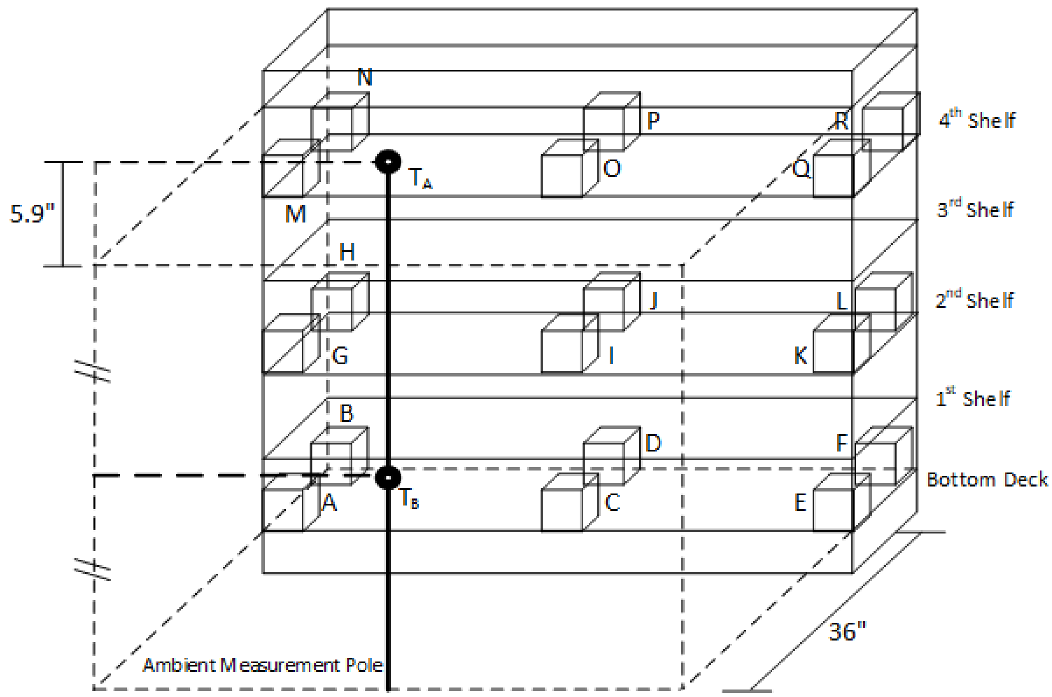


Figure 7. Product Simulator Temperature Measurement Locations (A–R), and Ambient Temperature Locations (TA and TB)

(Image modified from ASHRAE 2018 [2])

The laboratory is located at high altitude in Golden, Colorado, at 1,773 m. At this altitude, air density is nearly 20 % less than at sea level, which reduces the volumetric flow rate of the evaporator fan, thereby reducing heat transfer through the evaporator. To compensate for the effect of altitude, the temperature setpoint was adjusted on each condensing unit controller until the average of product simulator temperatures were maintained within AHRI/FDA requirements. With the baseline condensing unit, which contains a fixed-speed compressor, adjusting the setpoint directly reduces the cut-in and cut-out temperatures that trigger the compressor to turn on or off. With the ECM condensing unit, changing the setpoint affects an internal control scheme within the CIMS unit that responds to the discharge air temperature at the air curtain discharge grille. With each condensing unit, an integer setpoint was selected that generated mean simulator temperatures closest to 3 °C (37.4 °F). Each time setpoints were readjusted, the case was stabilized until deviation in all simulator temperatures was less than 0.22 °C (0.4 °F) across a 12-h period prior to initiating testing. The final setpoint and cut-in and cut-out temperatures used for each condensing unit are listed in Table 3 along with the resultant mean simulator temperature from steady-state testing.

Table 3. Selected Condensing Unit Controller Air Temperature Setpoints and Resultant Mean Product Simulator Temperature

Condition:	Setpoint Temperature (°C / °F)	Cut-In/Cut-Out Temperature (°C / °F)	Steady-State Mean Product Simulator Temperature (°C / °F)
Baseline	1.11 / 34.0	-1.11 – 3.33 / 30.0 – 38.0	3.06 / 37.5
ECM @ 108 °F water inlet	0.56 / 33.0	N/A	3.28 / 37.9
ECM @ 80 °F water inlet	0.56 / 33.0	N/A	2.78 / 37.0
ECM @ 55 °F water inlet	0.56 / 33.0	N/A	2.50 / 36.5

The net usable interior volume of the cases was loaded with "filler material" to simulate the thermal mass of food product. Although ASHRAE standards require using either propylene glycol solution or wood as filler material, NREL safety requirements have prohibited the use of glycol solutions at the required volume, and previous refrigerator case studies at NREL have shown that >70 % internal volume filled with wood can inhibit internal air circulation. Therefore, 11" tall, 1 L water bottles were used as filler material since the circular shape generates openings between the bottles to allow for more uniform airflow. Seventy percent of the net usable volume was not able to be filled with the water bottles due to their dimensions, and so the largest net usable volume was used. The net usable volume for the case is reported as 83.7 cu ft but calculated at 92.7 cu ft in which a maximum 1056 bottles will be able to be stored, yielding 44.8 % of the internal volume (40.5 % if calculated) with product simulators.

Filler bottles were organized in a 7 x 31 pattern on each pair of shelves except those that contained product simulators. This was the maximum number of bottles that fit between the product load lines on each shelf. Since the bottles nearly reached the full height of each shelf, shelves containing product simulators had six bottles removed to fit the simulators at the front and rear of the shelf, as shown in Figure 8. On the bottom deck, which had more depth than the shelves, 9 bottles were fit across, and so a 2x6 pattern between simulators was used instead of the 2x4 pattern used on the shelves as shown below.

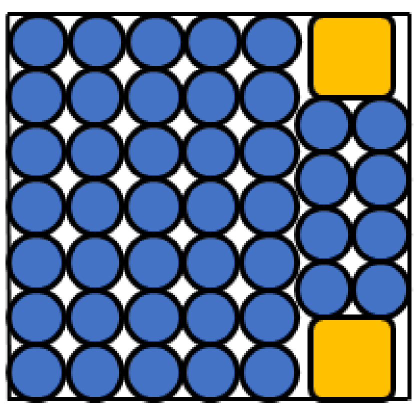


Figure 8. Diagram of Filler Material (Blue) and Product Simulator (Yellow) Configuration on an Example Right-Side Shelf

A diagram modified from ASHRAE 72-2018 of the simulator locations with filler material is shown below in Figure 9 [2]. Simulators are to be placed above filler material on the bottom deck and below the filler material on the shelves. Since bottles were used as filler material, the bottom deck simulators were instead stacked on top of two additional simulators not containing thermocouples to match the height of the bottles.

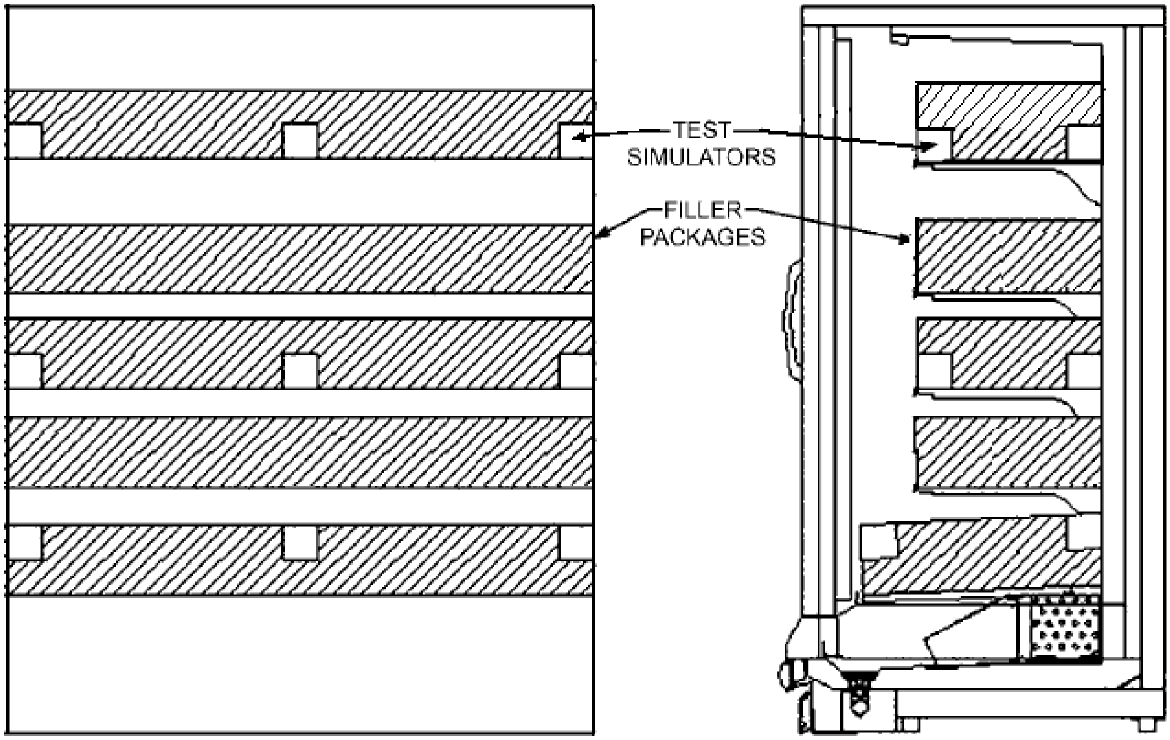


Figure 9. Location of Product Simulators with Respect to Filler Material

(Image modified from ASHRAE 2018 [2])

Chamber Condition Instrumentation/Control

To monitor and maintain conditions within the environmental test chamber, an “ambient measurement pole” was mounted 24” from the front of the case. ASHRAE standards require placement at 36” as shown in Figure 7, however the limited space in the environmental chamber required placing the test pole closer. Measurements were collected at different heights along the pole specified by T_A and T_B. Location T_A is 5.9" above the top edge of the case air curtain discharge, and location T_B is at the height of the geometric center of the air curtain (53” from the floor, 58.9” below T_A). Location T_A was fitted with both a thermocouple probe and a dew-point hygrometer probe. Location T_B was fitted with only a thermocouple probe. The same model 1/16” T-type probes used to measure product simulators were used to measure DBT at T_A and T_B. An EdgeTech DewTrak II DPS3 model chilled mirror dew-point hygrometer was used with ± 0.20 °C (± 0.40 °F) accuracy to measure dewpoint at T_A. The hygrometer probe pulled air through a hose wrapped in an insulated electrical heating tube (to prevent internal condensation) at a flowrate of 3 – 5 fpm per hygrometer manufacturer specifications. For each condensing unit evaluation, the dry-bulb temperature (DBT) at T_A was maintained as close as possible to 24.0 ± 2 °C (75.2 ± 1.8 °F). The DBT at T_B was attempted to be maintained a temperature gradient of ≤ 1 °F/ft from T_A which translates to a range of 24.0 ± 3.95 °C (75.2 ± 6.71 °F). The dew point temperature (DPT) at T_A was maintained within a temperature range of 15.3 ± 2.10 °C (59.6 ± 3.76 °F). The upper and lower limits for each of these three ambient measurements is listed below in Table 4 for the three environmental conditions evaluated.

Table 4. Environmental Condition Setpoints and Upper and Lower Limits

T_A DBT	T_A DBT Upper	T_A DBT Lower	T_A DPT	T_A DPT Upper	T_A DPT Lower	T_B DBT Upper	T_B DBT Lower
24.0 °C (75.2 °F)	25.0 °C (77.0 °F)	23.0 °C (73.4 °F)	15.4 °C (59.8 °F)	17.4 °C (63.4 °F)	13.2 °C (55.8 °F)	28.0 °C (81.9 °F)	20.1 °C (68.5 °F)

A handheld vane anemometer was placed at the location of T_B during steady state monitoring of each condensing unit to spot-check that airflow was less than 0.1 m/s into the case and less than 0.25 m/s away from the case, and that horizontal/vertical velocities were less than 0.25 m/s. Once baffling and booster fans were installed to the chamber to prevent warm air ejected from the condenser from recirculating in the chamber and entraining into the air curtain, airflow in all three directions at T_B was maintained within required speeds. Airflow horizontally and into/away from the case was 0 m/s, and vertically was 0.1 m/s. To ensure that equal conditions were maintained between the baseline and ECM evaluations, the baffling system and fan speed settings were set up exactly the same. Images of the baffling system set up on the case is shown in Figure 10.

Figure 10. (Left) Baffling System Used to Direct Condenser Rejected Heat, (Center) Booster Fan, (Right) Baffling Discharge to the Chamber Return Grille

Air speed through the baffling system was adjusted using a VFD in order to maintain airflow that would not cause warm air ejected by the condenser fan to recirculate back through the air-cooled baseline condenser. This VFD speed was maintained throughout all evaluations, and was used to maintain equivalent airflow within the chamber when also analyzing the ECM condensing unit, despite containing the liquid-cooled condenser which did not reject heat into the environmental chamber.

The baffling structure was 24" tall above the case and extended 38" deep, flush with the rear of the case. The baffling connected to the circular blower inlet flush with the right side of the case, giving the baffling system a 92" total length across the top of the case. A 21" x 24" square opening on the left side of the baffling system was used as the inlet, aligned with the position of the condenser heat exchanger.

On the right side, the 10”-diameter blower made a tight seal with the baffling. The blower had a 11.75” x 8” rectangular outlet that was directed downward into an extended section of baffling 20” x 19” normal to the direction of airflow, and 18” deep. The blower was a McMaster-Carr model 1963K31 with 230 – 460 VAC and rated to 1500 RPM at ¾ in of water. An Invertek Optidrive E3 IP66 VFD model number ODE-3-220105-1F4B(HP) was used to control the speed of the blower, which was set to a constant frequency of 58 Hz. The bottom section of baffling extending below the centrifugal blower was connected to the chamber air return grille via an approximately 5’ section of 12”-diameter, low-resistance flexduct that connected to a final section of baffling. This final section extended behind the rear of the case to seal over half of the chamber’s return grille. The flex duct connected to this final 12” x 18” section at a point to the right side of the case, where air was forced to the left through a 63” distance up to the chamber return.

Power Measurements, Data Acquisition

Total and component power was measured through six Continental Control Systems WMC-3Y-208-MB model Wattnode power meters. A “meter box” containing each of the Wattnodes was constructed to allow the case to be plugged directly to the box to monitor total plug power while also allowing individual current transformers (CTs) to be connected to monitor component consumption. Five Accu-CT ACTL-0750 model CTs were instrumented to each of the three refrigerated cases’ components. For both condensing units, two 50 A CTs were clamped to the wiring to each case’s total plug power and compressor due to their 200-230V/1Ph, and two 5 A CTs were clamped around the three evaporator fans and lighting since both were 115V/1Ph. On the baseline case, two 20 A CTs were clamped to the 200-230V/1Ph condenser fan, and a 5 A CT was clamped to the 115V/1Ph controller. On the ECM condensing unit, no condenser power was measured, and the 200 – 230V/1Ph controller was clamped by two 5 A CTs. The CT power measurement accuracy was ± 0.5 %. An image of the constructed meter box is shown in the left image of Figure 11. Energy was calculated by integrating measured power consumption.

NREL’s local data acquisition system was used to record measurements and control the FCM and baffling blower VFD. The data acquisition system recorded all measurement data at a sampling rate of 1 Hz, and averaged outputs across 1 minute. Thermocouples, voltage inputs (e.g., the dew-point hygrometer, pressure transducers), current inputs (the condensate weigh scale, liquid coolant flow meter) and digital output wiring (e.g., FCM pump and blower VFDs, FCM valves) were connected to terminal panels situated throughout the evaluation laboratory (Figure 11, middle image). Power measurements from the Wattnode meter box were supplied via Modbus through an RJ50 cable that was connected to a data acquisition Modbus interface (Figure 11, right image). Within the data acquisition software’s user interface, separate power measurements at each Wattnode phase were selected from different Modbus registers.

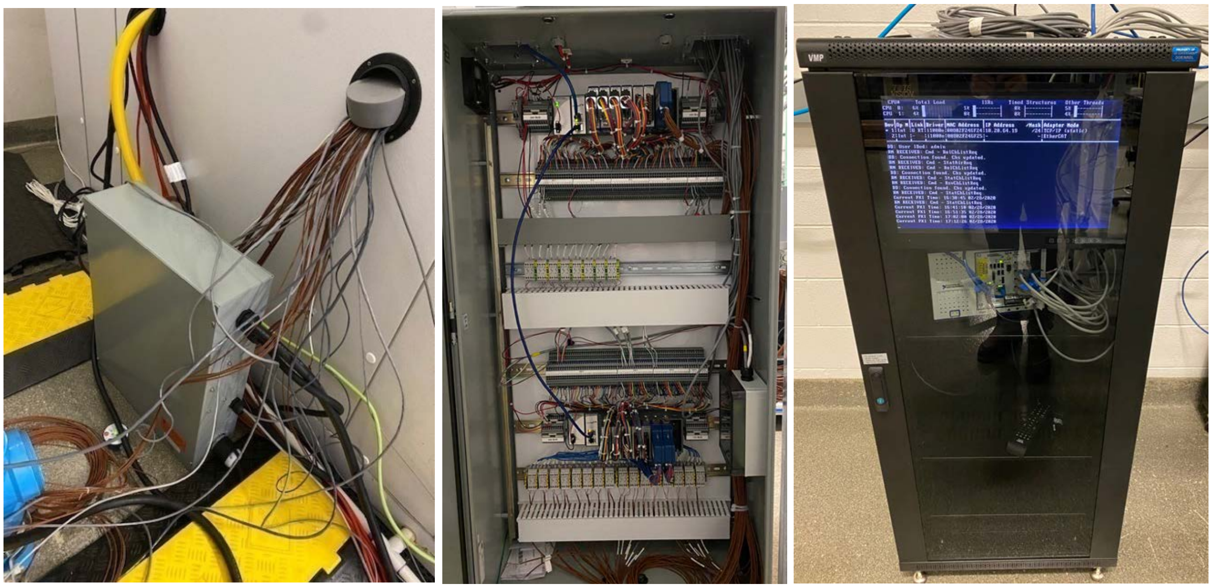


Figure 11. (Left) Wattnode "Meter Box" to Measure Component Power, (Middle) Data Acquisition System Terminal Panel, (Right) Communication Interface for Data Acquisition System

Air Temperature Measurements

In addition to the product simulator and chamber ambient temperature measurements, five other types of interior air temperature measurements were recorded. This included evaporator air temperatures at the left, center, and right side of the air curtain discharge grille, the left, right and center of the return grille, both the inlet and outlet of each of the three evaporator fans, and the air temperature at the geometric center of the case interior. These five measurements were recorded using the same model 1/16" T-type thermocouple probes used for the product simulator and ambient measurements. A diagram of the case showing the direction of air flow from the evaporators to the air curtain, and the location of each of the air temperature measurements, is shown in Figure 12. The measurements are as follows:

1. Air Curtain Return
2. Evaporator Inlet
3. Evaporator Outlet
4. Interior Centroid
5. Air Curtain Discharge

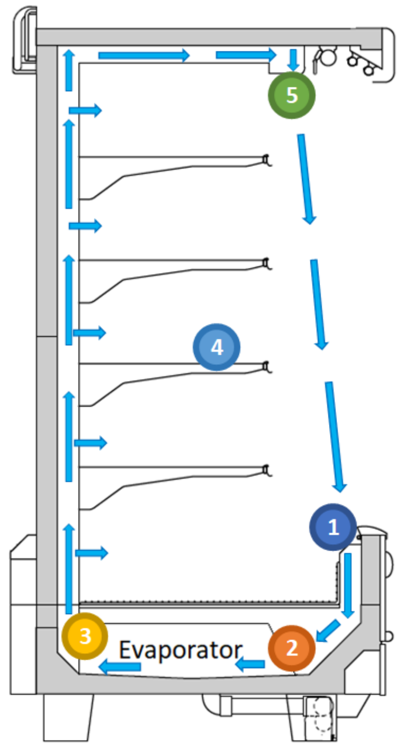


Figure 12. Air Temperature Measurement Locations

Refrigerant Temperatures, Pressures, and Charging

Refrigerant temperatures (mounted on the surface of pipes and insulated) and pressures were recorded to aid NREL engineers in understanding case operation. Omega engineering model SA1-T-SRTC Type-T surface temperature thermocouples with ± 0.90 °F accuracy were used for collecting refrigerant temperatures. Omega brand model PX309-1KG5V multimedia pressure transducers with ± 0.25 % accuracy were used to measure pressure along the liquid and vapor refrigerant lines at the condenser outlet and compressor suction. Only surface temperature thermocouples on the outside of the refrigerant piping wrapped in insulation were used, and not thermocouple taps. Pressure sensors were also only installed at service valves. Since the purpose of this project was to assess the energy consumption of these technologies as they are provided commercially, it was critical to avoid any instrumentation that would tamper with the refrigeration system in a manner that could affect performance. Due to conductive resistances in the piping, surface temperatures cannot be considered an accurate representation of actual refrigerant temperatures.

Refrigerant temperature measurements were collected at six locations on the case and condensing unit. At the condensing unit, these were located at the compressor suction line, the compressor discharge, and the condenser outlet. Below the case in the evaporator panel, refrigerant temperature measurements were collected at the evaporator outlet, the expansion valve inlet, and the evaporator inlet/expansion valve outlet. Images of the surface temperature measurement locations on the baseline unit are color-coded in Figure 13. The left image shows the baseline top-mount condensing unit and the right image shows the evaporator and TXV below the case bottom deck panel.

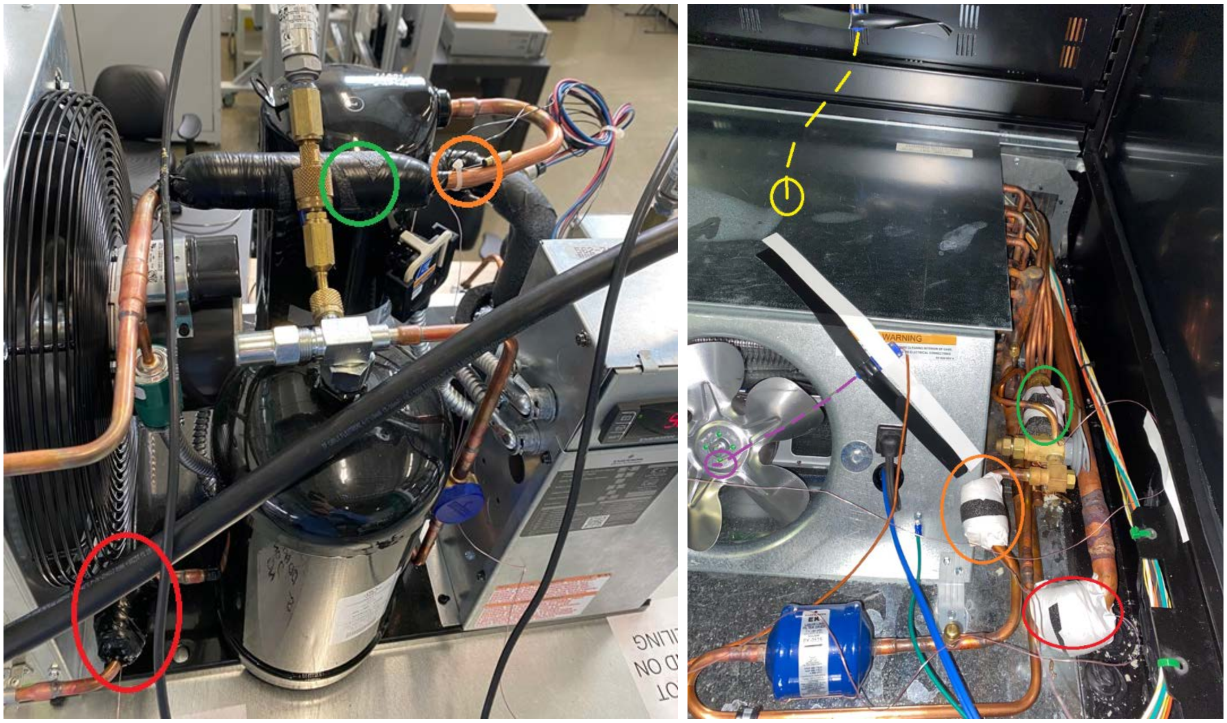


Figure 13. Refrigerant Surface Thermocouple Locations

RIGHT: Condenser outlet (red), Compressor suction (orange, rear), Compressor discharge (green); LEFT: Expansion valve inlet (orange), evaporator inlet (green), evaporator outlet (red). Purple and yellow lines show the locations of probes used to measure evaporator inlet and discharge air temperatures, respectively.

Refrigerant measurements were taken at equivalent locations on the ECM condensing unit as in the left image in Figure 13. Pressure transducers were tapped at the service ports located at the compressor suction and condenser discharge. On both condensing units, the pressure taps at the condenser discharge line were made at a service port on top of the refrigerant receiver. The compressor suction pressure transducer was tapped at a service port on a refrigerant accumulator on the baseline condensing unit, however no accumulator existed on the ECM unit and instead, a service port was found in the same location.

Initially, only 7 lb of R448-a refrigerant was charged to each case's condensing unit. Once refrigerant temperatures and pressure instrumentation was installed, they were used to monitor case performance and adjust the quantity of refrigerant charged until performance was achieved according to parameter limits subscribed by the manufacturer. The manufacturer requested that the subcooling temperature differential at the condenser should be calculated and monitored to ensure appropriate operation and to charge the appropriate amount of refrigerant. Condenser subcooling was calculated using the equation listed below, where the bubble point was calculated from the saturation table for R448a at the measured pressure at the condenser outlet:

$$\Delta T_{subcooling} = T_{bubble\ point} - T_{condenser\ outlet}$$

The manufacturers requested for the baseline condensing unit, that the subcooling temperature difference should have a minimum value of 0 °C when the compressor cycled on. This required increasing the refrigerant charge to 8 lb. For the ECM case, the manufacturers requested a 5 °C subcooling, however this was measured using the ECM controller sensors and not the measurements used in this evaluation. This required the refrigerant charge to be adjusted to 7.5 lb. Alternatively, the manufacturers requested monitoring evaporator superheating to ensure adequate charge and operation of the unit. Evaporator superheating was calculated similarly using the equation listed below, where the dew point was calculated from the saturation table for R448a at the measured pressure at the compressor suction:

$$\Delta T_{superheat} = T_{evaporator\ outlet} - T_{dew\ point}$$

Refrigerant temperatures and pressures, as well as these calculated temperatures are reported in Appendix C.

2.2 FCM Controls and Evaluation Conditions

In order to supply coolant to the condenser at a controlled temperature and flowrate, laboratory infrastructure was modified so that piping was connected from a fluid-conditioning module (FCM) to the environmental chamber. The ECM condenser can require use either distilled, deionized, or a glycol solution with water, so distilled water supplied by the FCM was used. The FCM has been pre-fabricated for laboratory use by related projects and is shown below in Figure 14. Hot water and chilled water lines from a boiler and chiller in a nearby lab are diverted from the ceiling in the upper-right corner. These lines pass through a series of controlled valves that regulate their flowrate through two heat exchangers to condition the FCM water flowing to the refrigerator case. A 5-HP Price CD150BF centrifugal pump circulates the FCM water through another series of Belimo ARX24-EP valves to regulate flow either through the heat exchangers or bypass lines in order to maintain temperature at a setpoint value specified in the data acquisition system. To control flow rate, the pump is connected to an Invertek Optidrive P2 variable frequency drive module with model number ODP-2-24050-3HF4Y-TN. Power consumption by the pump and VFD will be measured, however the pump is sized for multiple laboratory projects and is oversized for the refrigerator application seen here. In a normal supermarket setting, a pump of this size would not be used for refrigerator case condenser cooling and therefore the FCM pump power will not be reported here. Instead, an estimated pump power is reported based on the hydraulic power at a range of estimated pump efficiencies.

The ECM condenser requires a potential flowrate of 5 – 10 gallons per minute, however has a liquid control valve that regulates its own flowrate in order to maintain a temperature difference across the condenser. To monitor flowrate and temperature at the inlet and outlet of the condenser, a case liquid measurement stand was constructed in the environmental in Figure 15, containing a Coriolis flow meter, thermocouples, and pressure transducers. The apparatus contains pressure taps at the inlet and outlet to monitor pressure drop, and a bypass line to stabilize flow temperature. A list of the sensors used in this measurement apparatus and their accuracies are listed in Table 5. The condenser maxes out at a pressure drop beyond 10 psig across the condenser, however maxing out the FCM VFD and configuring the FCM valves in a manner to maximize pressure did not yield this pressure drop at the case due to too great of losses along the coolant piping across the laboratory.

Figure 14. Pre-Fabricated Fluid Conditioning Module (FCM)

Figure 15. Case Liquid Measurement Stand

Table 5. List of Measurement Sensors Used to Control/Monitor FCM Water Conditions

MEASUREMENT:	Brand/Model	Type	Accuracy
Flowrate to Condenser	Emerson-Micromotion/CMF050M322N2 meter, 2700R12B transmitter	Coriolis flow meter	± 0.05 %
Inlet/Outlet Temperature	Martin/K28G-006-00-4	1/8" T-Type Thermocouple Probe	± 0.50 °C (± 0.90 °F)
Inlet Pressure	Ashcroft/G2 UPC	0 – 50 PSIG liquid pressure transducer	± 0.50 %
Outlet Pressure	Omega/PX309-050GI	0 – 50 PSIG liquid pressure transducer	± 0.25 %
Flowrate across Bypass	McMaster Carr/4215K75	0 – 10 GPM High-Temp Analog Flowmeter	± 4.00 %

To measure estimated pump power, the hydraulic power was calculated from the product of the measured Coriolis flowrate and pressure at the condenser inlet. Then, the pump power was calculated from the hydraulic power according to the below equation, where Q is flowrate, Pr is pressure, and η is pump efficiency. Estimated pump daily energy was calculated across a range of typical pump efficiencies ranging between 40 and 80 %. Transient pump power was measured at three efficiencies chosen across this range at 40 %, 60 %, and 80 %. This maximum efficiency was selected assuming a motor efficiency below 90 % and pump system efficiencies below 90 %. The pump power was estimated using the flow work formula for hydraulic power (below). The pump energy was then estimated by integrating the pump power across each 24-h evaluation at each efficiency.

$$P_{\mathrm{Pump,est.}} = \frac{Q * (\mathrm{Pr_{out}} - \mathrm{Pr_{in}})}{\eta}$$

The coolant's liquid inlet temperature will affect the condensing temperature, thereby, the case performance and energy consumption. Therefore, the inlet water temperature to the condenser was varied across the operating range of the Emerson CIMS unit to investigate its impact on the power and energy consumption. The following three water inlet temperatures which represent the most common scenarios across a range of typically available temperatures were used in this evaluation:

1. 55 °F – represents a closed loop application where water is mechanically chilled prior to entering the condenser

2. 80 °F – represents scenarios including open loops with high ground water temperature or closed loops with cooling towers

3. 108 °F – represents an extreme scenario where the saturated condensing temperature of the water-cooled unit is approximately the same as the air-cooled baseline unit

Future research would benefit from assessing energy savings at other condenser inlet temperatures. However, this study was limited to three temperature scenarios and so only the most relevant temperatures were selected across the condenser's capable range. The FCM controls temperature by regulating the flow of water through two heat exchangers designed to raise and lower temperature by interfacing with a boiler and chiller supply water line, respectively. When calling for heating, a PID controller opens a valve to bypass more flow through the boiler heat exchanger while closing a valve that bypasses flow through the chiller heat exchanger. Although this PID controller was fine-tuned to optimize fluid temperature stability, some fluctuation persists. Therefore, a mean temperature limited to within an approximately 5 % variation was maintained for each condition.

The CIMS unit can operate at a maximum inlet temperature of 42.2 °C (108 °F) and a minimum temperature of 4.40 °C (40.0 °F). The saturated condensing temperatures were calculated as the mean of the bubble point and dew point, which were calculated from the R448a saturation tables from the pressure measured at the condenser outlet. This was measured to be 50.0 °C (122 °F) following baseline evaluation. Since this saturated condensing temperature was found to correspond to a 108 °F water inlet temperature, the upper limit of the CIMS unit was used as the test matching the conditions of the baseline condensing unit. A midpoint temperature of 26.7 °C (80.0 °F) was selected to best represent a typical city main water or water tower temperature [9]. The lower limit temperature of the CIMS was however unable to be safely achieved using the available fluid conditioning system, and therefore a low temperature of 12.8 °C (55.0 °F) was used which is representative of chiller or winter water line conditions. Each 24-h evaluation was conducted at least twice at each inlet temperature until stable and approximately equivalent conditions were maintained between evaluations.

3 Results Summary

The baseline unit consumed 47.0 kWh/day. At 80 °F water inlet temperature, which is the most representative temperature, the ECM consumed 30.6 kWh/day, and at the lowest setpoint (55 °F water inlet temperature), the ECM consumed 21.6 kWh/day, constituting savings of 34.9 % and 54.0 %, respectively. This does not include estimated pump energy. At the highest extreme condenser water inlet temperature of 108 °F (matching the baseline saturated condensing temperature), the ECM consumed 48.0 kWh/day, which is 2.10 % higher than the air-cooled baseline. Experiments were conducted repeatedly until equivalent results converged across at least two 24-h assessment cycles. The total and component daily energy consumption is reported as an average of each individual assessment. The mean daily energy consumption of the baseline and ECM at each water inlet temperature is provided below in Figure 16 (not including estimated pump energy). Tabulated results are shown in Appendix E.

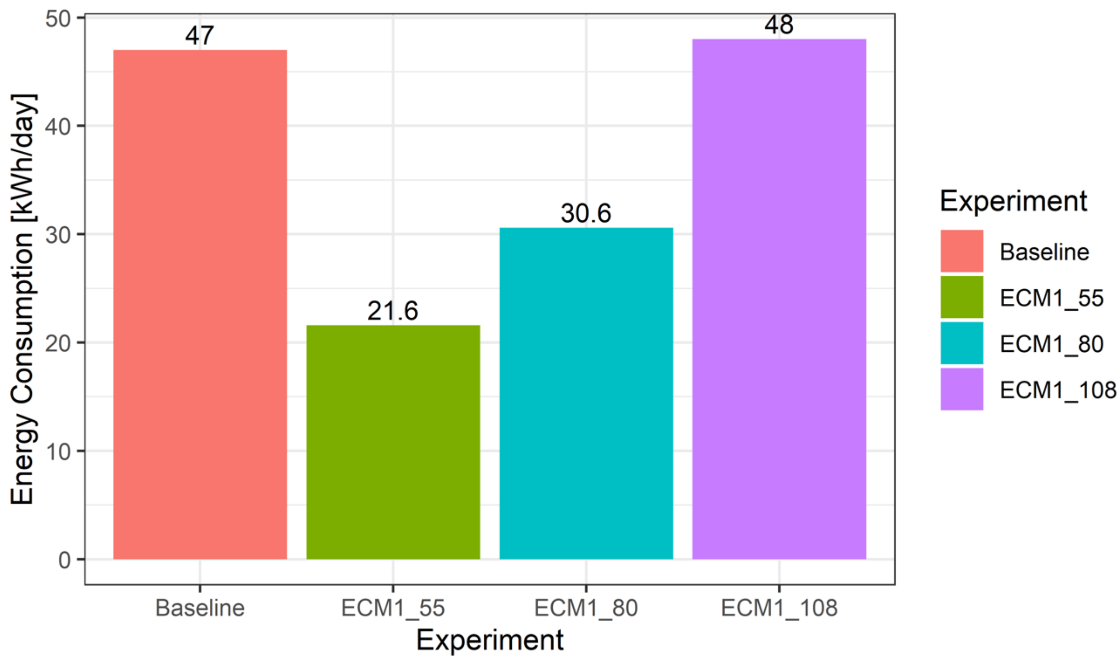


Figure 16. Comparison of Total Daily Energy Consumption

For an 80 % efficient pump, the estimated pump energy was 0.18 kWh/day at 55 °F, 0.12 kWh/day at 80 °F, and 0.79 kWh/day at 108 °F. The pump energy was significantly higher at the highest temperature due to the additional flowrate (and hydraulic power) required to reject heat, as seen in Appendix D. Adding the pump to the total energy consumption, the total savings was altered by 53.6 % (55 °F), 34.5 % (80 °F), and -3.99 % (108 °F). Assuming a very low-efficiency (40 %) pump, this would consume only 0.37 kWh/day at 55 °F, 0.24 kWh/day at 80 °F, and 1.57 kWh/day at 108 °F, which would only reduce energy savings to 53.2 % (55 °F), 34.3 % (80 °F), and -5.65 % (108 °F). The pump energy at other efficiencies is provided in the following sections, and tabulated results are provided in Appendix E.

The mean daily power consumption during each experiment when the compressor was on is provided in Figure 17 (not including mean estimated pump power). Error bars indicate standard deviation across compressor on-cycle time, excluding outliers during startup. At the highest condenser liquid inlet temperature of 108 °F (closest to matching the baseline saturated condensing temperature), the ECM consumed an average 2,288.7 W during compressor on-cycling, whereas the baseline case consumed 2,461.7 W, constituting a 7.00 % reduction in power consumption. At the midpoint condenser liquid inlet temperature (80 °F), the ECM consumed 1,457.7 W, constituting a 40.8 % reduction in power consumption, and at the lowest condenser liquid inlet temperature setpoint (55 °F), the ECM consumed 1,021.4 W, constituting a 58.5 % reduction in power consumption. Tabulated mean power results are provided in Appendix E. The component energy and mean power consumption (during compressor on-cycling) is provided in the following sections. Transient 24-hour power consumption data for the total case and component power is also provided. Results were approximately equivalent across individual 24-h evaluations at the same liquid inlet temperature conditions, therefore transient results are only provided for the most recent assessment conducted.

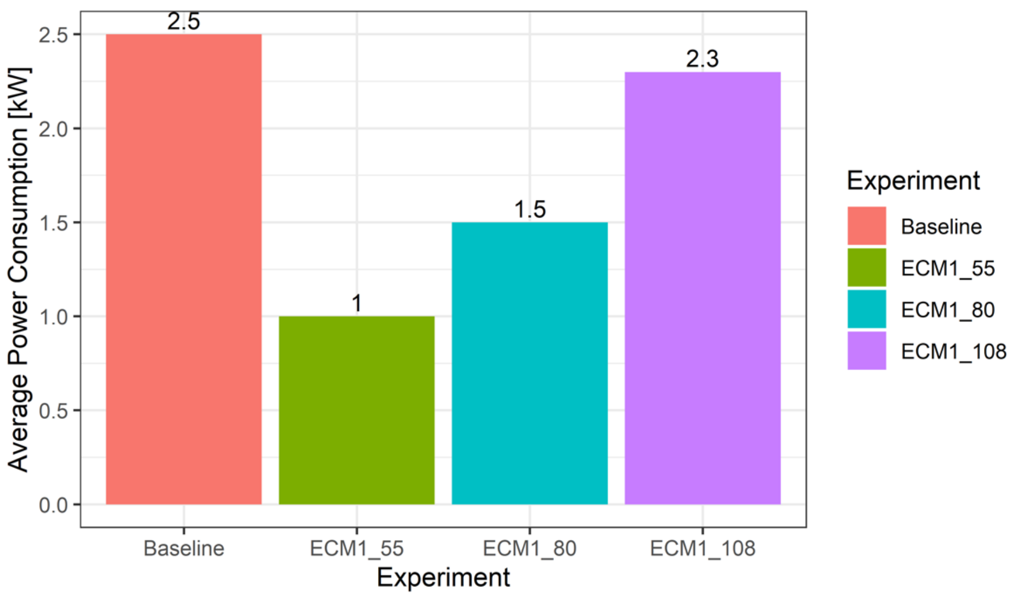


Figure 17. Comparison of Average Power Consumption

For an 80 % efficient pump, the average estimated pump power was 8.72 W at 55 °F, 5.79 W at 80 °F, and 37.3 W at 108 °F. The pump power was significantly higher at the highest temperature due to the additional flowrate required, as seen in Appendix D. Added to the total mean power, this resulted in a mean power reduction of 58.2 % (55 °F), 40.6 % (80 °F), and 5.51 % (108 °F) from the baseline including pump power. Again, if assuming a pump on the low end of the efficiency range (40 %), mean power savings would only be reduced to 57.8 % (55 °F), 40.3 % (80 °F), and 4.00 % (108 °F). The pump power at other efficiencies is provided in the following sections, and tabulated in Appendix E.

One of the parameters that was measured in this project was the total mass of condensate that the refrigeration system removed from the ambient air inside the environmental chamber. Mass of condensate was used as another check to ensure the latent load removed by the display case under each experimentation scenario was similar. Since the controlled environment chamber was programmed to maintain equivalent dew point temperature for all experiments, it was expected that the total mass of condensate removed from the system be similar between each evaluation. Figure 18 shows the total daily mass of condensate for each experiment. The total condensate mass generated by the case with the ECM unit was within 7 % of the baseline across individual assessments.

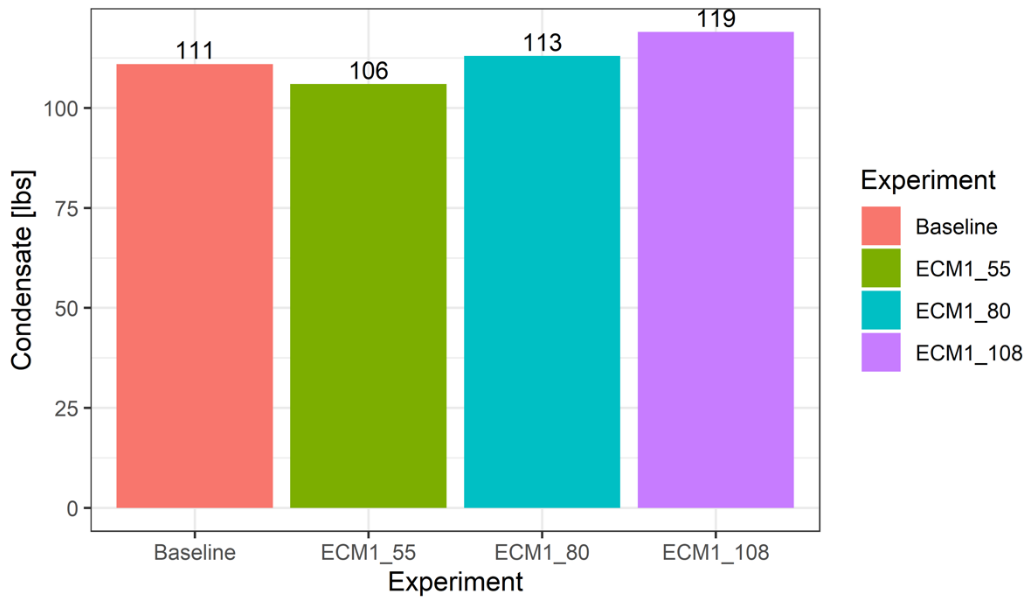


Figure 18. Comparison of Total Condensate Mass

3.1 Baseline Condensing Unit Results

The daily energy consumption, and the mean power consumption of the baseline components is shown in Figure 19. Tabulated results can be found in Table 11. The difference between the total power consumption and the sum of the components (~20 – 40 W) can be accounted for by the power consumed by the Wattnode power meters, as well as losses within the internal circuitry of the power meter box. This is noted in the following figures as P_Other. The power standard deviation was calculated only when the compressor was on for each component.

The compressor was clearly the most dominant energy-consuming component. It consumed most of the energy at 41.6 kWh/day, ranging between 88 % and 89 % across individual 24-hour evaluations. The condenser fan consumed much less energy at only 2.50 kWh/day, or 5 %. The lighting consumed nearly as much energy as all three evaporator fans combined (1.10 and 1.20 kWh/day, respectively), which was only 2 % of the total energy for both. Finally, the controller, which only consumed a few watts, consumed a negligible fraction of the total energy consumption at 0.10 kWh/day.

Compressor cycling was set to cut-in at 3.33 °C (38.0 °F) and cut-out at -1.11 °C (30.0 °F). At the controlled chamber environmental conditions under which the case was evaluated, the compressor cycled on 48 ± 5.7 times throughout each 24-h evaluation. The baseline condensing unit controlled defrost to terminate after 30 minutes, or earlier if the evaporator coil sensor reached 50.0 °F. Sometimes, defrost would terminate earlier than thirty minutes, and sometimes defrost would appear to terminate later than thirty minutes if the defrost period combined with an off-cycle related compressor cut-out. The transient component power by the case with the baseline condensing unit is shown in Figure 20. The frequency of compressor cycles can be observed as well as the duration of the four defrost cycles. The total mean defrost time across each 24-h evaluation was 30.7 ± 0.5 minutes.

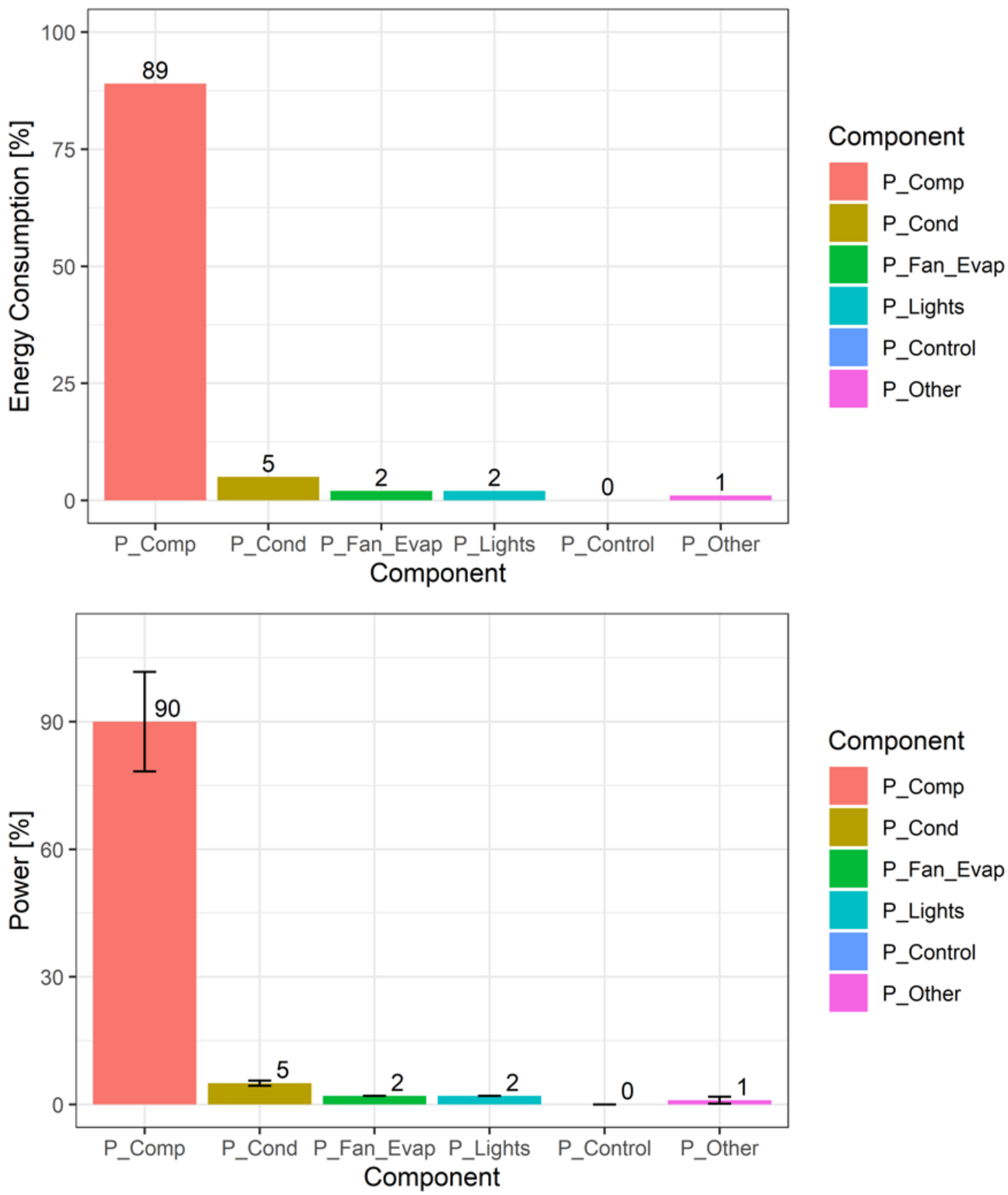


Figure 19. Baseline Component Daily Energy Consumption (kWh/day, top) and Average Power Consumption (W, bottom)

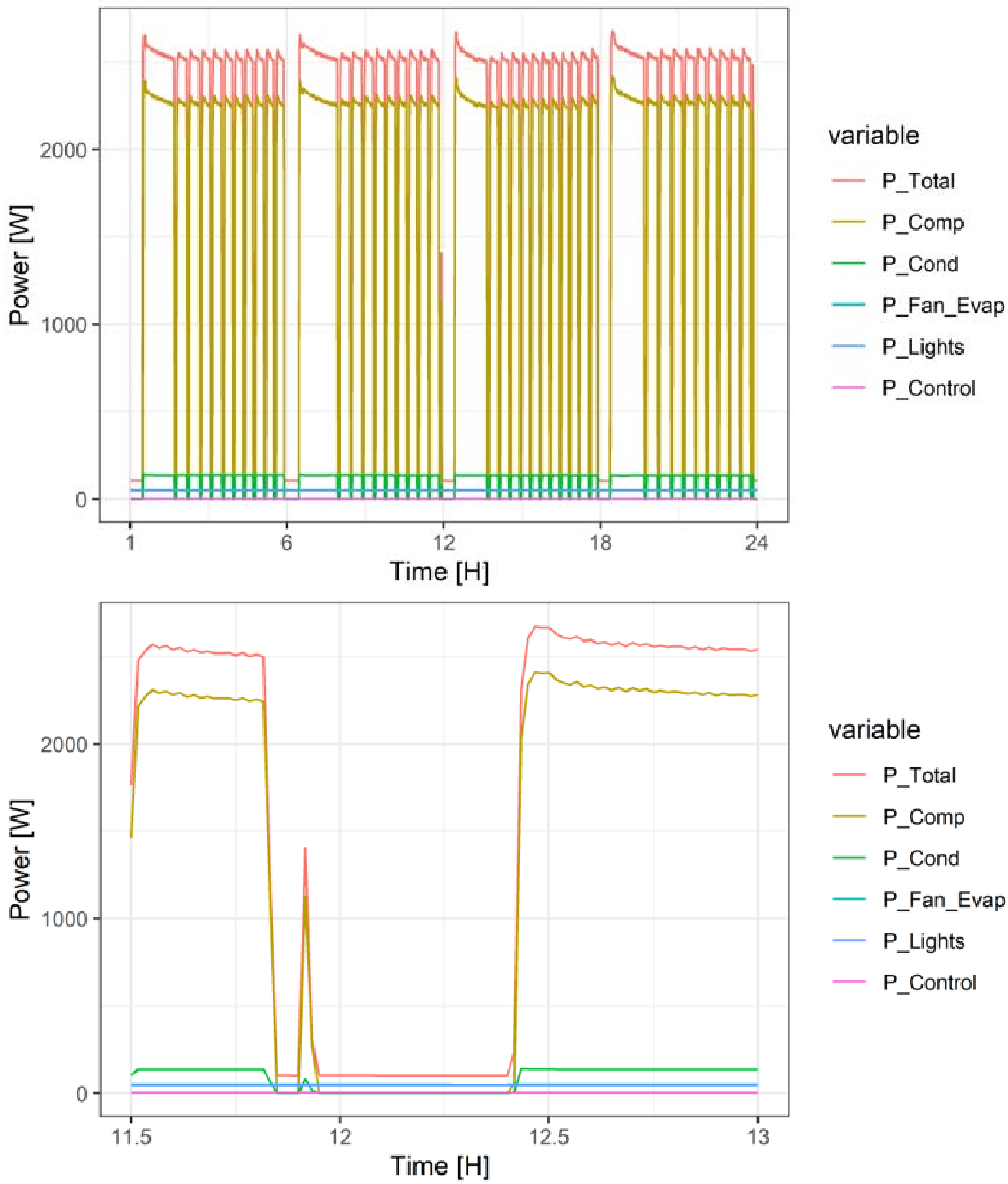


Figure 20. Baseline Total and Component Power Consumption (top: full 24-h cycle; bottom: zoomed-in around hour 12)

3.2 ECM Condensing Unit Results

The daily energy consumption, and the mean power consumption (when the compressor was on), of the case components with the ECM condensing unit is shown in Figure 21, Figure 22, and Figure 23 at each of the three condenser liquid inlet temperatures evaluated. Tabulated results at each of the three temperatures can be found in Tables 12, 13, and 14. The difference between the total power consumption and the sum of the components (~20 – 40 W) can be accounted for by the power consumed by the Wattnode power meters, as well as due to losses within the internal circuitry of the power meter box. The power standard deviation is calculated across the total compressor on-cycle run time.

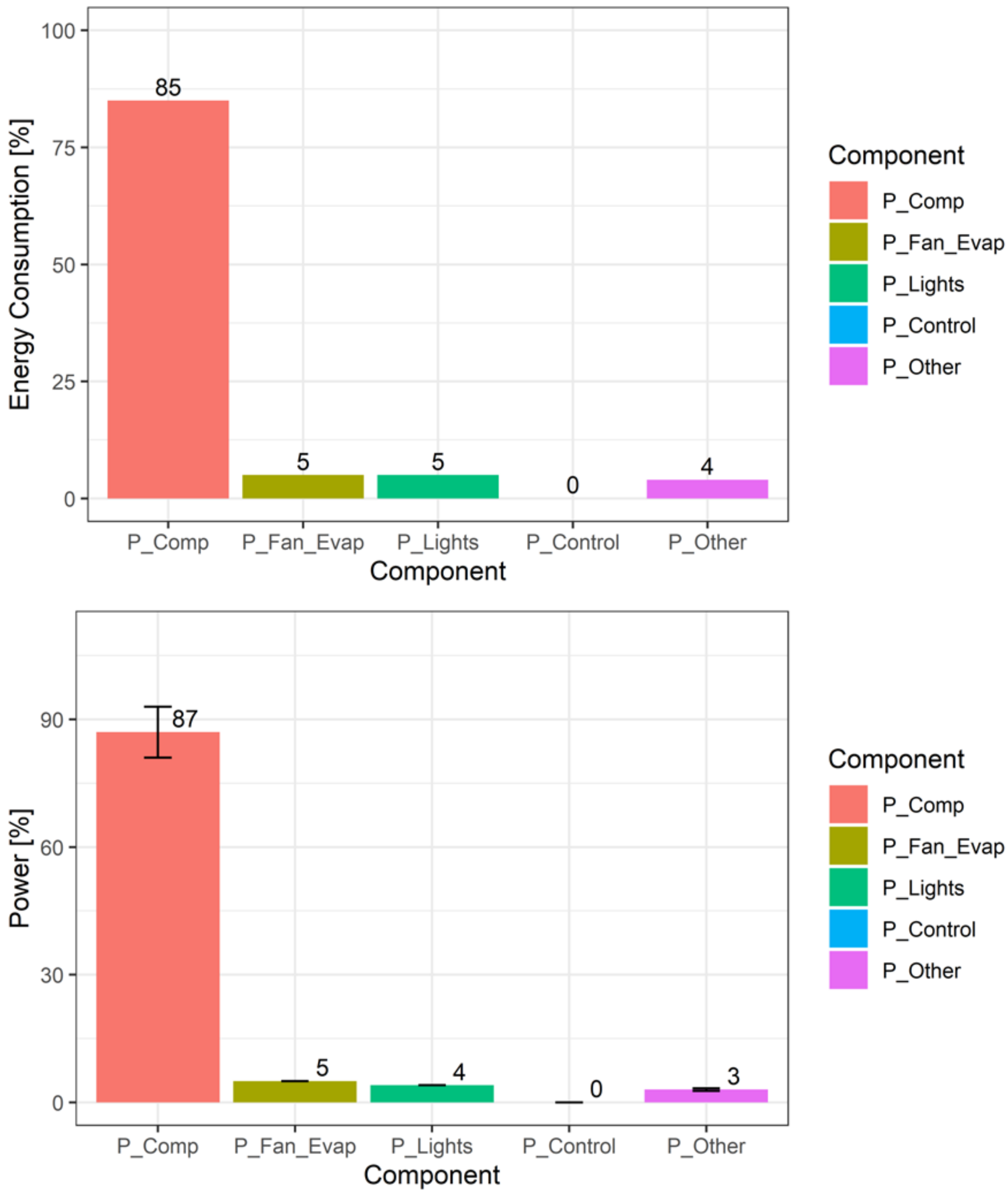
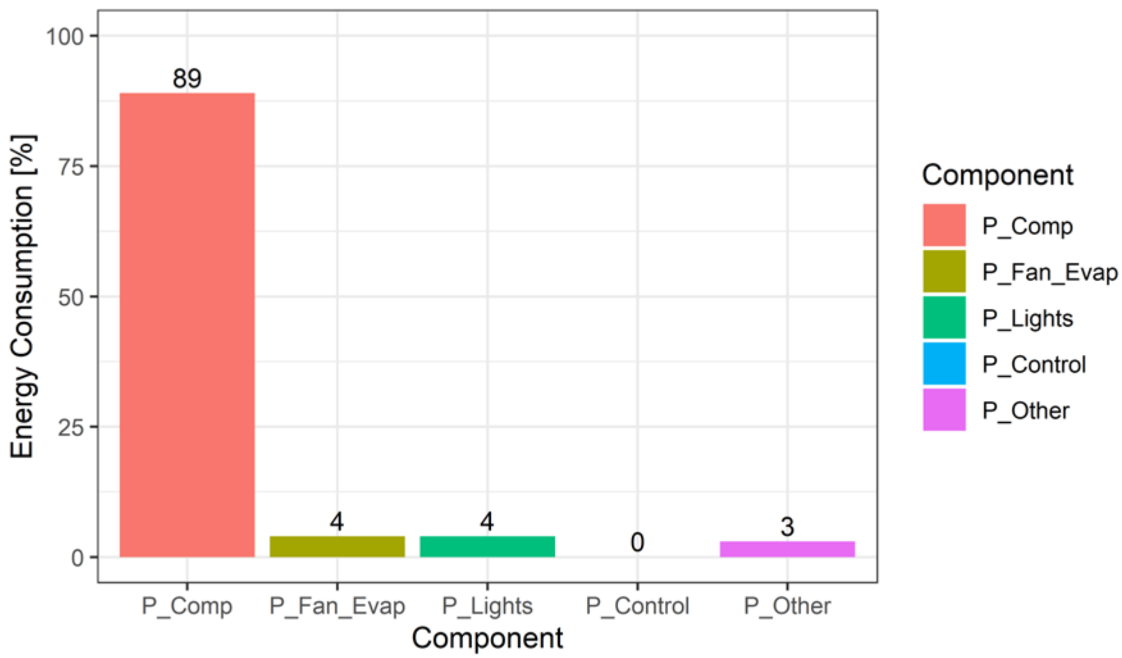


Figure 21. ECM Component Daily Energy (kWh/day, top) and Mean Compressor On-Cycle Power Consumption (W, bottom) at 55 °F Condenser Water Inlet Temperature



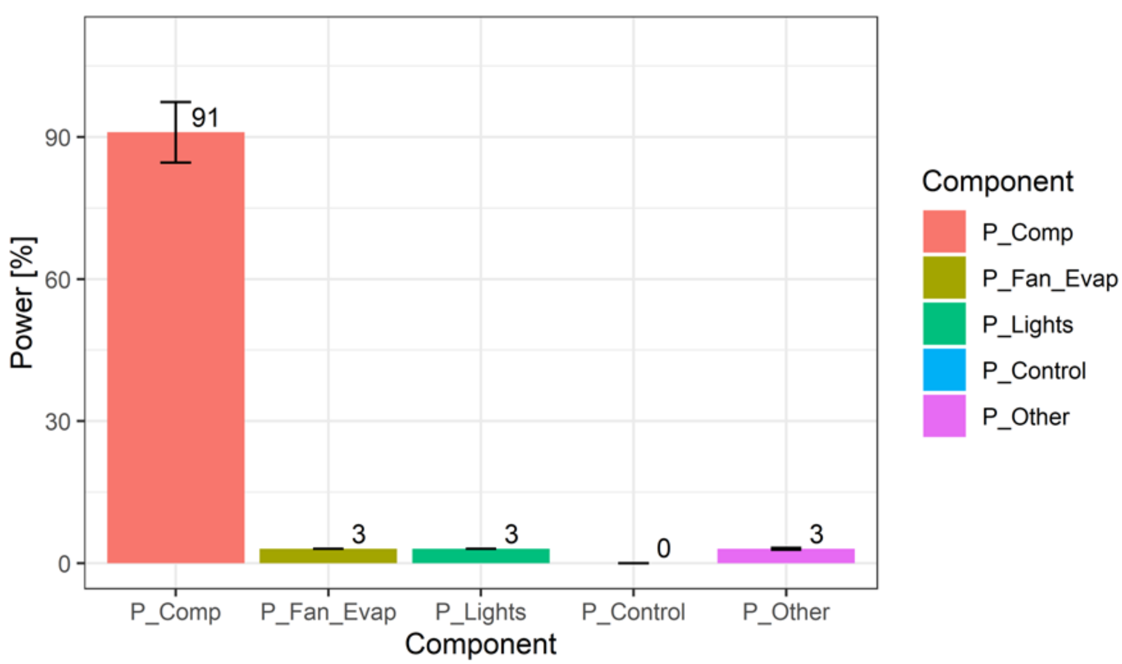


Figure 22. ECM Component Daily Energy (kWh/day, top) and Mean Compressor On-Cycle Power Consumption (W, bottom) at 80 °F Condenser Water Inlet Temperature

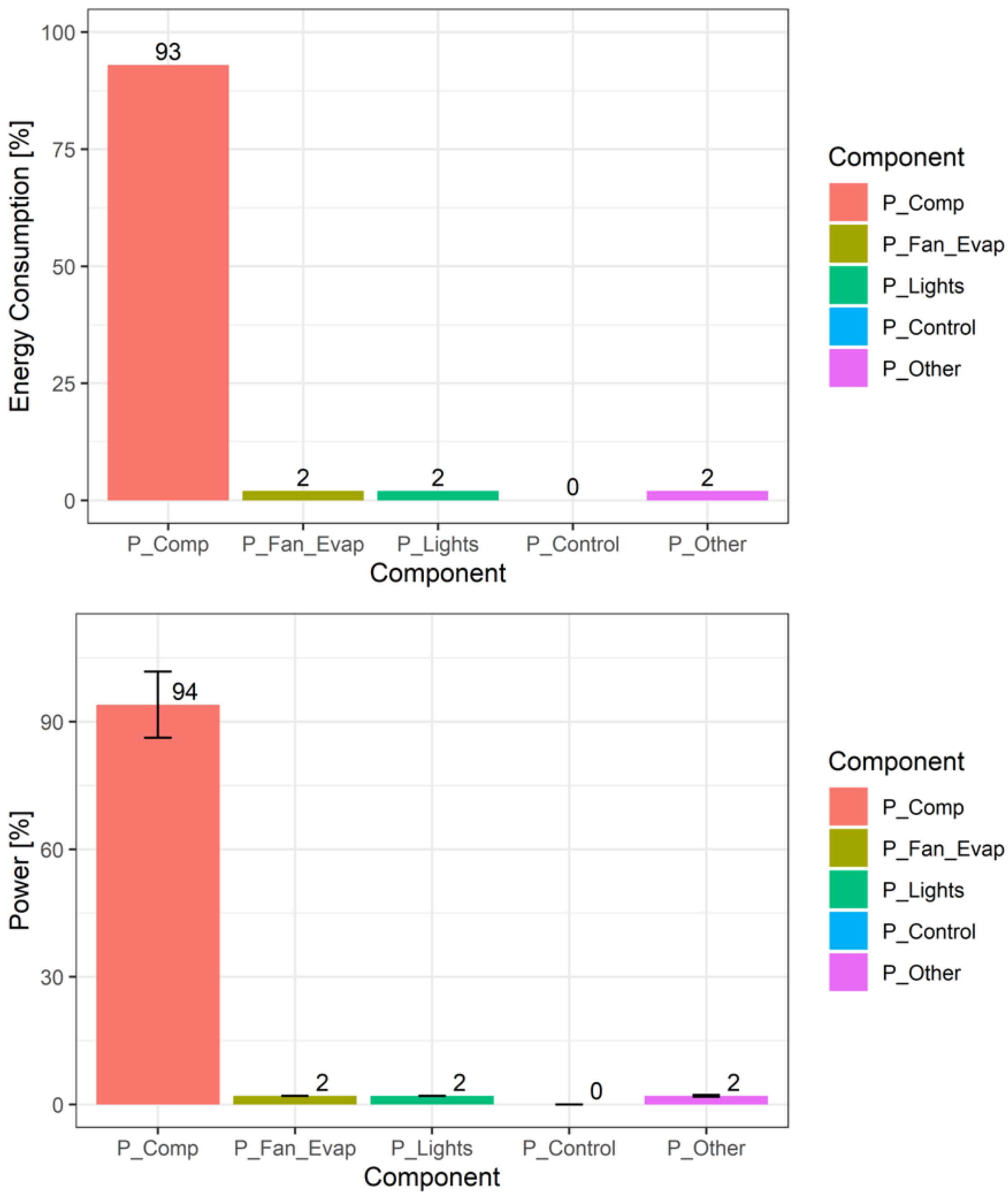


Figure 23. ECM Case Component Daily Energy (kWh/day, top) and Mean Compressor On-Cycle Power Consumption (W, bottom) at 108 °F Condenser Water Inlet Temperature

The compressor consumed the majority of the energy at 18.4 kWh/day at the 55 °F inlet condition, 27.4 kWh/day at the 80 °F inlet condition, and 44.7 kWh/day at the 108 °F inlet condition (between 85 % and 93 % of the total consumption). The lighting and the three evaporator fans consumed approximately the same energy as the baseline case at every inlet condition at 1.10 and 1.20 kWh/day, respectively (between 2 % and 5 % of the total energy across inlet conditions) since they were turned on continuously and not affected by the condensing unit. Finally, the controller, which only consumed a few watts, consumed just 0.10 kWh/day at each inlet condition (less than a percent of the total energy consumption). Since the compressor consumed most of the energy, the effect of water inlet temperatures on the lift required by the compressor was the biggest factor on overall energy consumption.

The energy consumed by the ECM components breaks down the source of differences in total consumption between the baseline case and ECM. Because increasing the water temperature increased the saturated condensing temperature, the compressor consumed more power due to the increased lift which caused the VFD to increase RPM. Compressor cycling did not change since the compressor always stayed on except during defrost. Therefore, energy consumption was almost entirely affected the different inlet water temperatures, which is shown in the percent energy consumption for the compressor. The controller also consumed a slight increase in power and energy when the water temperatures increased.

For all three ECM experiments the compressor cycled off only during defrost cycles throughout each 24-h evaluation. Because the compressor never cycled off except during defrost, the additional defrost cycles compared to the baseline (6 versus 4) further reduced energy consumption. The ECM controlled defrost to initiate every four hours (six times over 24 hours), and terminate at an evaporator coil sensor temperature of 48.0 °F. This defrost termination control was unable to be used at the highest condenser water temperature due to control issues, and therefore the controller was switched to a 32-minute time-terminated defrost control based on the advisement of the controller manufacturer.

The transient component power by the ECM at the lowest condenser water inlet temperature (55 °F) is shown in Figure 24. The transient component power at the middle water inlet temperature (80 °F) is shown in Figure 25 and the component power at the highest water inlet temperature (108 °F) is shown in Figure 26. The frequency of compressor cycles can be observed in the figures as well as the duration of the six defrost cycles. The total mean defrost time across each 24-h evaluation was 33 minutes at the middle condenser inlet water temperature (80 °F), 33 minutes at the lowest condenser inlet water temperature (55 °F), and 32 minutes at the highest condenser inlet water temperature (108 °F) since defrost cycles were terminated by time rather than by temperature.

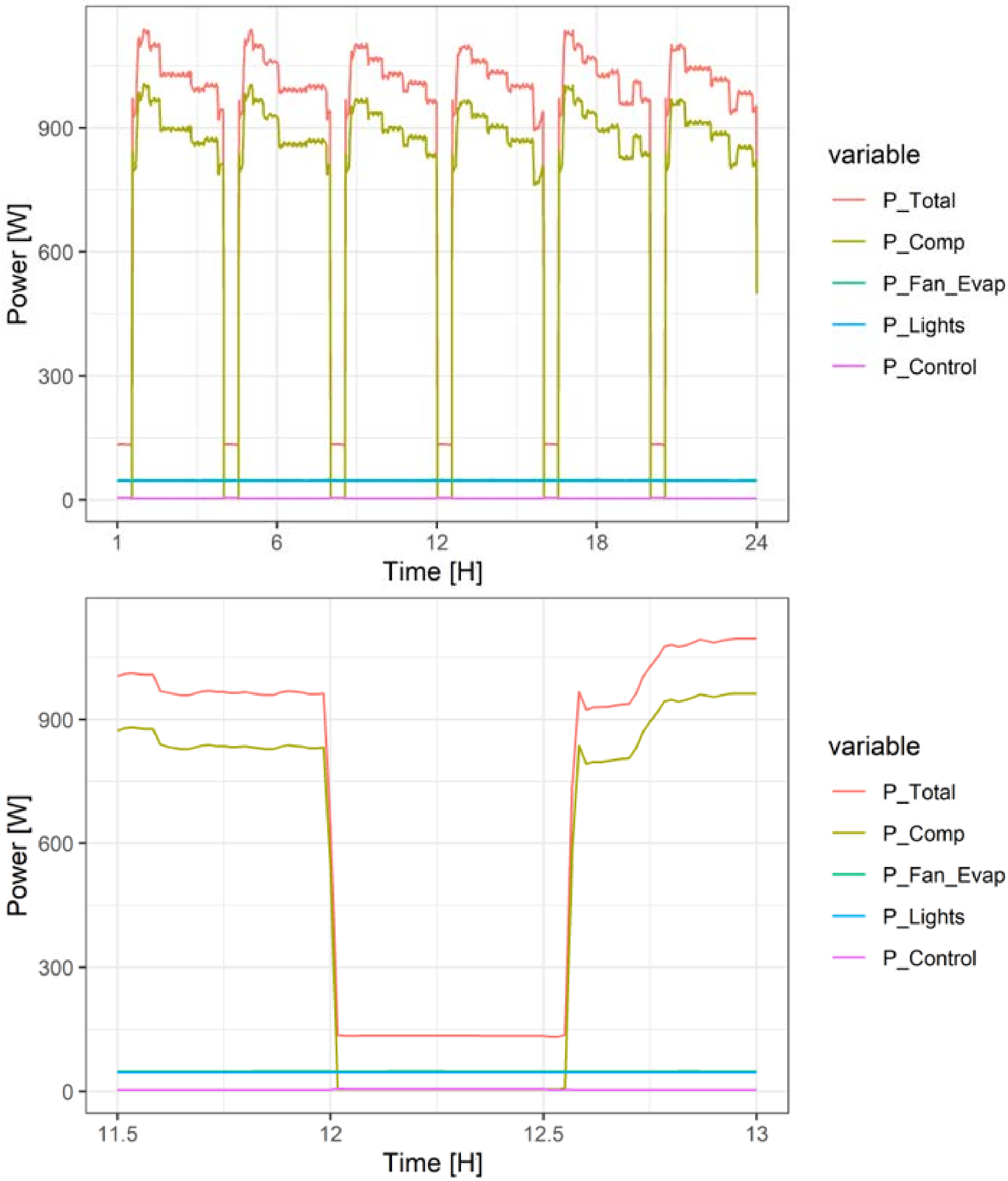


Figure 24. ECM Total and Component Power Consumption at 55 °F Condenser Inlet Water Temperature (top: full 24-h cycle; bottom: zoomed-in around hour 12)

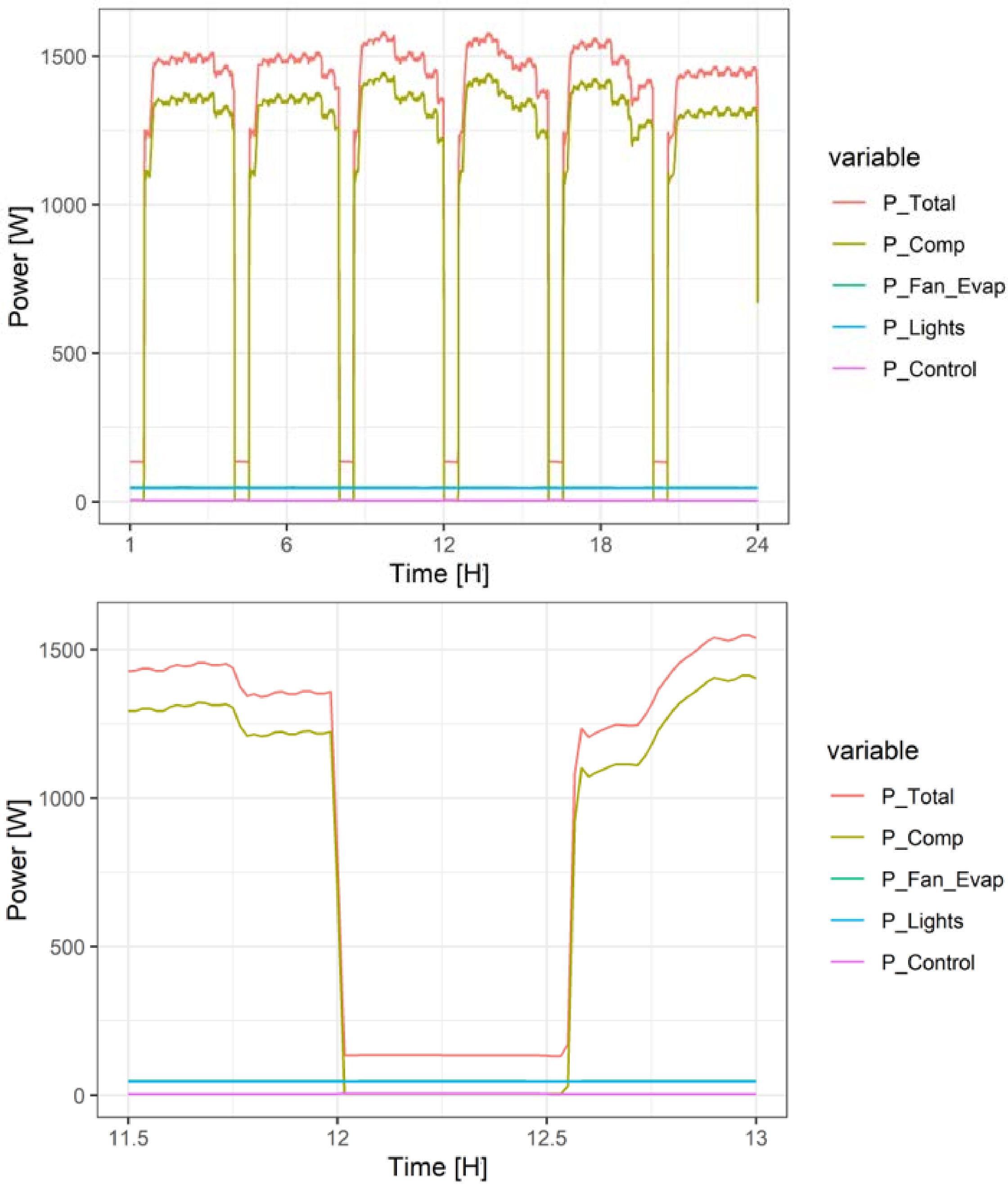


Figure 25. ECM Total and Component Power Consumption with ECM Condensing Unit at 80 °F Condenser Inlet Water Temperature (top: full 24-h cycle; bottom: zoomed-in around hour 12)

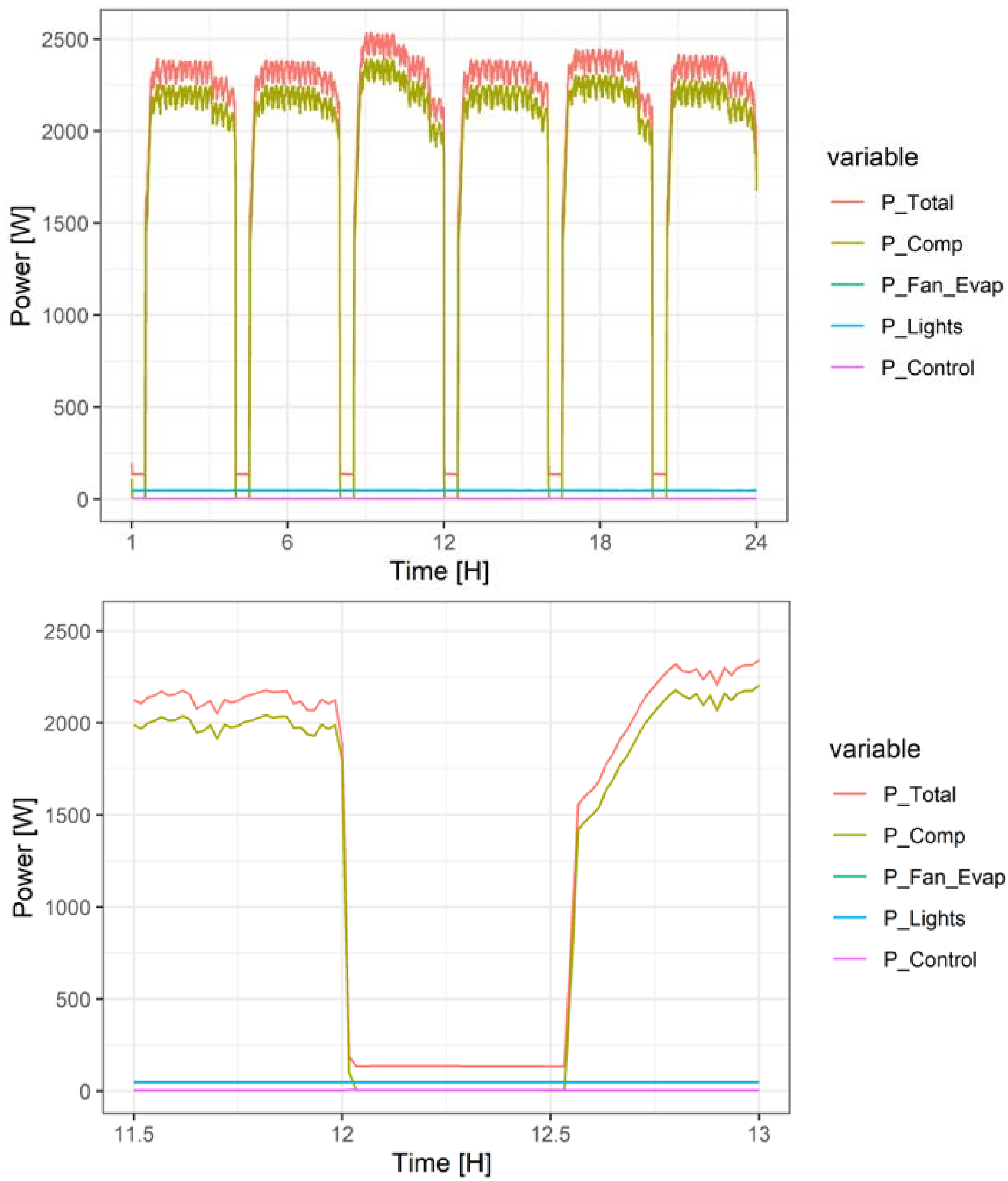


Figure 26. ECM Total and Component Power Consumption at 108 °F Condenser Inlet Water Temperature (top: full 24-h cycle; bottom: zoomed-in around hour 12)

The estimated pump energy over 24 h across a range of efficiencies is shown below in Figure 27. Since the highest inlet temperature experiment was at the edge of the operating envelope for the ECM, a much higher flowrate was required when supplying 108 °F supply temperatures than when supplying 80 or 55 °F temperatures. This caused the hydraulic power and thereby, the estimated pump power, to increase, generating greater overall energy consumption. Based on the flowrates and pressures shown in Appendix D, the transient estimated pump power is shown below for each inlet water temperature condition. The estimated pump power at 55 °F is shown in Figure 28, the power at 80 °F is shown in

Figure 29, and the power at 108 °F is shown in Figure 30. Transient pump power in each figure is provided at each of the three efficiencies evaluated.

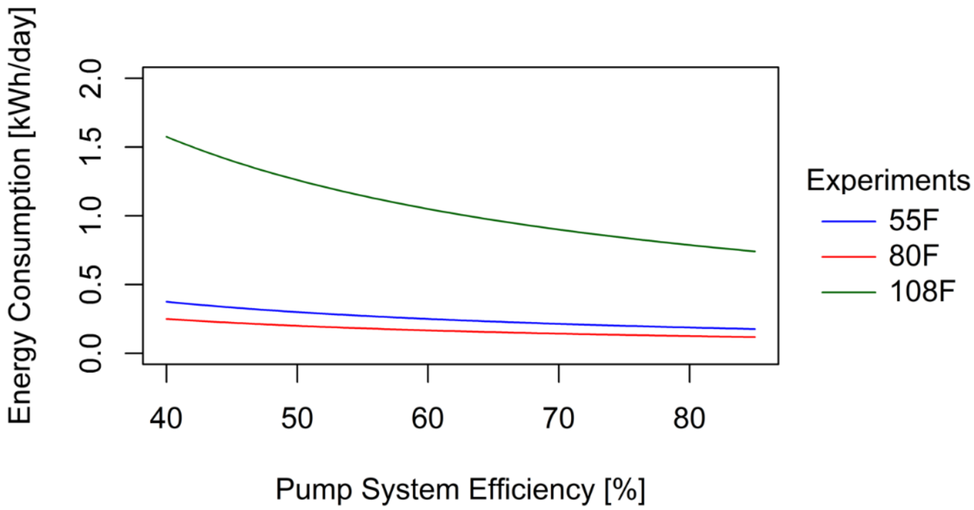


Figure 27. Total Estimated Pump Energy Over 24 Hours vs. Pump Efficiency at Each Condenser Water Inlet Temperature

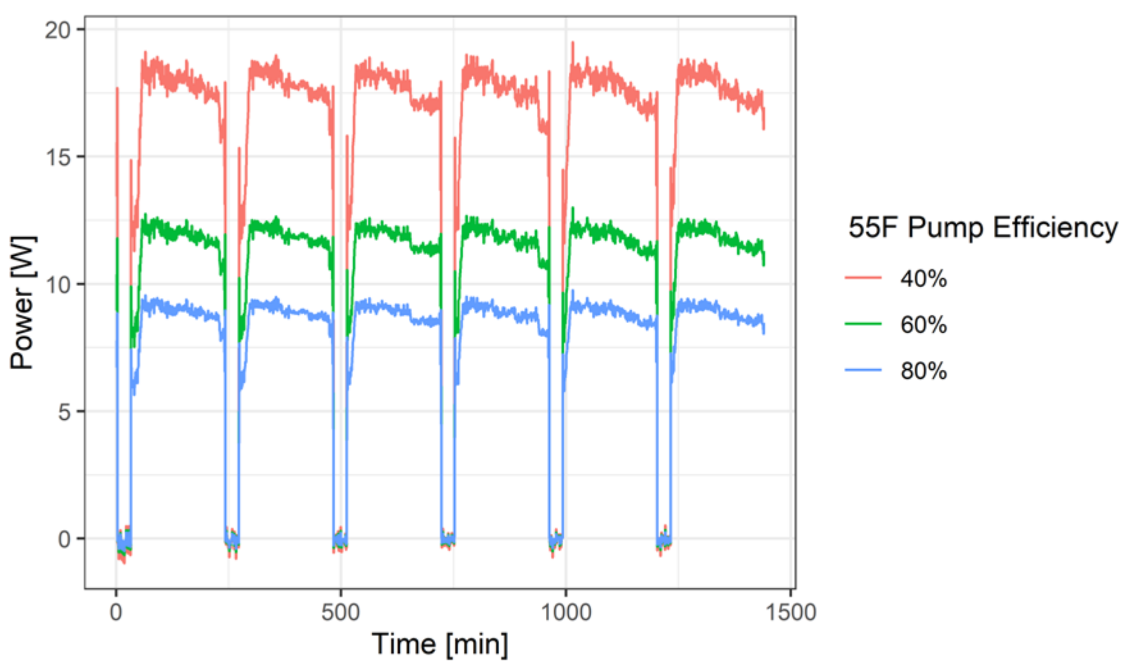


Figure 28. Transient Estimated Pump Energy Across Range of Efficiencies at 55 °F Condenser Inlet Water Temperature

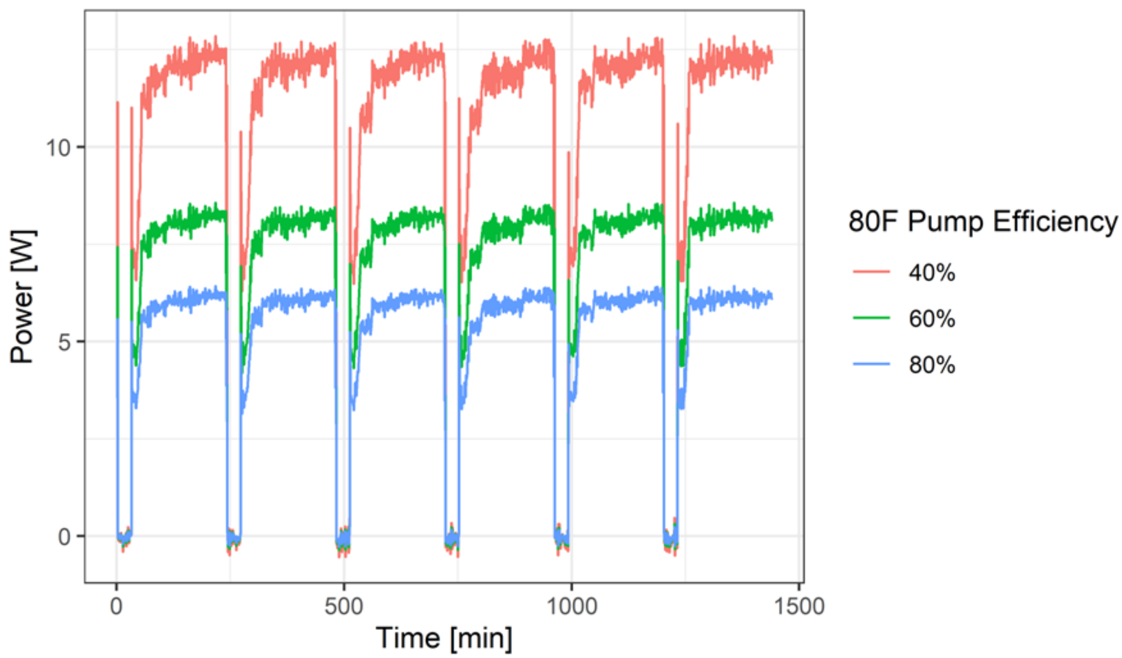


Figure 29. Transient Estimated Pump Energy Across Range of Efficiencies at 80 °F Condenser Inlet Water Temperature

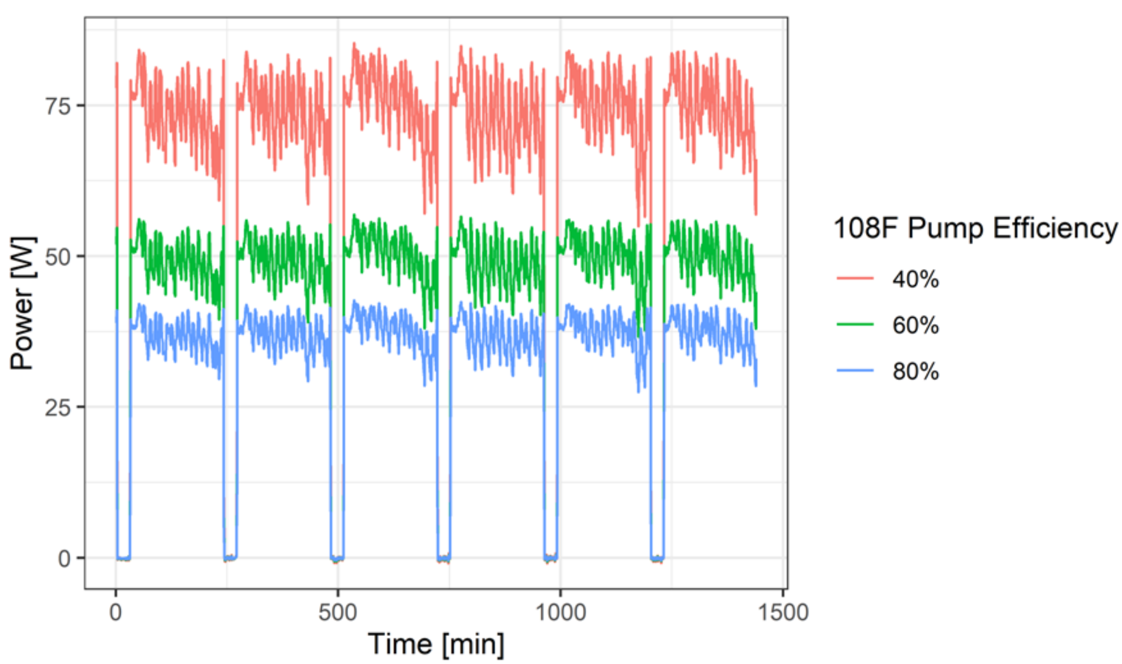


Figure 30. Transient Estimated Pump Energy Across Range of Efficiencies at 108 °F Condenser Inlet Water Temperature

The estimated pump energy and mean power at each pump efficiency and condenser inlet temperature is provided below in Table 6. The estimated total case energy and mean power consumption is also shown when the estimated pump energy and power is added, respectively. Additionally, the estimated energy savings and mean power reduction at each of these conditions is shown based on the estimated pump power.

Table 6. Case Total Energy Over 24 h and Mean Power Consumption at Each Pump Efficiency

Energy	40 % Efficiency (kWh/day)	60 % Efficiency (kWh/day)	80 % Efficiency (kWh/day)	Total Case Energy @ 40 % / 80 % Eff. (kWh/day)	Estimated % Energy Savings @ 40 % / 80 % Eff.
ECM at 108 °F Water Inlet Temperature	1.57	1.05	0.79	49.6 / 48.8	-5.65 / -3.99
ECM at 80 °F Water Inlet Temperature	0.24	0.16	0.12	30.9 / 30.7	34.3 / 34.5
ECM at 55 °F Water Inlet Temperature	0.37	0.24	0.18	22.0 / 21.8	53.2 / 53.6
Mean Power	**40 % Efficiency (W)**	**60 % Efficiency (W)**	**80 % Efficiency (W)**	**Total Case Mean Power @ 40 % / 80 % Eff. (W)**	**Estimated % Power Reduction @ 40 % / 80 % Eff.**
ECM at 108 °F Water Inlet Temperature	74.6	49.7	37.3	2363.3 / 2326.0	4.00 / 5.51
ECM at 80 °F Water Inlet Temperature	17.4	11.6	8.72	1469.2 / 1463.5	40.3 / 40.6
ECM at 55 °F Water Inlet Temperature	11.6	7.71	5.79	1038.8 / 1030.1	57.8 / 58.2

4 Conclusions

Retrofitting the baseline air-cooled, constant speed system with the water-cooled, variable-speed ECM generated daily energy savings of 34.5 % and 53.6 % under 80 °F and 55 °F condenser inlet water conditions. Water temperatures at or below 80 °F are realistic for most installations in the ComEd territory because they are representative of a typical summer ground water or cooling tower temperature. At all temperatures, the ECM provided more consistent cooling, and did not cycle off except for defrost. This resulted in more uniform product temperature. One post-retrofit observation was increased runtime at lower demand and more defrost cycles than the baseline. Pump energy was estimated based on the hydraulic power across the ECM condenser. Even assuming a low-efficiency (40 %) pump, the contribution to total energy was negligible. However, building-specific supply needs, such as a cooling loop using a chiller or a cooling tower could negatively impact savings.

In summary, the findings of this project clearly indicate that leveraging a liquid-cooled condenser coupled with variable-speed compression technology can yield promising energy savings. Based on the favorable findings from this project, it is highly recommended to consider this technology for field evaluations. Further studies could focus on the cumulative and interactive energy savings by integrating several cases into a central liquid-loop system, which could enhance whole-building energy savings.

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Appendix A. Case Product and Air Temperatures

The product simulator and internal air temperatures measured throughout each 24-h evaluation are provided in this Appendix. Eighteen product simulators were situated throughout the case at the corners of the top and middle shelf and bottom deck as shown in Figure 7. Product simulator temperatures are shown with their average, and the prescribed temperature limits according to AHRI and FDA regulations [2]. Only the average, and not individual simulator temperatures, are required to be maintained within this range. The average temperature of each simulator is also provided in the following sections.

Case air temperatures were measured at the locations shown in Figure 12. Evaporator inlet and outlet air temperatures were averaged from thermocouple probe measurements located at the left, center, and right evaporator fans. Air curtain discharge and return temperature measurements were averaged from three probes each located in-line with the evaporator fans at the left, center, and right side of the case. The interior ambient temperature was measured at the geometric center of the case interior. For the following figures, the nomenclature is:

- "T_Case_Cent" is the case centroid temperature
- "T_Evap_AirIn", "T_Evap_AirOut" are the evaporator inlet and outlet temperatures,
- "T_Curtain_Ret" and "T_Curtain_Dis" are the curtain return and discharge temperatures,
- and "T_PS" is the product simulator mean temperature.

A.1 Baseline Air and Product Temperature Measurements

During the baseline evaluation, the average of all product simulator temperatures was maintained within AHRI/FDA limits throughout operation as shown in Figure 31. Mean product simulator temperatures across the operation period are provided in Table 7, color-coded based on temperature differential beyond the AHRI/FDA limits. The bottom left rear (BLR) product simulator exhibited both the coldest average temperature (0.86 °C/33.5 °F) and the coldest peak temperature (0.08 °C/32.1 °F). The middle right front (MRF) product simulator exhibited the warmest average temperature (5.80 °C/42.4 °F) and the bottom right front (BRF) product simulator exhibited the warmest peak temperature (6.59 °C/43.9 °F). The average product simulator temperature was 3.50 °C (38.3 °F). Product simulator temperatures varied with compressor cycling and defrost cycles. During defrost, the average simulator temperature increased to 39.9 °F and during cycling, decreased to 37.5 °F. The bottom left rear simulator was the coldest due to the direction of airflow through perforations at the back of the case since bottom rear simulators are closest to the evaporator discharge. The bottom right front and middle right front simulators were the warmest since these simulators are closest to the air curtain return grille, and therefore exposed to the warmest air.

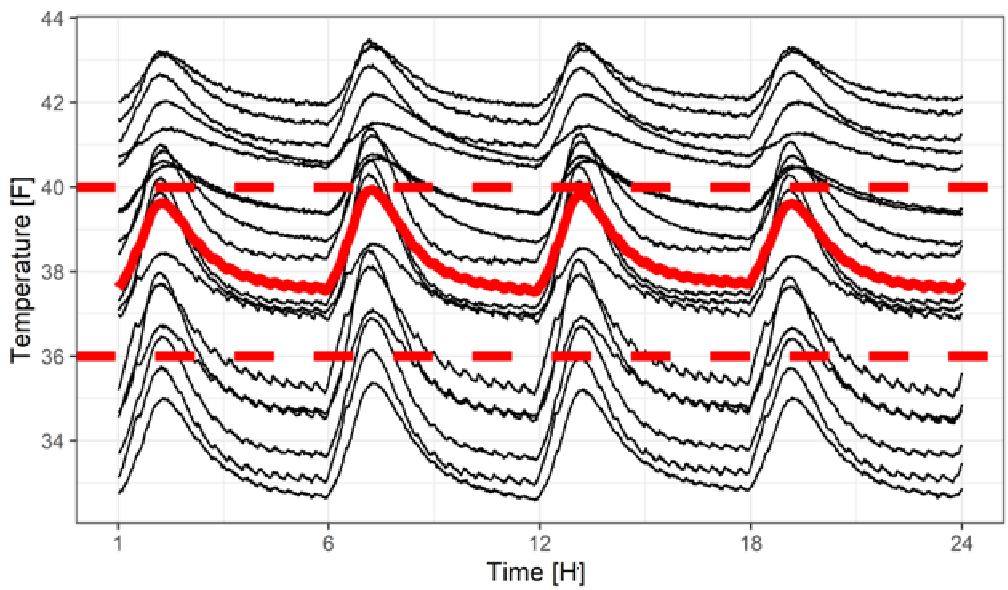

Figure 31. Transient Product Simulator Temperatures During Baseline Evaluation

The mean product simulator temperature is shown in red.

Table 7. Average Product Simulator Temperatures as a Function of Position in the Baseline Evaluation

Color-coding is based on temperature differential beyond the AHRI/FDA limits (required only for the average).

		x-position		
Shelf	**y-position**	**Left**	**Center**	**Right**
Top	Rear	1.45 °C	2.05 °C	3.41 °C
	Front	3.07 °C	4.37 °C	5.35 °C
Middle	Rear	1.11 °C	1.89 °C	3.96 °C
	Front	4.38 °C	4.94 °C	5.80 °C
Bottom	Rear	0.86 °C	2.31 °C	3.30 °C
	Front	5.15 °C	4.16 °C	5.65 °C

The case air temperatures during the baseline evaluation are shown in Figure 32. The average simulator temperature is shown for comparison. Evaporator outlet temperatures dropped to a minimum of -0.47 °C (31.2 °F) during compressor on-cycling. Curtain discharge and return air temperatures ranged between 0.15 °C (32.3 °F) and 11.7 °C (53.0 °F) and 9.14 °C (48.5 °F) and 16.3 °C (61.3 °F), respectively.

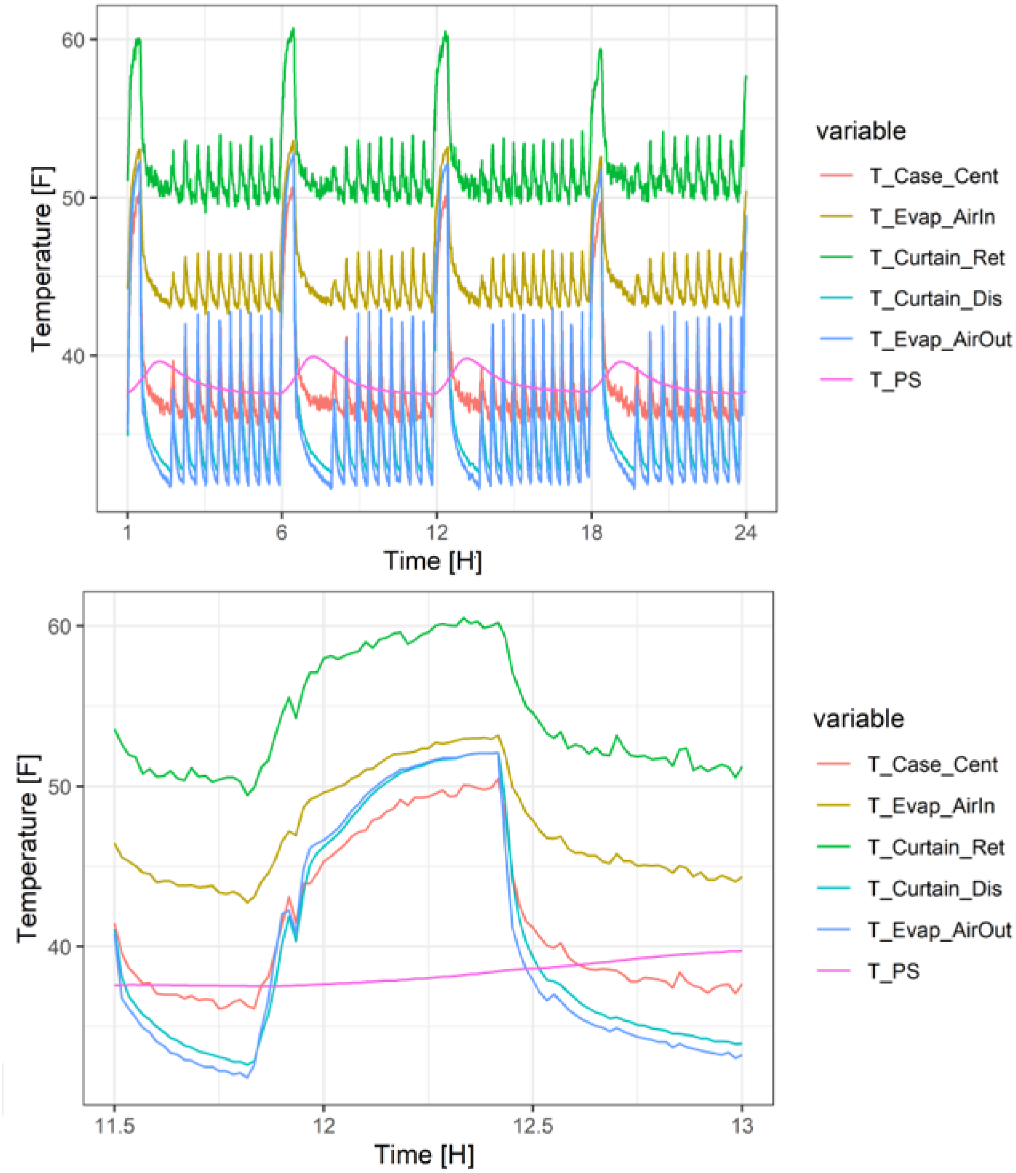

Figure 32. Case Air Temperature Measurements During Baseline Evaluation

"T_Case_Cent" is the case centroid temperature, "T_Evap_AirIn" and "T_Evap_AirOut" are the evaporator inlet and outlet temperatures, "T_Curtain_Ret" and "T_Curtain_Dis" are the curtain return and discharge temperatures, and "T_PS" is the product simulator mean temperature.

A.2 ECM Air and Product Temperature Measurements – 55 °F Liquid Inlet Temp

During the ECM evaluation at the lowest condenser liquid inlet temperature, the average of all product simulator temperatures was maintained within AHRI/FDA limits throughout operation as shown in Figure 33. Mean product simulator temperatures across the operation period are provided in Table 8, color-coded based on temperature differential beyond the AHRI/FDA limits. The bottom right rear (BRR) product simulator exhibited both the coldest average temperature (0.76 °C/33.4 °F) and coldest peak temperature (-0.15 °C/31.7 °F). The middle center front (MCF) product simulator exhibited both the warmest average temperature (6.88 °C/44.4 °F) and warmest peak temperature (7.73 °C/45.9 °F). The average product simulator temperature was 3.03 °C (37.4 °F). Product simulator temperatures varied with defrost cycles. During defrost, the average simulator temperature increased to 38.7 °F and during compressor cycling, decreased to 36.5 °F. The bottom right rear simulator was the coldest due to the direction of airflow through perforations at the back of the case since bottom rear simulators are closest to the evaporator discharge. The middle center front simulator exhibited the warmest temperature. This simulator is not as close to the air curtain return grille as the bottom shelf front simulators; however some front simulators exhibit higher temperatures at different locations across individual 24-h evaluations.

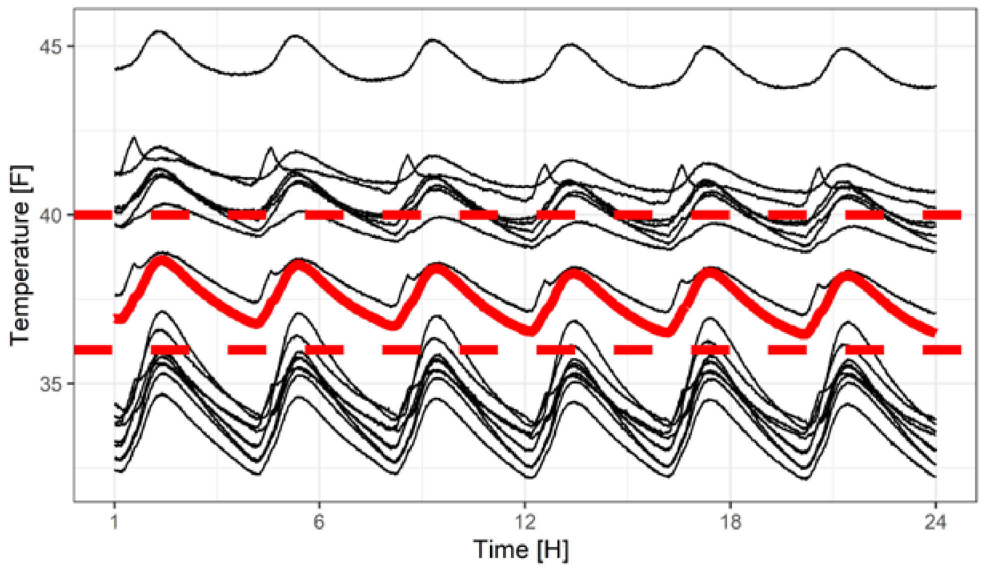

Figure 33. Transient Product Simulator Temperatures During ECM Evaluation at 55 °F Liquid Inlet Temperature.

The mean product simulator temperature is shown in red.

Table 8. Average Product Simulator Temperatures as a Function of Position in the ECM Evaluation at 55 °F Liquid Inlet Temperature.

Color-coding is based on temperature differential beyond the AHRI/FDA limits (required only for the average).

		x-position		
Shelf	y-position	Left	Center	Right
Top	Rear	1.82 °C	1.68 °C	1.23 °C
	Front	4.48 °C	4.52 °C	3.29 °C
Middle	Rear	1.18 °C	1.38 °C	1.28 °C
	Front	5.11 °C	4.97 °C	4.18 °C
Bottom	Rear	1.04 °C	1.41 °C	0.76 °C
	Front	6.88 °C	4.64 °C	4.65 °C

The case air temperatures during the ECM evaluation at the 55 °F liquid inlet temperature are shown in Figure 34. The average simulator temperature is shown for comparison. Evaporator outlet temperatures dropped to a minimum of -1.21 °C (29.8 °F) during compressor on-cycling. The average of curtain discharge and return air temperatures ranged between -0.41 °C (31.3 °F) and 11.0 °C (51.8 °F) and 8.17 °C (46.7 °F) and 15.9 °C (60.6 °F), respectively.

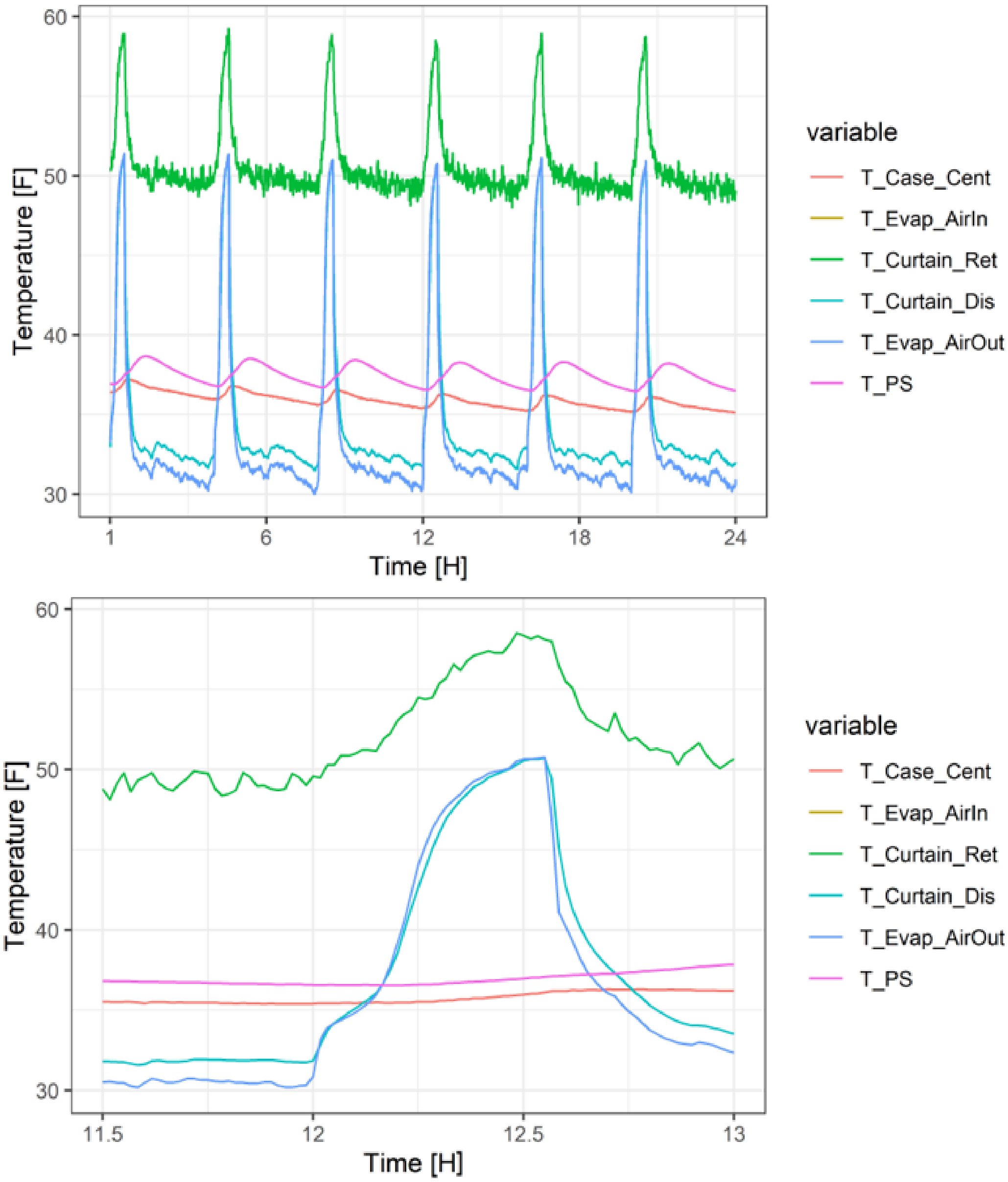

Figure 34. Case Air Temperature Measurements (Averaged Across Left, Right, and Center Probes) During ECM Evaluation at 55 °F Liquid Inlet Temperature.

A.3 ECM Air and Product Temperature Measurements – 80 °F Liquid Inlet Temp

During the ECM evaluation at the midpoint condenser liquid inlet temperature, the average of all product simulator temperatures was maintained within AHRI/FDA limits throughout operation as shown in Figure 35. Mean product simulator temperatures across the operation period are provided in Table 9, color-coded based on temperature differential beyond the AHRI/FDA limits. The bottom left rear (BLR) product simulator exhibited the coldest average temperature (1.42 °C/34.6 °F) and the middle left rear

(MLR) product simulator exhibited the coldest peak temperature (0.27 °C/32.5 °F). The bottom left front (BLF) product simulator exhibited both the warmest average temperature (7.19 °C/44.9 °F) and the warmest peak temperature (7.95 °C/46.3 °F). The average product simulator temperature was 3.38 °C (38.1°F). Product simulator temperatures varied with defrost cycles. During defrost, the average simulator temperature increased to 39.1 °F and during compressor cycling, decreased to 37.0 °F. The bottom left rear simulator was the coldest due to the direction of airflow through perforations at the back of the case since bottom rear simulators are closest to the evaporator discharge. The bottom left front simulator was the warmest since the bottom front simulators are closest to the air curtain return grille, and therefore exposed to the warmest air.

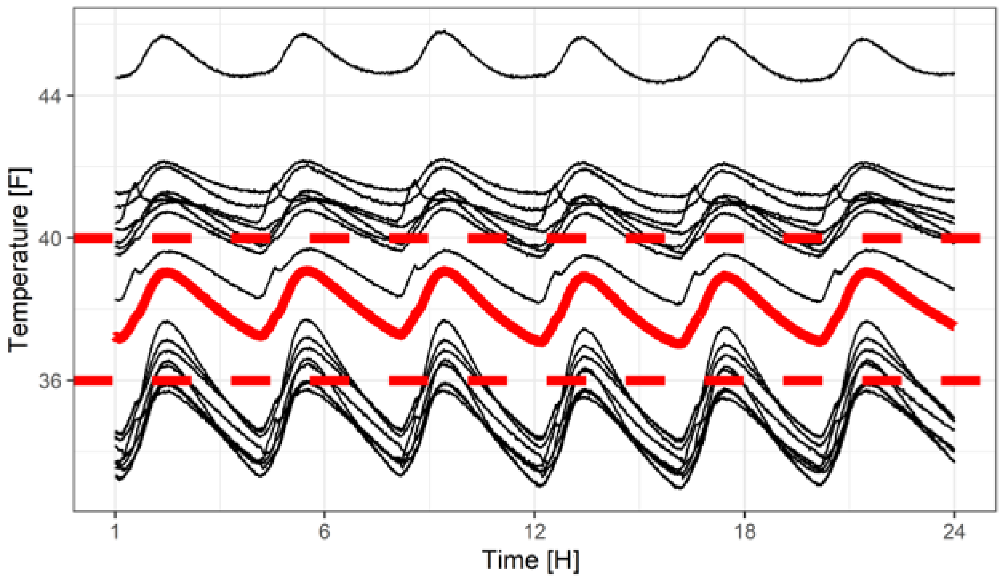

Figure 35. Transient Product Simulator Temperatures During ECM Evaluation at 80 °F Liquid Inlet Temperature.

The mean product simulator temperature is shown in red.

Table 9. Average Product Simulator Temperatures as a Function of Position in the ECM Evaluation at 80 °F Liquid Inlet Temperature.

Color-coding is based on temperature differential beyond the AHRI/FDA limits (required only for the average).

		x-position		
Shelf	**y-position**	**Left**	**Center**	**Right**
Top	Rear	2.16 °C	1.76 °C	2.04 °C
	Front	4.73 °C	4.53 °C	3.90 °C
Middle	Rear	1.55 °C	1.48 °C	2.10 °C
	Front	5.35 °C	4.93 °C	4.80 °C
Bottom	Rear	1.42 °C	1.52 °C	1.52 °C
	Front	7.19 °C	4.69 °C	5.18 °C

The case air temperatures during the ECM evaluation at the 80 °F liquid inlet temperature are shown in Figure 36. The average simulator temperature is shown for comparison. Evaporator outlet temperatures dropped to a minimum of -1.01 °C/30.2 °F during compressor on-cycling. Curtain discharge and return air temperatures ranged between -0.20 °C/31.6 °F and 10.8 °C/51.5 °F, and 8.62 °C/47.5 °F and 16.0 °C/60.7 °F, respectively.

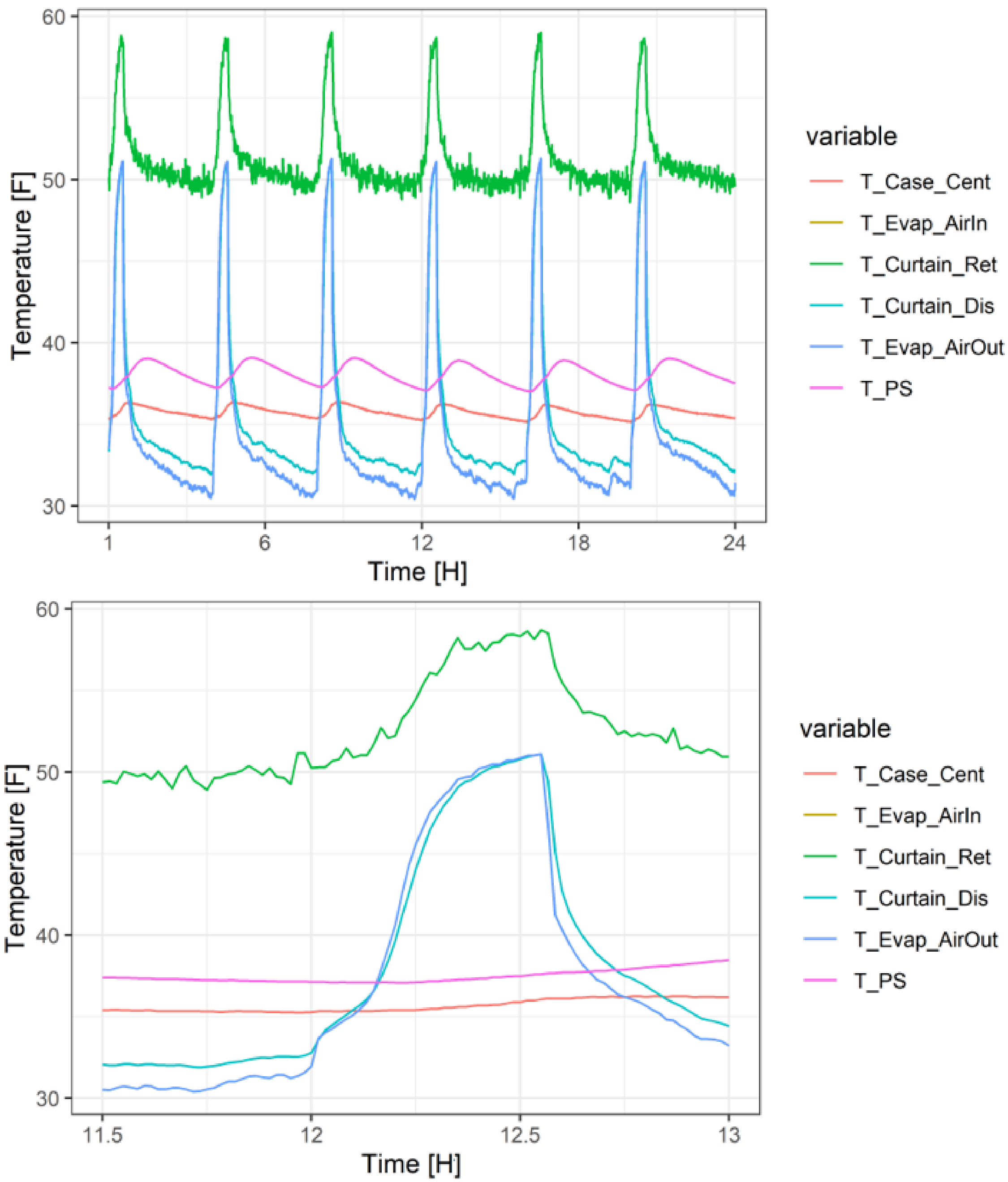

Figure 36. Case Air Temperature Measurements (Averaged Across Left, Right, and Center Probes) During ECM Evaluation at 80 °F Liquid Inlet Temperature.

A.4 ECM Air and Product Temperature Measurements – 108 °F Liquid Inlet Temp

During the ECM evaluation at the highest condenser liquid inlet temperature (closest to the baseline saturated condensing temperature), the average of all product simulator temperatures was maintained within AHRI/FDA limits throughout operation as shown in Figure 37. Mean product simulator temperatures across the operation period are provided in Table 9, color-coded based on temperature differential beyond the AHRI/FDA limits. The bottom left rear (BLR) product simulator exhibited both the coldest average temperature (1.18 °C/34.1 °F) and the coldest peak temperature (0.08 °C/32.1 °F). The bottom left front (BLF) product simulator exhibited both the warmest average temperature (7.50 °C/45.5 °F) and warmest peak temperature (8.29 °C/46.9 °F). The average product simulator temperature was 3.29 °C (37.9 °F). Product simulator temperatures varied with defrost cycles. During defrost, the average simulator temperature increased to 38.9 °F and during compressor cycling, decreased to 36.9 °F. The bottom left rear simulator was the coldest due to the direction of airflow through perforations at the back of the case since bottom rear simulators are closest to the evaporator discharge. The bottom left front simulator was the warmest since the bottom front simulators are closest to the air curtain return grille, and therefore exposed to the warmest air.

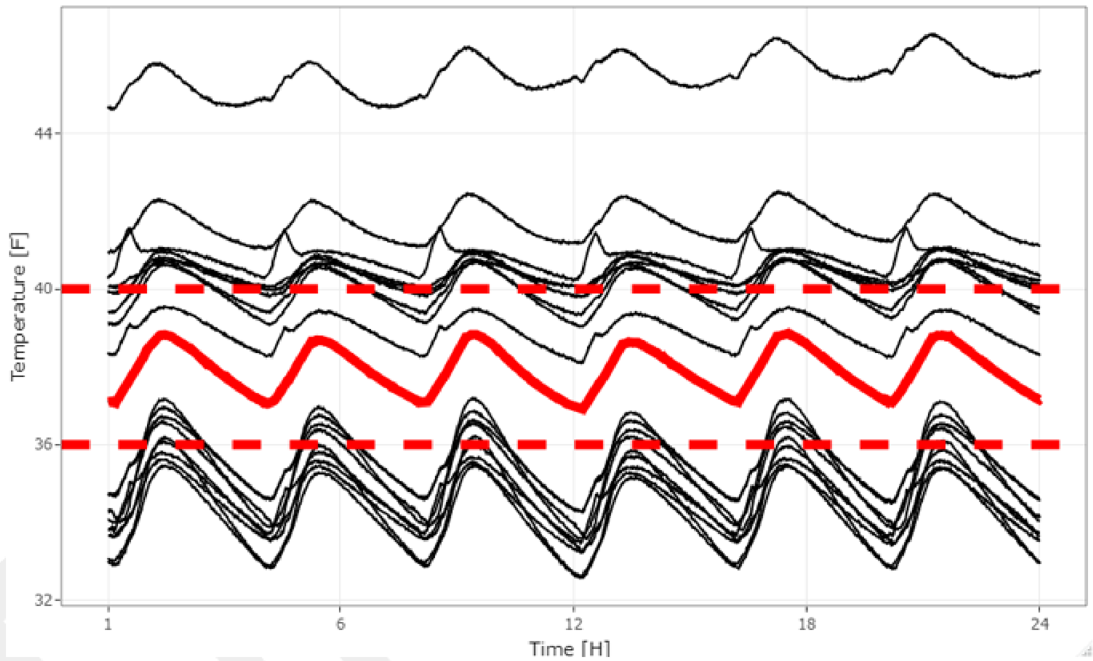

Figure 37. Transient Product Simulator Temperatures During ECM Evaluation at 108 °F Liquid Inlet Temperature.

The mean product simulator temperature is shown in red.

Table 10. Average Product Simulator Temperatures as a Function of Position in the ECM Evaluation at 108 °F Liquid Inlet Temperature.

Color-coding is based on temperature differential beyond the AHRI/FDA limits (required only for the average).

		x-position		
Shelf	y-position	Left	Center	Right
Top	Rear	1.91 °C	1.79 °C	1.99 °C
	Front	4.42 °C	4.48 °C	3.85 °C
Middle	Rear	1.36 °C	1.52 °C	2.03 °C
	Front	4.69 °C	4.89 °C	4.66 °C
Bottom	Rear	1.18 °C	1.53 °C	1.38 °C
	Front	7.50 °C	5.38 °C	4.66 °C

The case air temperatures during the ECM evaluation at the 108 °F liquid inlet temperature are shown in Figure 38. The average simulator temperature is shown for comparison. Evaporator outlet temperatures dropped to a minimum of -1.28 °C/29.7 °F during compressor on-cycling. Curtain discharge and return air temperatures ranged between 0.03 °C/32.1 °F and 7.77 °C/46.0 °F, and 9.23 °C/48.6 °F and 15.3 °C/59.5 °F, respectively.

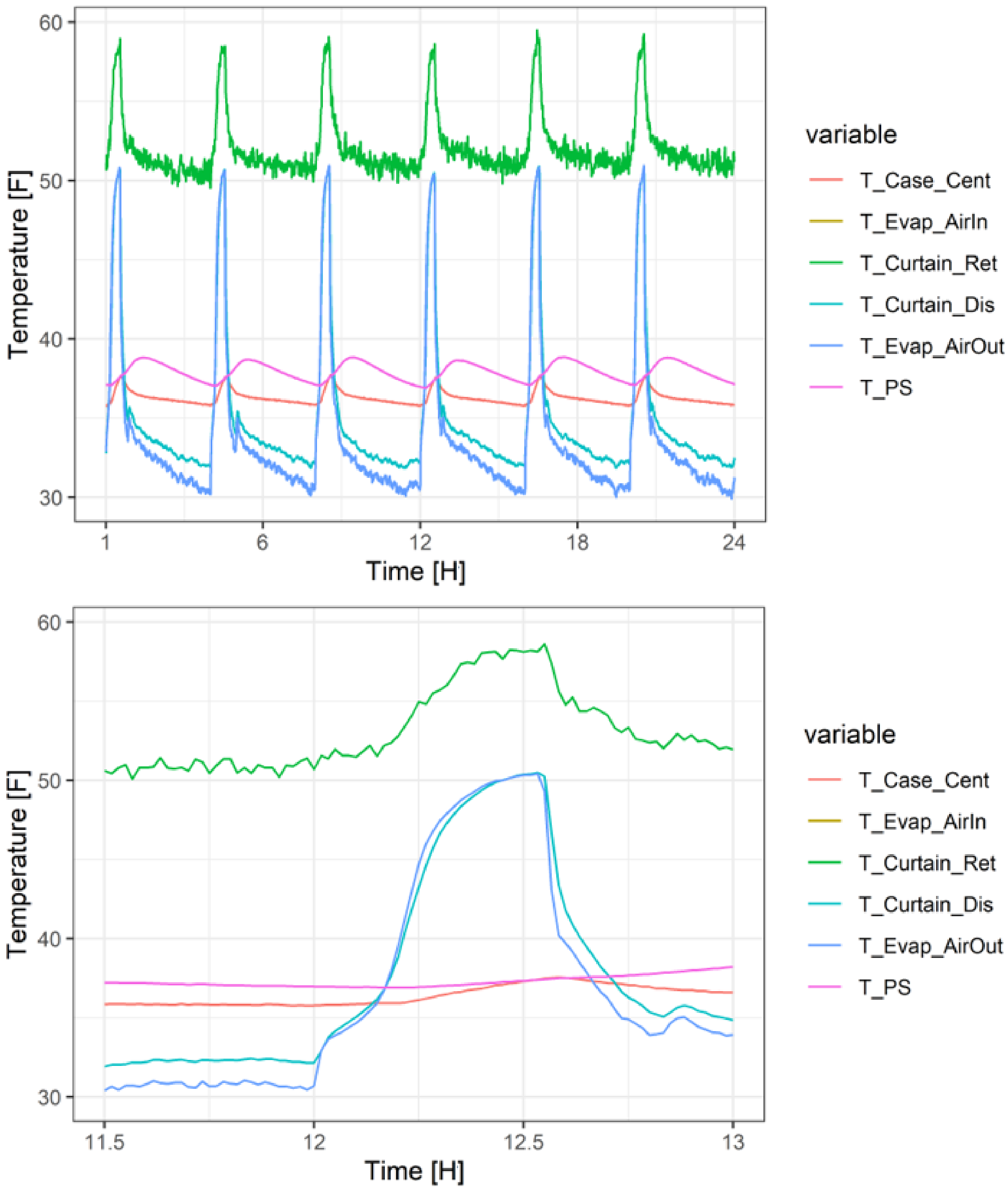

Figure 38. Case Air Temperature Measurements (Averaged Across Left, Right, and Center Probes) During ECM Evaluation at 108 °F Liquid Inlet Temperature.

Appendix B. Environmental Chamber Conditions

The temperature measurements recorded in the environmental chamber during each assessment are provided in this appendix section. Dry-bulb temperatures (DBTs) were recorded on the ambient test measurement pole at the locations listed in Figure 7 titled T_A and T_B. The dew-point temperature (DPT) and relative humidity (RH) were measured on the test pole at location T_A. The prescribed limits for maintaining each DBT and DPT based on the values listed in Table 4 are shown in the provided figures.

In each figure in the following section, the DBT and DPT are shown at T_A within their prescribed limits. The DBT at T_A fluctuated moderately in the baseline case, however did not drift beyond the prescribed limits more than a few seconds at a time. The variation in DBT at T_B was steady across 24-h evaluations, although dropped below the prescribed limits. However, this can be explained by the location of T_B. Since the ambient test pole was located one foot closer to the air curtain than ASHRAE methodology, cold air ejected from the air curtain was measured by the thermocouple probe, reducing its temperature. Conditions were relatively the same across individual tests, although baseline conditions exhibited more fluctuation due to the presence of the air-cooled condenser within the baffling system. For the following Appendix B figures, the nomenclature is:

- "DB" is dry bulb temperature
- "DP" is dew point temperature

B.1 Baseline Evaluation Environmental Temperatures

The chamber environmental DBTs at location T_A and T_B, and the DPT at location T_A during evaluation of the baseline refrigerator display case are shown in Figure 39, along with their prescribed limits. The temperatures at T_A were maintained within their prescribed limits throughout evaluation and only occasionally dropped below those limits during peak compressor on-cycling. Temperatures at T_B were mostly maintained within its prescribed limits, however during on-cycling, were unable to be maintained at its prescribed limit due to proximity to the air curtain.

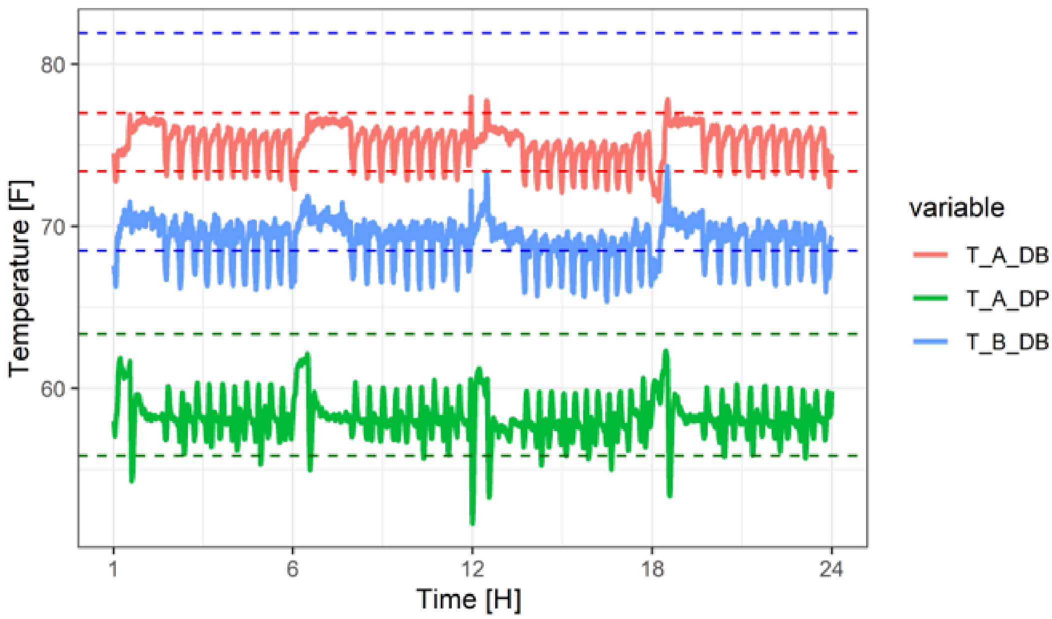

Figure 39. Chamber Dry-Bulb and Dew-Point Temperatures During Baseline Evaluation.

B.2 ECM Evaluation Environmental Temperatures

The chamber environmental DBTs at location T_A and T_B, and the DPT at location T_A during evaluation of the ECM are shown in the following figures. Environmental conditions at condenser liquid inlet temperatures of 55 °F are shown in Figure 40. Environmental conditions at condenser liquid inlet temperatures of 80 °F are shown in Figure 41. Environmental conditions at condenser liquid inlet temperatures of 108 °F are shown in Figure 42. The temperatures at T_A were maintained within their prescribed limits throughout evaluation and never approached those limits. Temperatures at T_B were maintained within its prescribed limit during most tests, however occasionally dropped below the lower limit due to proximity to the air curtain.

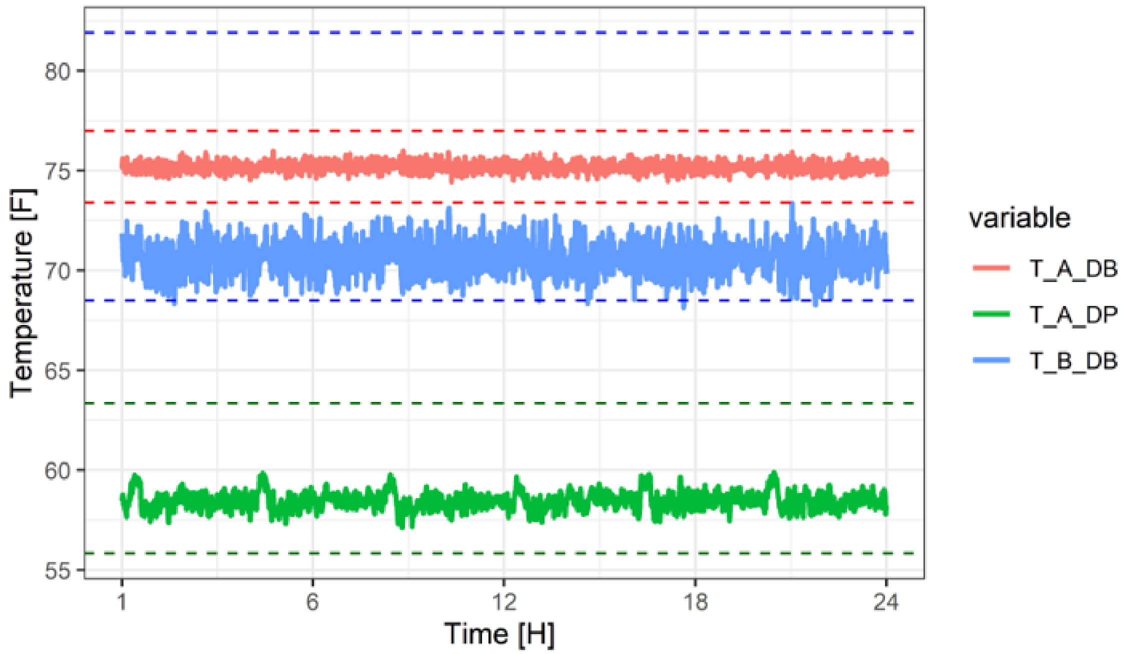

Figure 40. Chamber Dry-Bulb and Dew-Point Temperatures During ECM Evaluation at 55 °F Liquid Inlet Temperature.

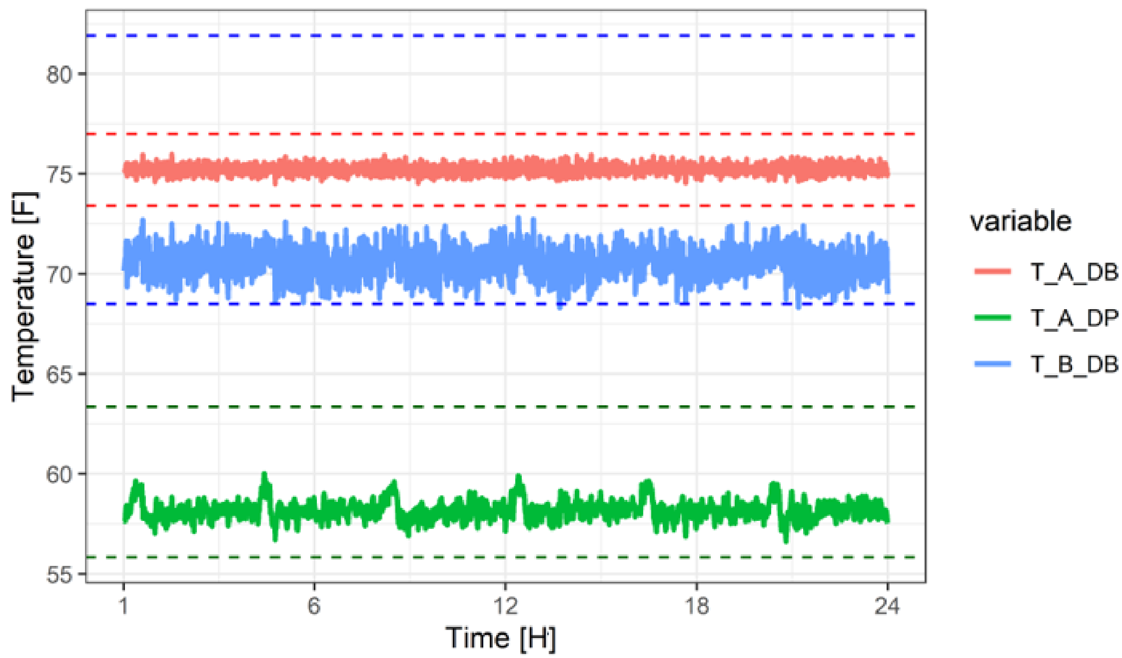

Figure 41. Chamber Dry-Bulb and Dew-Point Temperatures During ECM Evaluation at 80 °F Liquid Inlet Temperature.

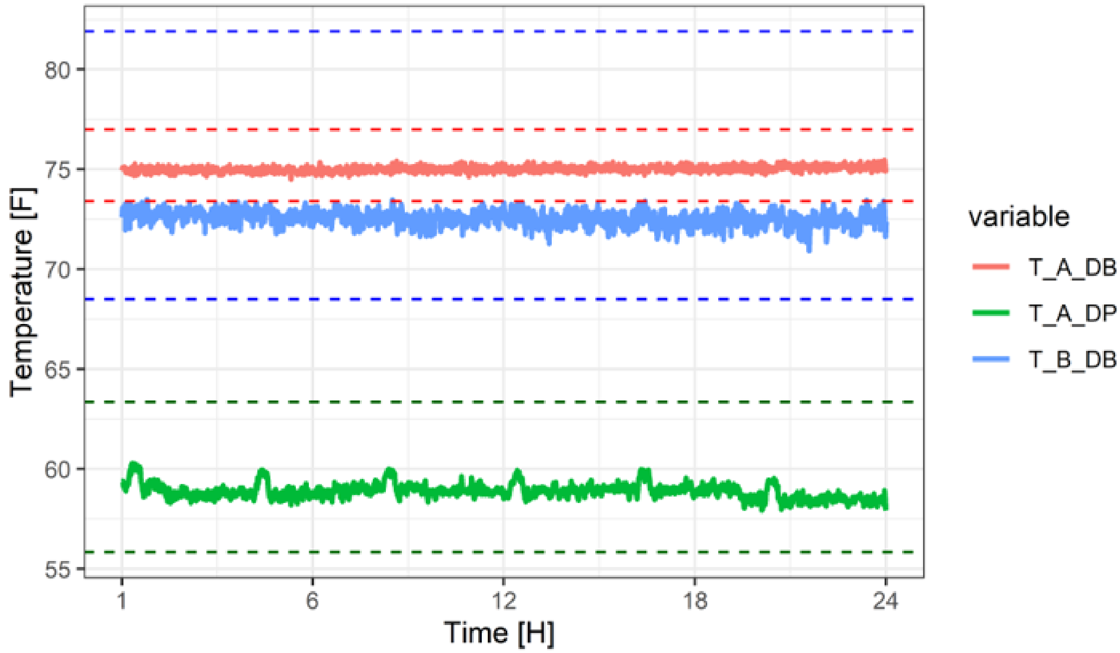

Figure 42. Chamber Dry-Bulb and Dew-Point Temperatures During ECM Evaluation at 108 °F Liquid Inlet Temperature.

Appendix C. Refrigerant Measurements

Surface temperature thermocouples were wrapped around the refrigerant piping and covered with insulation to record temperatures across each 24-h chamber evaluation as shown in Figure 13. The surface temperature of the refrigerant piping is not reflective of the actual refrigerant temperature due to the thermal resistance through the piping material. Refrigerant pressure transducers were tapped at service valves at the compressor suction (vapor) and condenser outlet (liquid). In order to evaluate the performance of commercially available refrigerated case technologies in a manner that most reflected customer use, it was critical to avoid conducting any measurements that could alter performance of the case. Therefore, thermocouple and pressure transducer taps were not made in the refrigerant lines. Here, surface temperature measurements were only used by NREL engineers to guide understanding of case performance. Therefore, the following refrigerant piping temperatures should not be considered performance indicators for these technologies under evaluated conditions. Refrigerant pressures were however measured directly and not subject to these limitations. For the following Appendix C figures, the nomenclature is:

- "T_Evap_RefOut" is the refrigerant surface temperature at the evaporator outlet
- "T_XV_RefIn" and "T_XV_RefOut" are the refrigerant surface temperatures at the inlet and outlet to the expansion valve, respectively
- "T_Cond_RefOut" is the refrigerant surface temperature at the outlet to the condenser
- "T_Comp_Dis" and "T_Comp_Suc" are the refrigerant surface temperatures at the discharge and suction side of the compressor, respectively
- "Pr_Cond" and "Pr_Evap" are the condenser and evaporator pressures measured at the service valves to the refrigerant receiver and accumulator, respectively
- "T_Cond_Sat" and "T_Evap_Sat" are the refrigerant saturation temperatures at the measured pressures for the condenser and evaporator
- "T_Cond_SC" and "T_Evap_SH" are the refrigerant subcooling and superheat temperature differentials, respectively, at the condenser and evaporator
- "T_Cond_FanIn" and "T_Cond_WatIn" is the air or water temperature, respectively, entering the condenser coil.

C.1 Baseline Refrigerant Measurements

The baseline refrigerant piping temperatures and pressures are shown below in the following figures. Refrigerant temperatures fluctuate accordingly with compressor cycles. Refrigerant piping temperatures are shown in Figure 43. Refrigerant pressures are shown in Figure 44, and calculated temperatures, including evaporator superheat and saturation temperature, condenser subcooling and saturation temperature are shown in Figure 45. Just as was done with case air temperatures and component power, the refrigerant piping measurements are shown across a zoomed-in 1.5 h period around the end of the scheduled door openings.

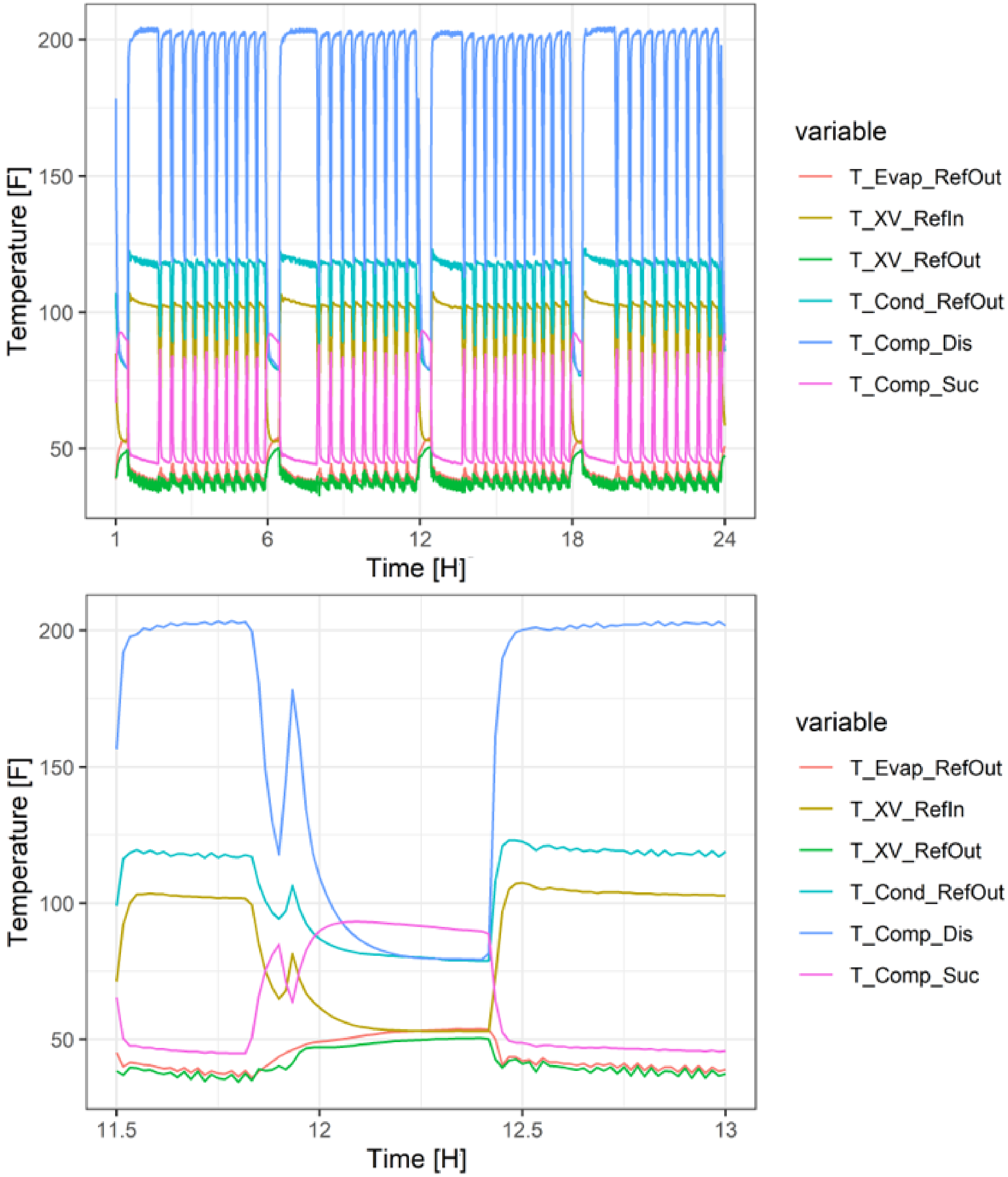

Figure 43. Refrigerant Piping Temperatures During Baseline Evaluation.

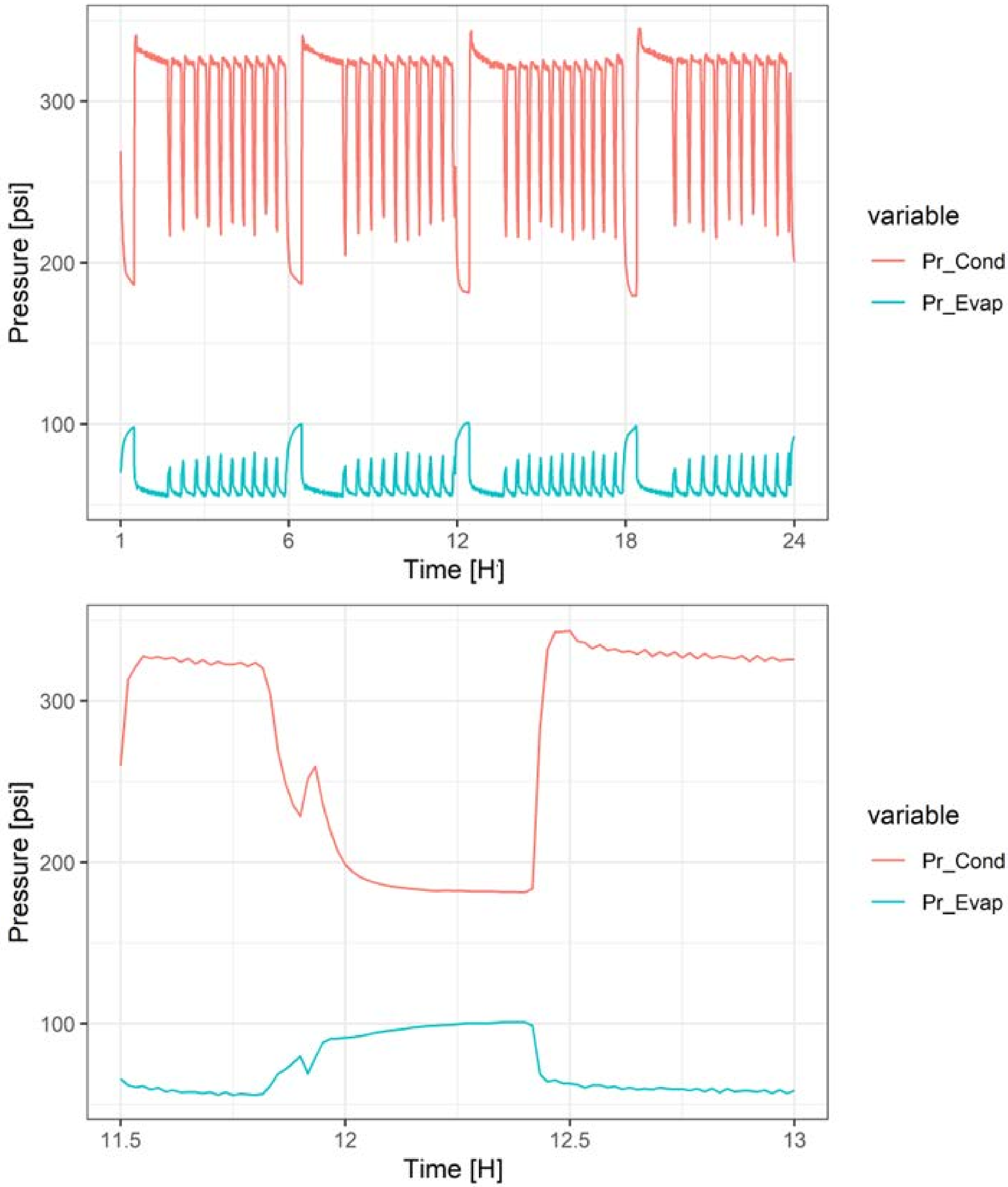

Figure 44. Refrigerant Piping Pressures During Baseline Evaluation.

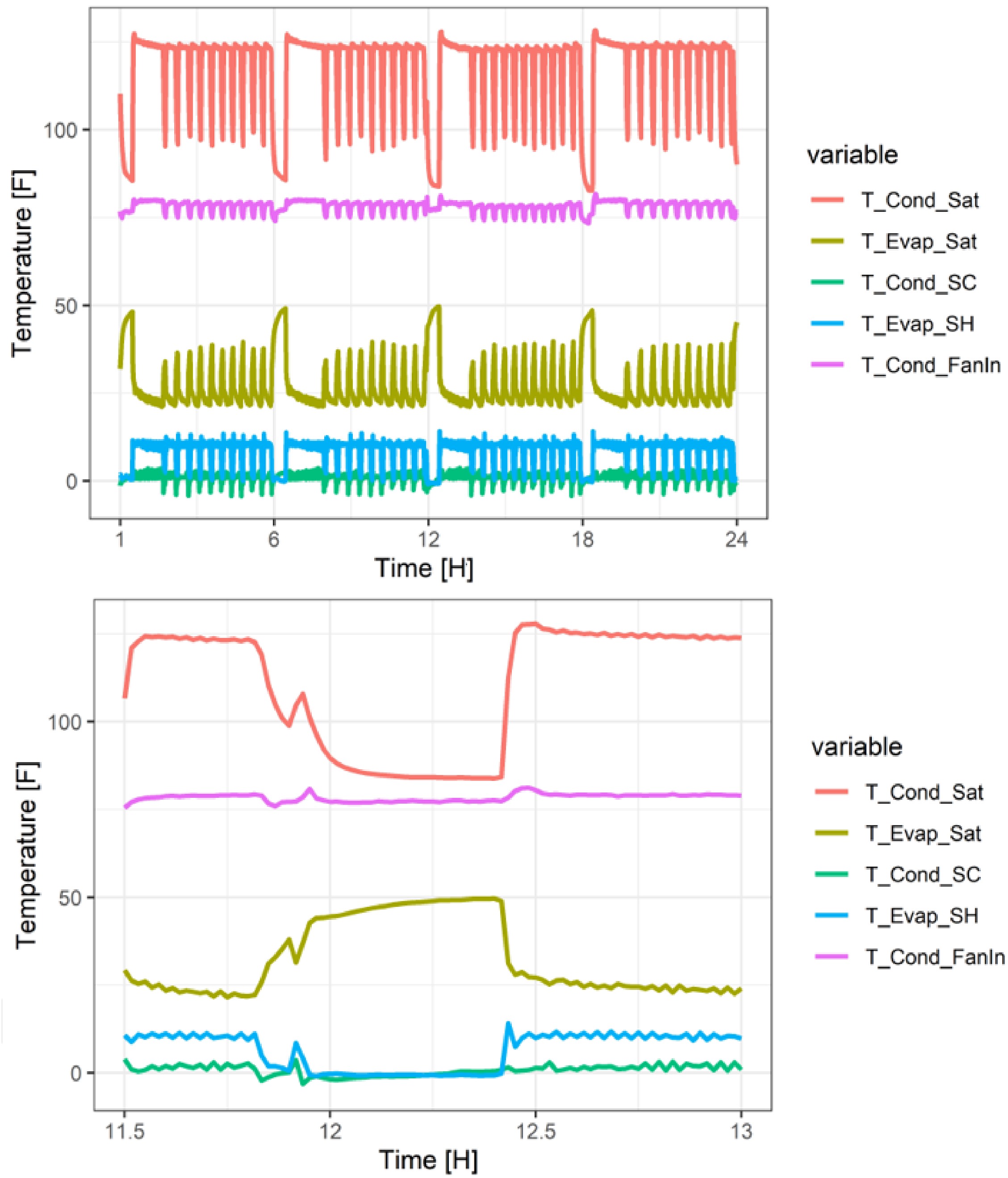

Figure 45. Calculated Refrigerant Piping Temperatures During Baseline Evaluation.

C.2 ECM Refrigeration Measurements – 55 °F Liquid Inlet Temp

The ECM refrigerant piping temperatures and pressures at a 55 °F liquid inlet temperature are shown below in the following figures. Refrigerant temperatures fluctuate accordingly with compressor cycles. Refrigerant piping temperatures are shown in Figure 46. Refrigerant pressures are shown in Figure 47, and calculated temperatures, including evaporator superheat and saturation temperature, condenser subcooling and saturation temperature are shown in Figure 48. Just as was done with case air temperatures and component power, the refrigerant piping measurements are shown across a zoomed-in 1.5-h period around the end of the scheduled door openings.

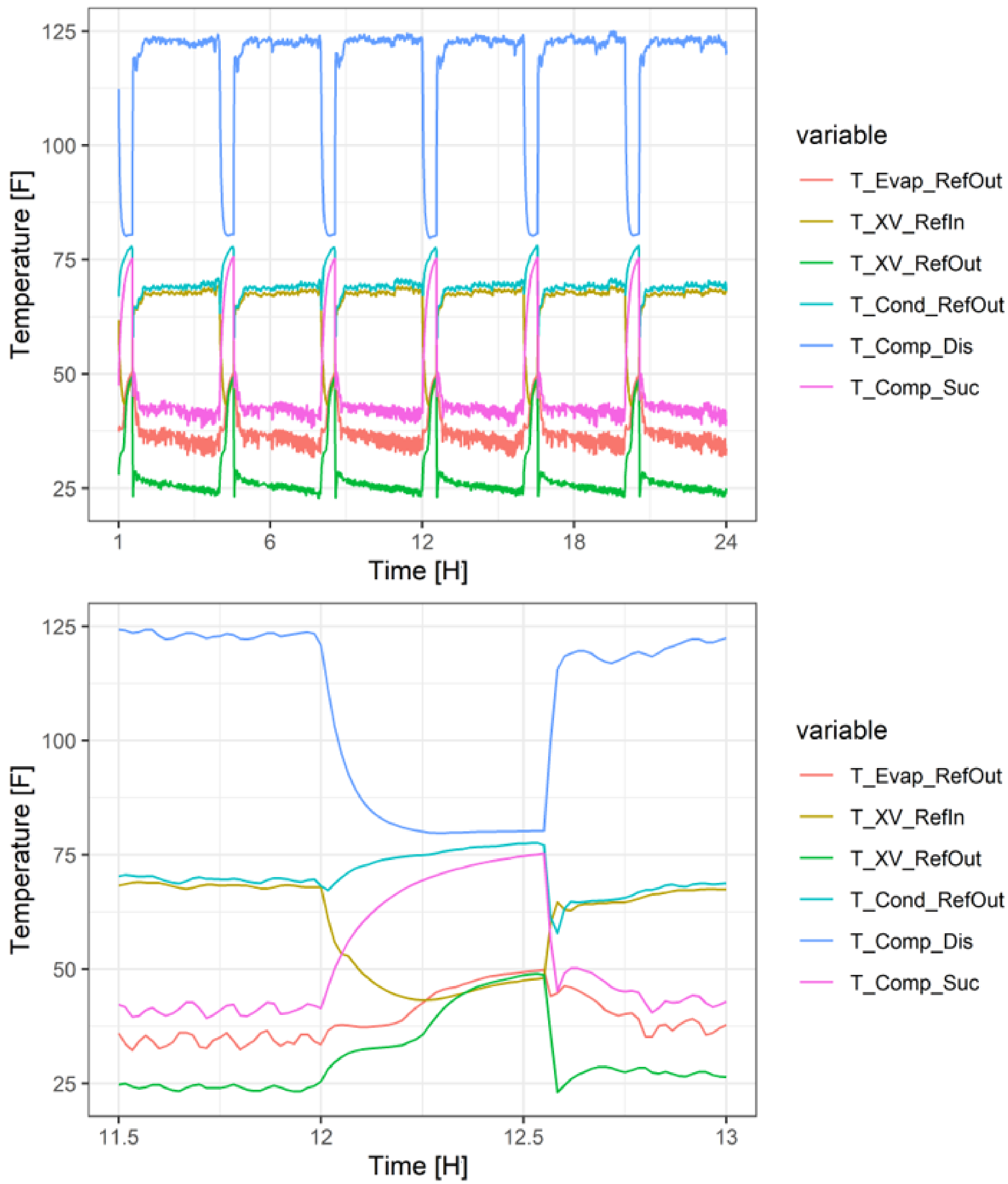

Figure 46. Refrigerant Piping Temperatures During ECM Evaluation at 55 °F Liquid Inlet Temperature.

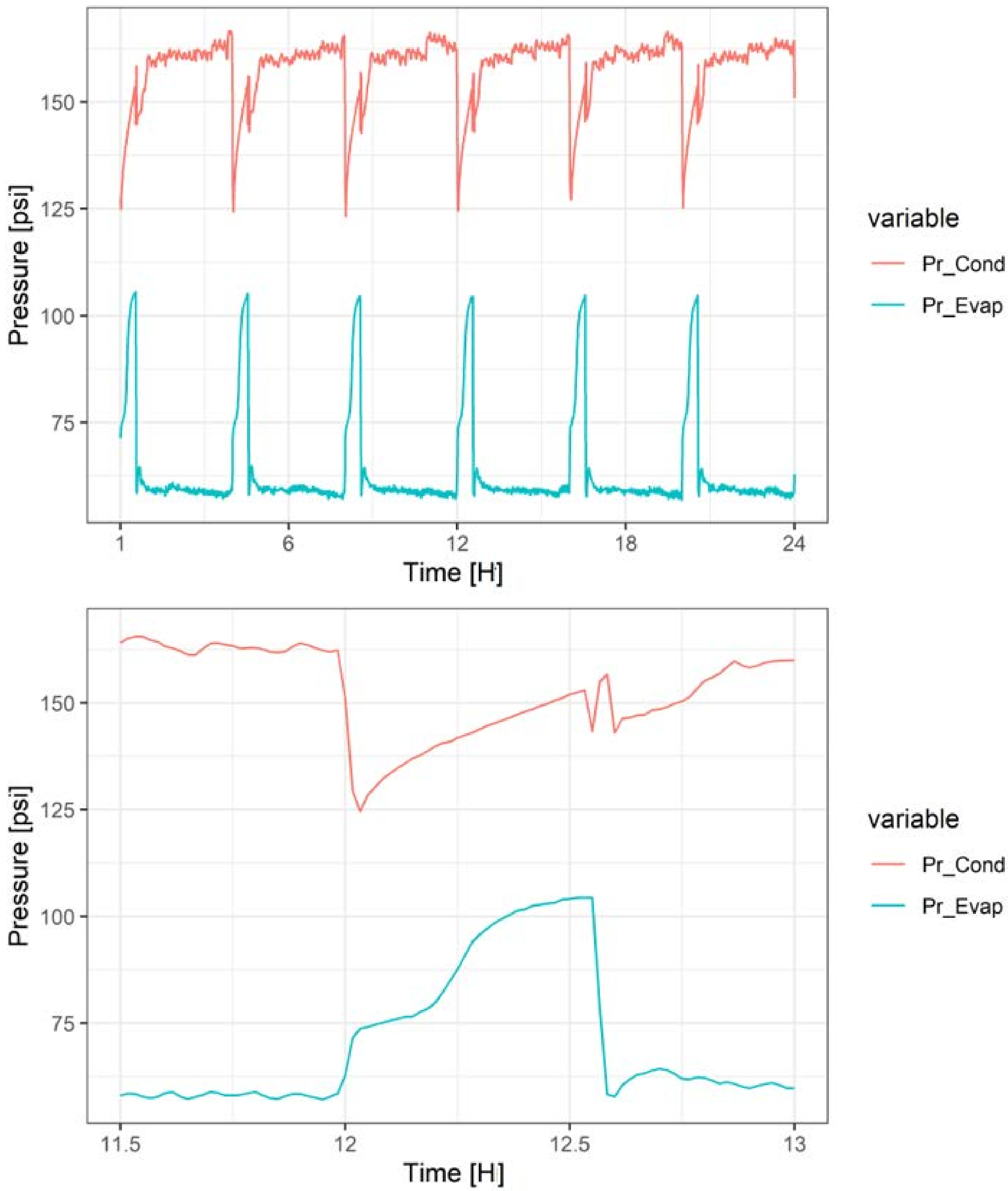

Figure 47. Refrigerant Piping Pressures During ECM Evaluation at 55 °F Liquid Inlet Temperature.

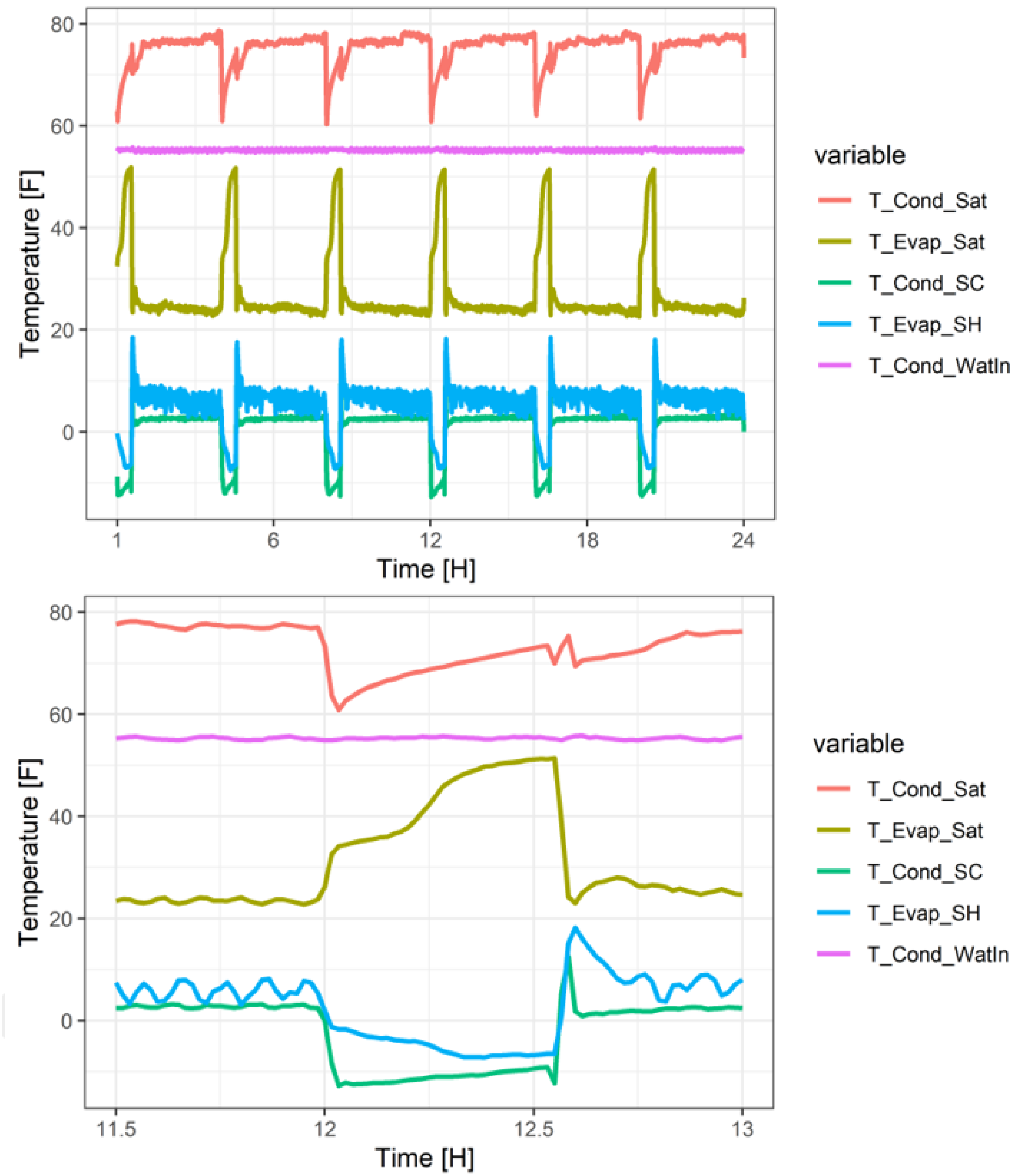

Figure 48. Calculated Refrigerant Piping Temperatures During ECM Evaluation at 55 °F Liquid Inlet Temperature.

C.3 ECM Refrigeration Measurements – 80 °F Liquid Inlet Temp

The ECM refrigerant piping temperatures and pressures at an 80 °F liquid inlet temperature are shown below in the following figures. Refrigerant temperatures fluctuate accordingly with compressor cycles. Refrigerant piping temperatures are shown in Figure 49. Refrigerant pressures are shown in Figure 50, and calculated temperatures, including evaporator superheat and saturation temperature, condenser subcooling and saturation temperature are shown in Figure 51. Just as was done with case air temperatures and component power, the refrigerant piping measurements are shown across a zoomed-in 1.5 h period around the end of the scheduled door openings.

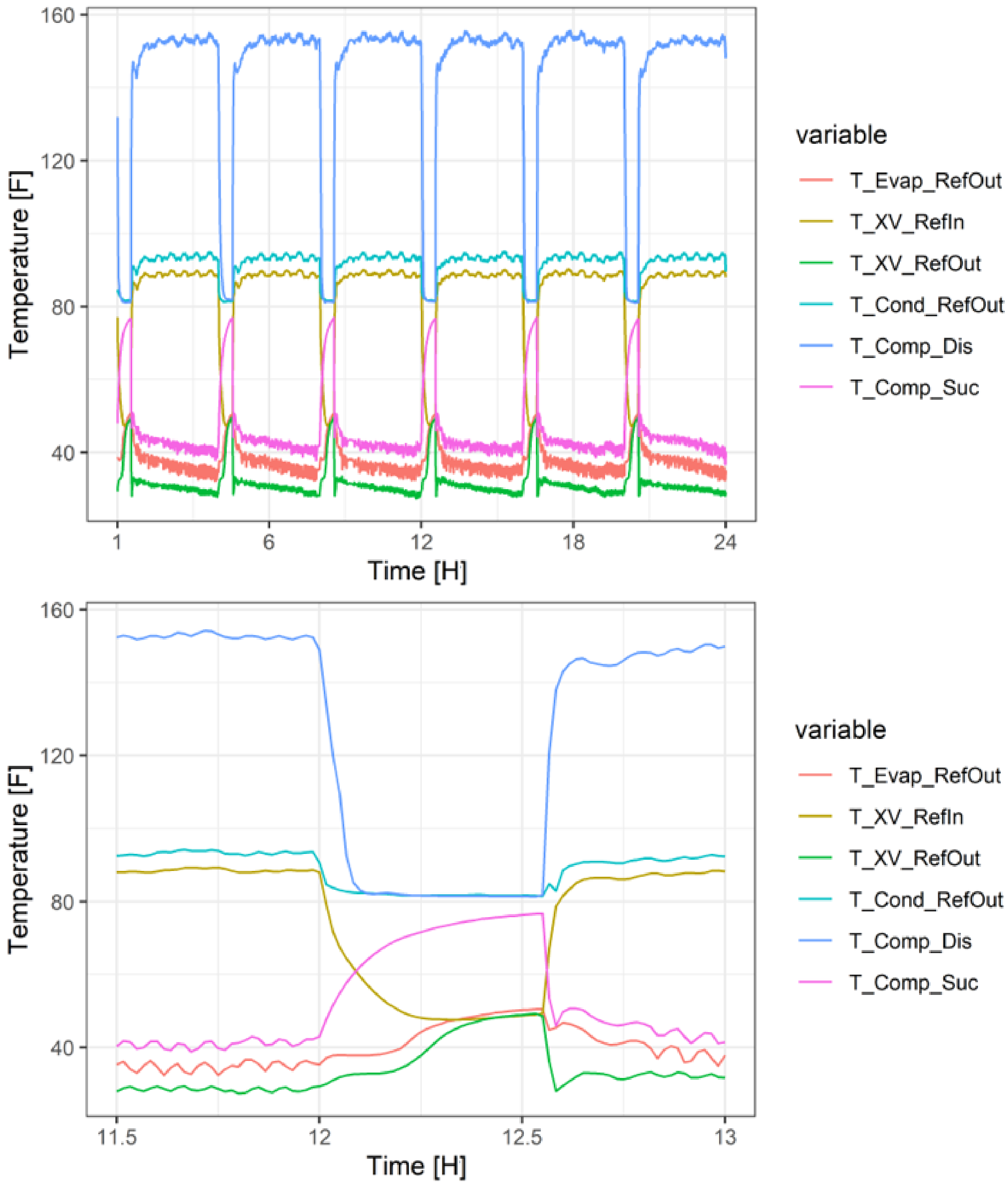

Figure 49. Refrigerant Piping Temperatures During ECM Evaluation at 80 °F Liquid Inlet Temperature.

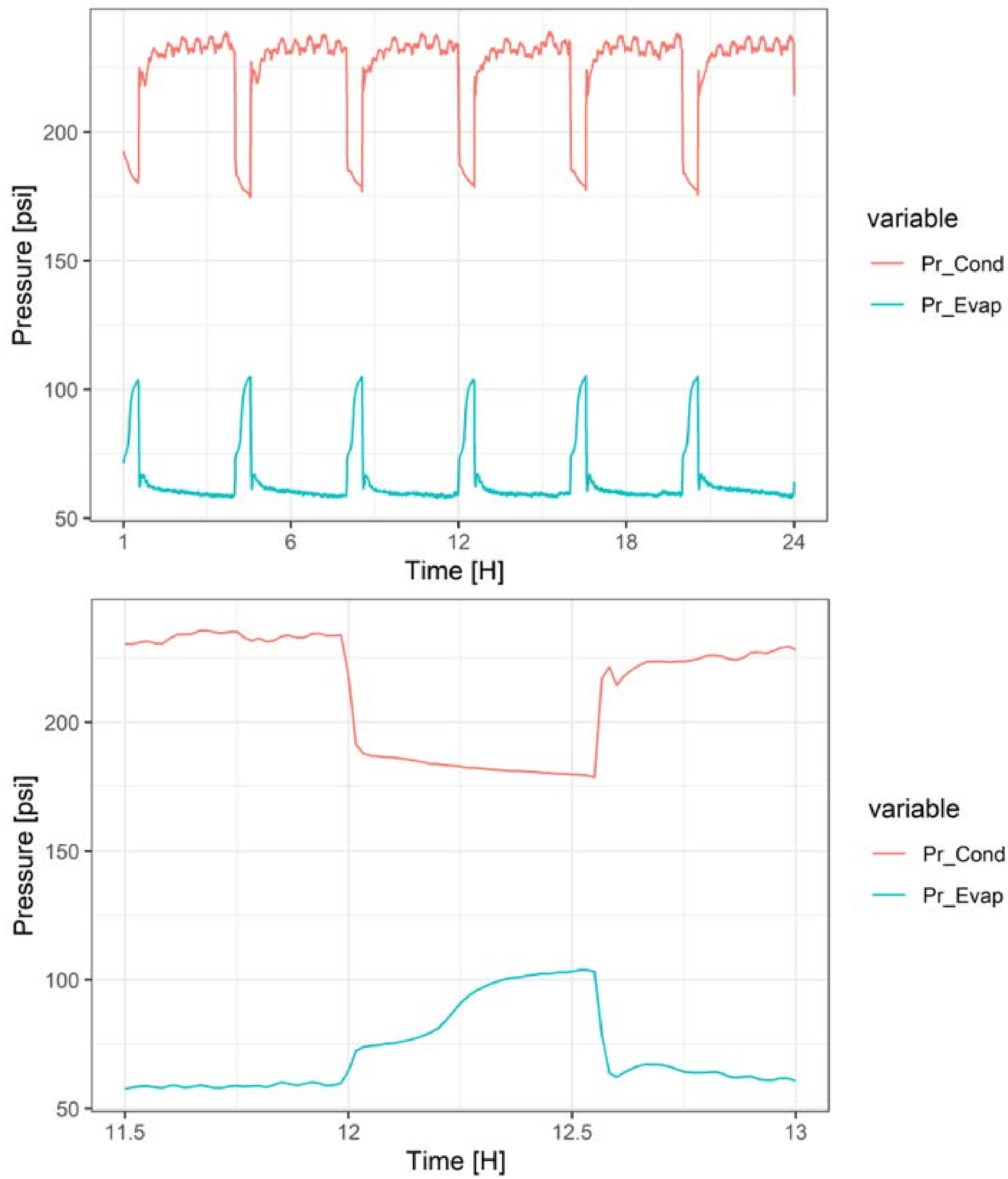

Figure 50. Refrigerant Piping Pressures During ECM Evaluation at 80 °F Liquid Inlet Temperature.

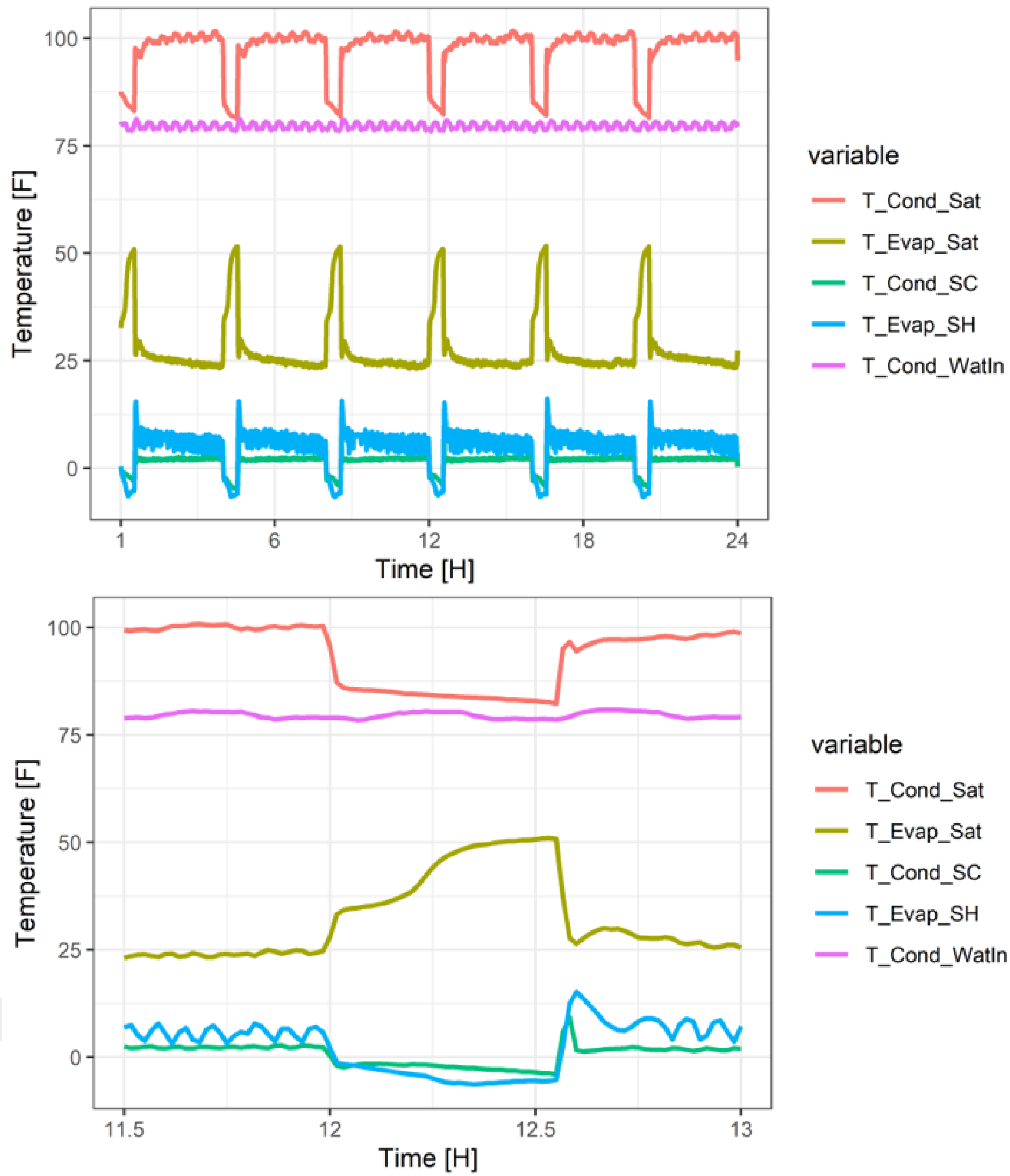

Figure 51. Calculated Refrigerant Piping Temperatures During ECM Evaluation at 80 °F Liquid Inlet Temperature.

C.4 ECM Refrigeration Measurements – 108 °F Liquid Inlet Temp

The ECM refrigerant piping temperatures and pressures at a 108 °F liquid inlet temperature are shown below in the following figures. Refrigerant temperatures fluctuate accordingly with compressor cycles. Refrigerant piping temperatures are shown in Figure 52. Refrigerant pressures are shown in Figure 53, and calculated temperatures, including evaporator superheat and saturation temperature, condenser subcooling and saturation temperature are shown in Figure 54. Just as was done with case air temperatures and component power, the refrigerant piping measurements are shown across a zoomed-in 1.5 h period around the end of the scheduled door openings.

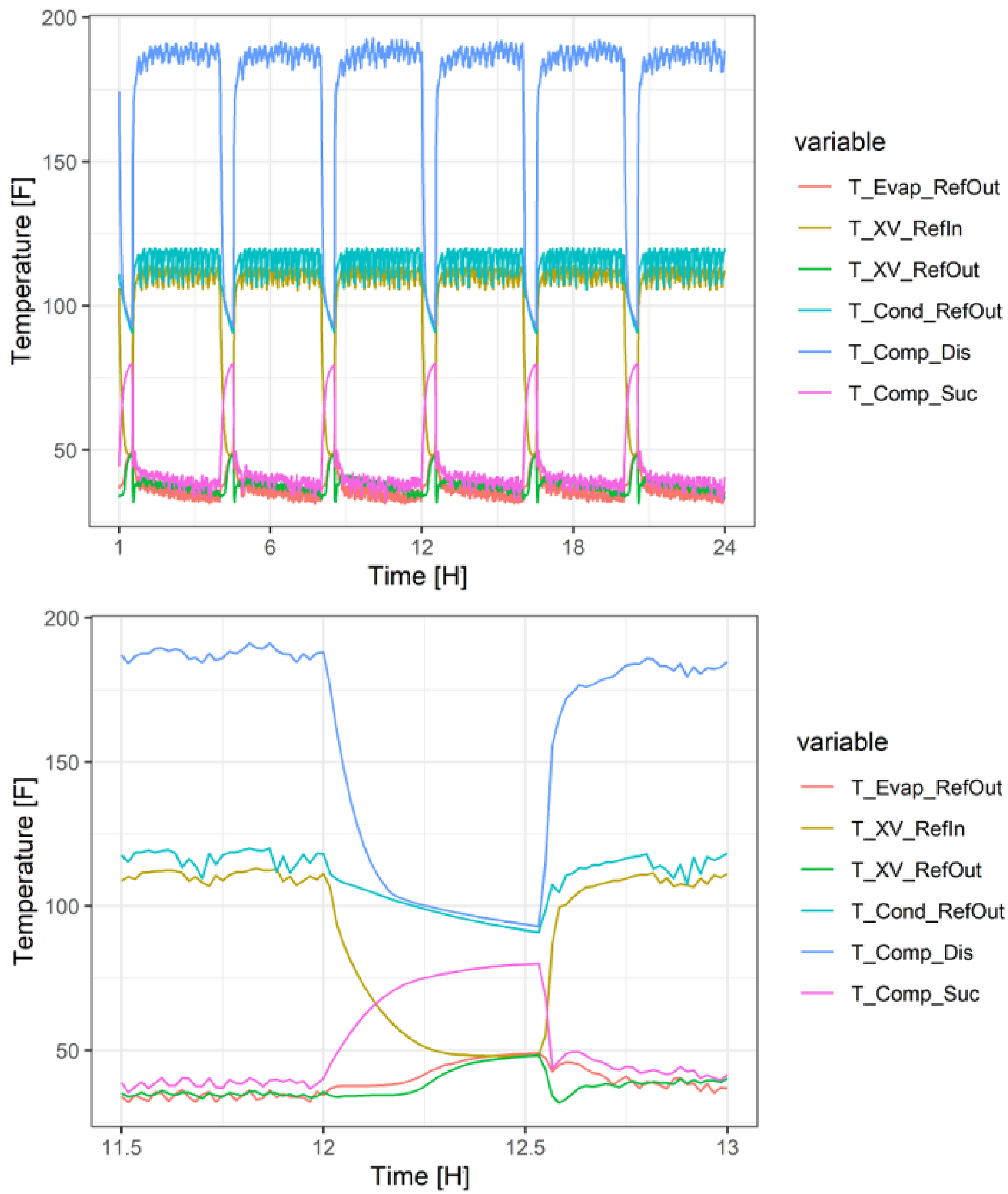

Figure 52. Refrigerant Piping Temperatures During ECM Evaluation at 108 °F Liquid Inlet Temperature.

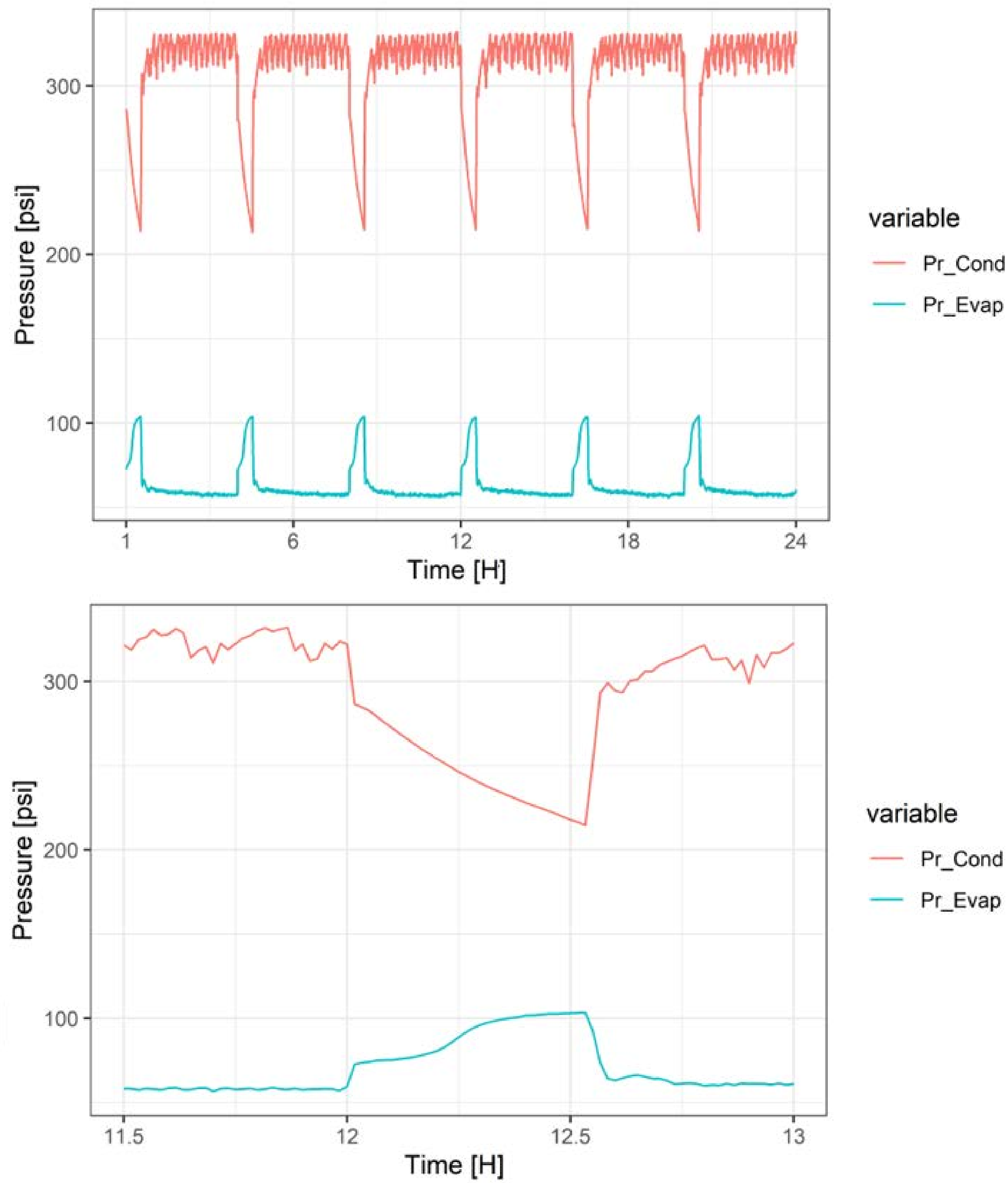

Figure 53. Refrigerant Piping Pressures During ECM Evaluation at 108 °F Liquid Inlet Temperature.

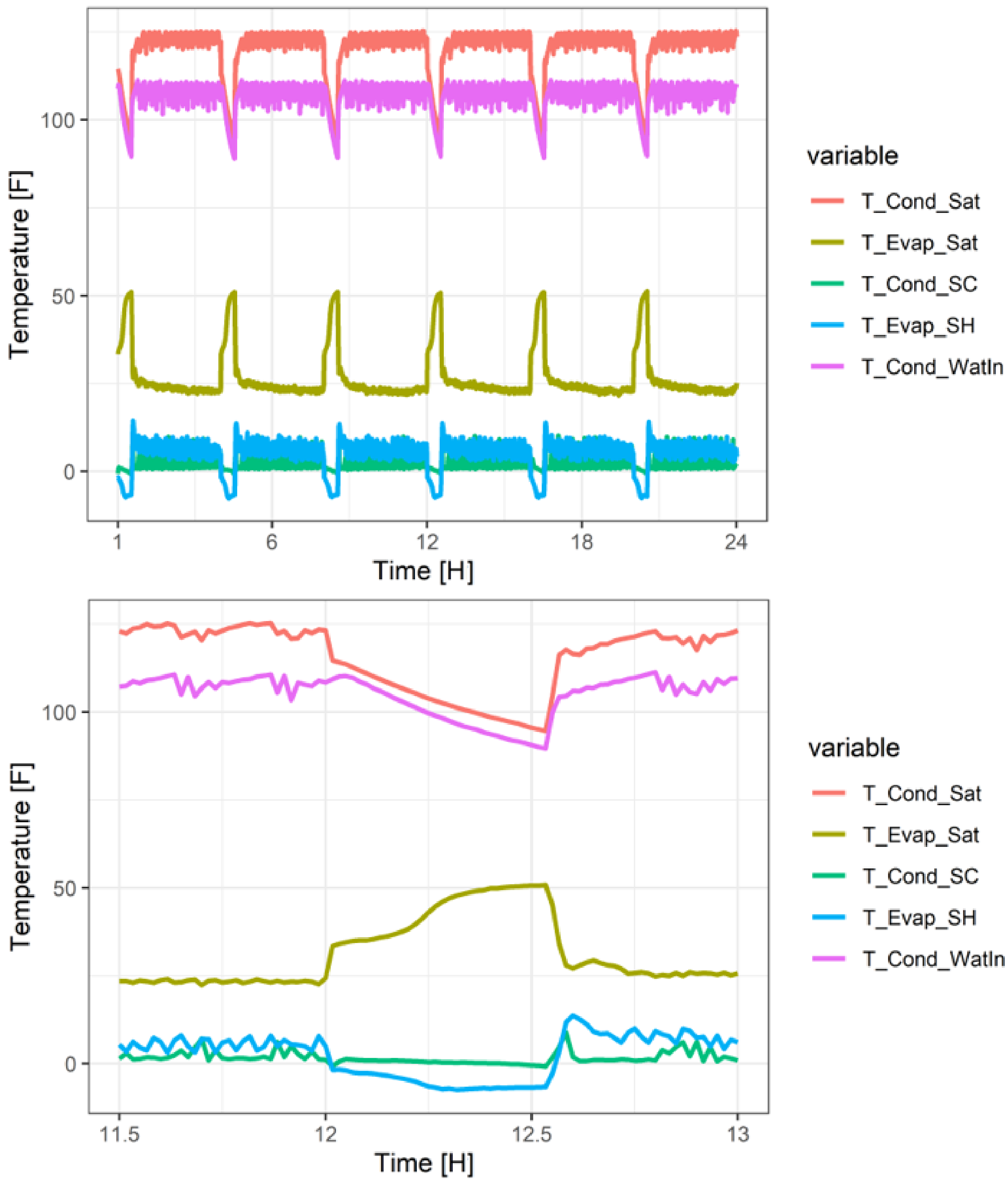

Figure 54. Calculated Refrigerant Piping Temperatures During ECM Evaluation at 108 °F Liquid Inlet Temperature.

Appendix D. Condenser Air (Baseline) and Liquid (ECM) Measurements

Air temperature was measured at the inlet and outlet to the baseline condenser, and liquid temperature was measured at the inlet and outlet to the ECM condenser as shown in Figure 15. Thermocouple probes were instrumented to the intake side of the baseline heat exchanger and at the outlet of the fan on the condenser's discharge side. Liquid temperature probes were tapped in thermocouple wells upstream of the bypass line on the inlet and outlet side of the flow-measurement apparatus. Coriolis flow measurements, and inlet and outlet pressures measured using the flow-measurement apparatus will also be provided in the following appendix sections.

The FCM controls the condenser inlet temperature by adjusting valves to two heat exchangers with a chiller and boiler. Piping between the FCM and environmental chamber is around 50 feet in length. Due to accumulated piping resistances at this length, a lag in response to the FCM controls causes fluctuation in temperature that is unable to be reduced below ~3 %. The average temperature of this fluctuation was used to match the setpoint liquid inlet temperature and was maintained as close as possible to the setpoint across the 24-hour evaluation. For the following Appendix D figures, the nomenclature is:

- "T_A_DB" is the dry bulb temperature at the chamber T_A location
- "T_Cond_FanOut" and "T_Cond_FanIn" are the outlet and inlet air temperatures, respectively, to the air-cooled baseline condenser coil
- "T_Cond_WatOut" and "T_Cond_WatIn" are the outlet and inlet water temperatures, respectively, to the water-cooled ECM condenser coil
- "Pr_Cond_WatIn" and "Pr_Cond_WatOut" are the inlet and outlet water pressures, respectively, to the water-cooled ECM condenser coil
- "FR_Cond" is the water flowrate to the water-cooled ECM condenser coil.

D.1 Baseline Condenser Air Temperature Measurements

The baseline condenser air temperatures are shown below in Figure 55. Air temperatures are shown across the full 24-h test period, as well as a zoomed-in 1.5-h period around hour 12.

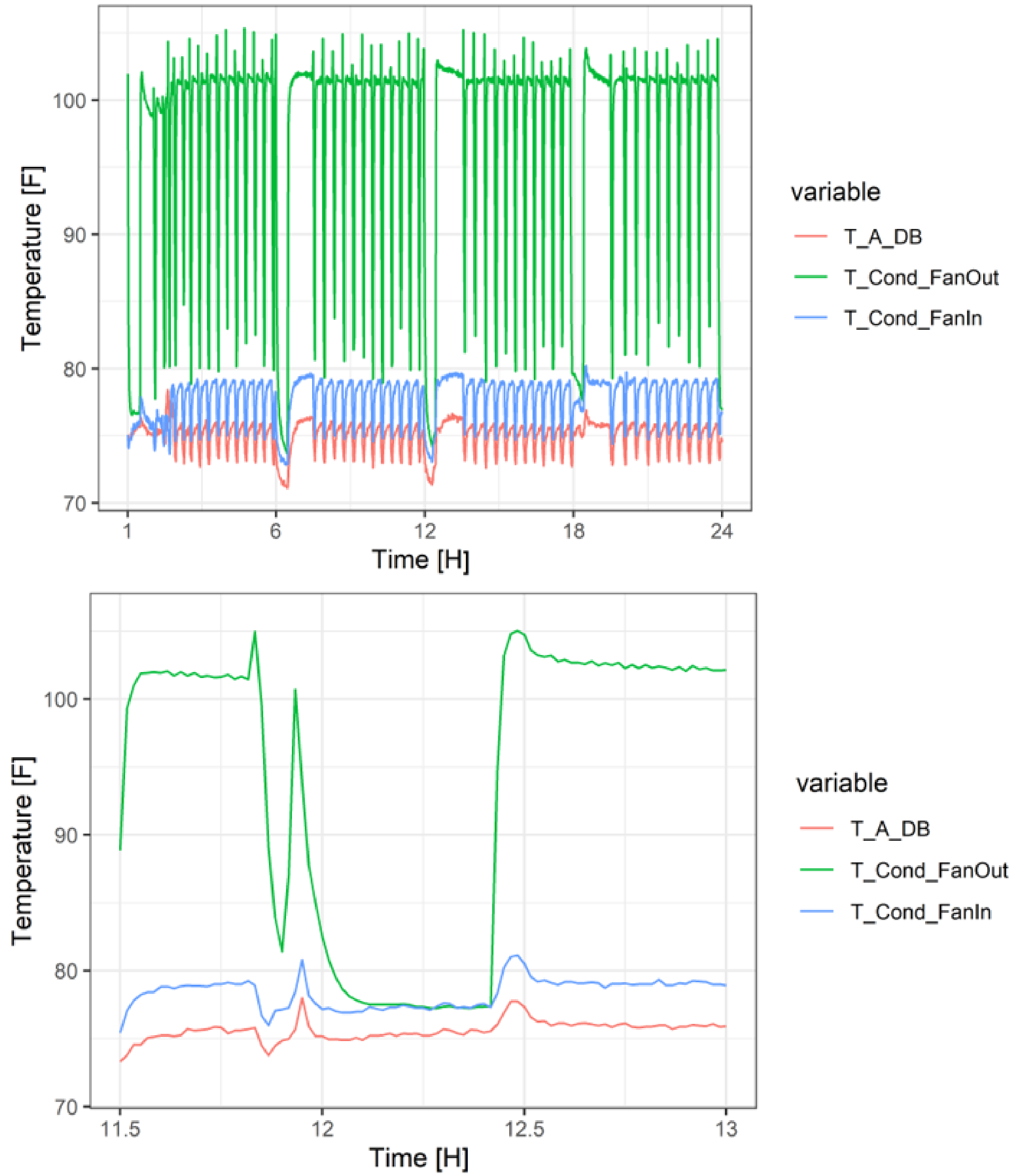

Figure 55. Baseline Condensing Unit Inlet and Outlet Air Temperatures at the Condenser

D.2 ECM Condenser Water Measurements – 55 °F Inlet Setpoint Temp

The ECM condenser liquid inlet and outlet temperatures, pressures, and inlet flowrate at the 55 °F inlet setpoint temperature are shown below in the following figures. Inlet pressure and flowrate were used to calculate hydraulic power across the condenser used to estimate pump power. Inlet and outlet temperatures are shown in Figure 56. Inlet and outlet pressure is shown in Figure 57, and flowrate is shown in Figure 58. Measurements are shown across the full 24-h cycle as well as across a zoomed-in 1.5-h period.

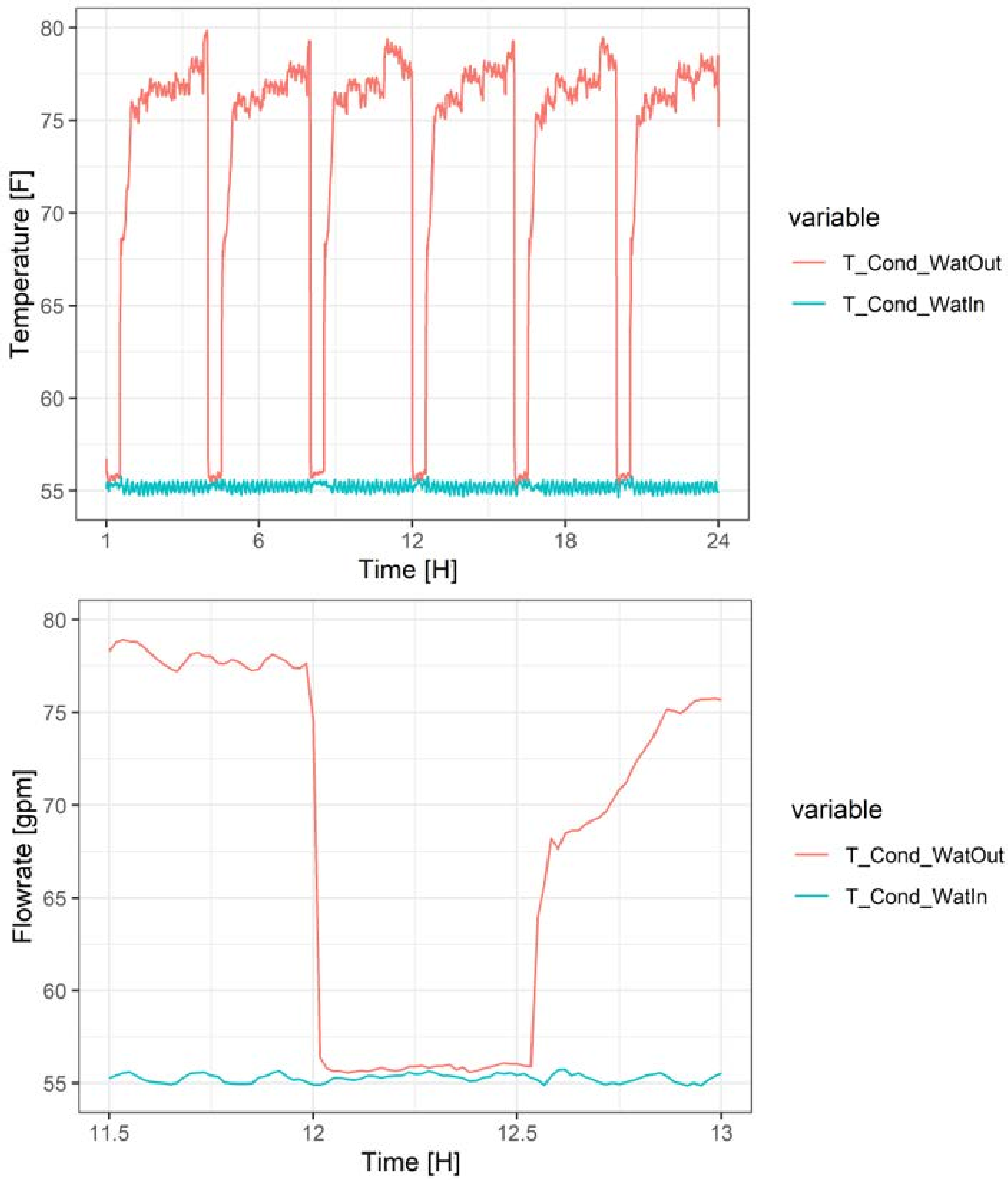

Figure 56. ECM Condenser Inlet and Outlet Water Temperature at 55 °F Setpoint.

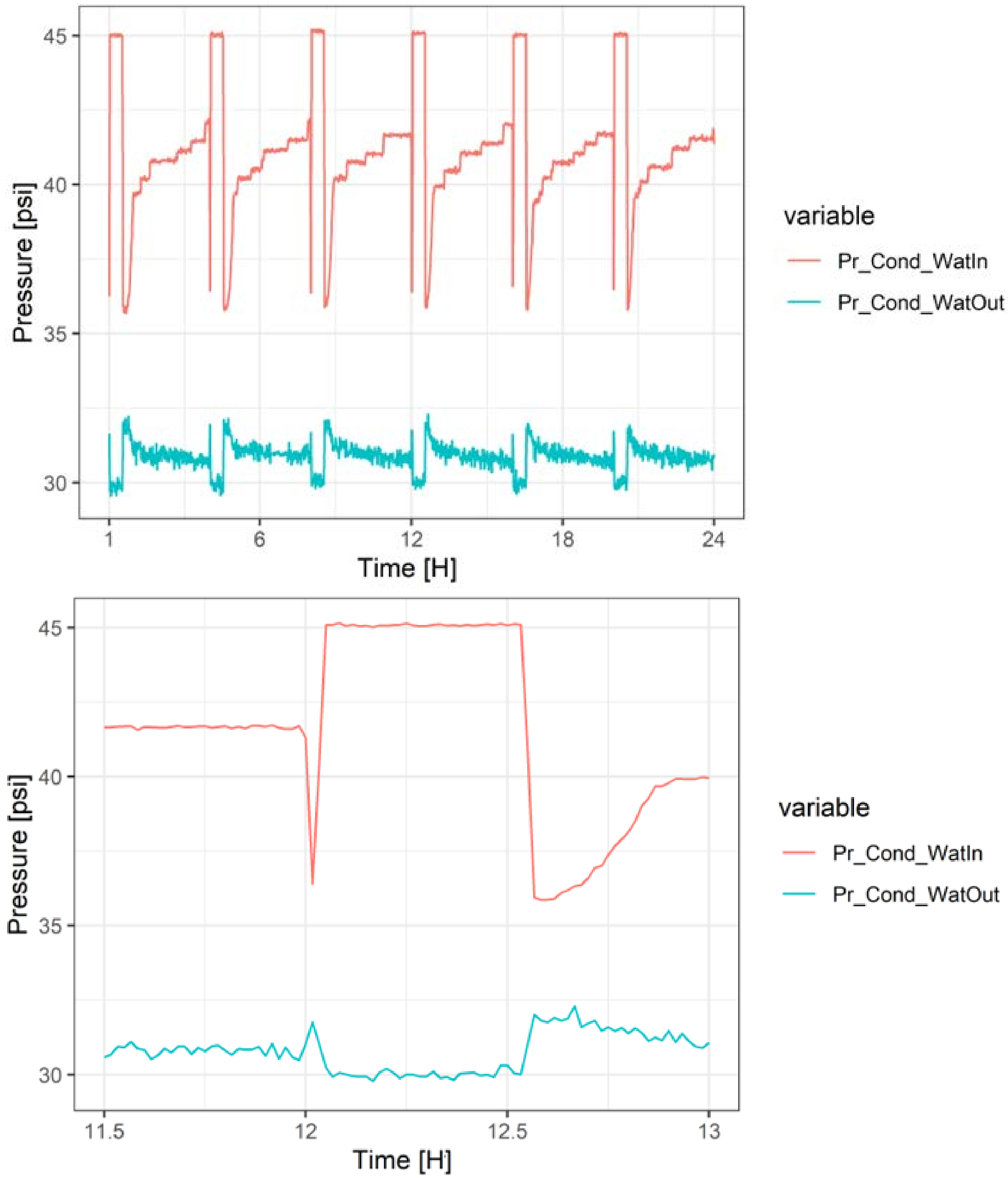

Figure 57. ECM Condenser Inlet and Outlet Water Pressure at 55 °F Setpoint Liquid Inlet Temperature.

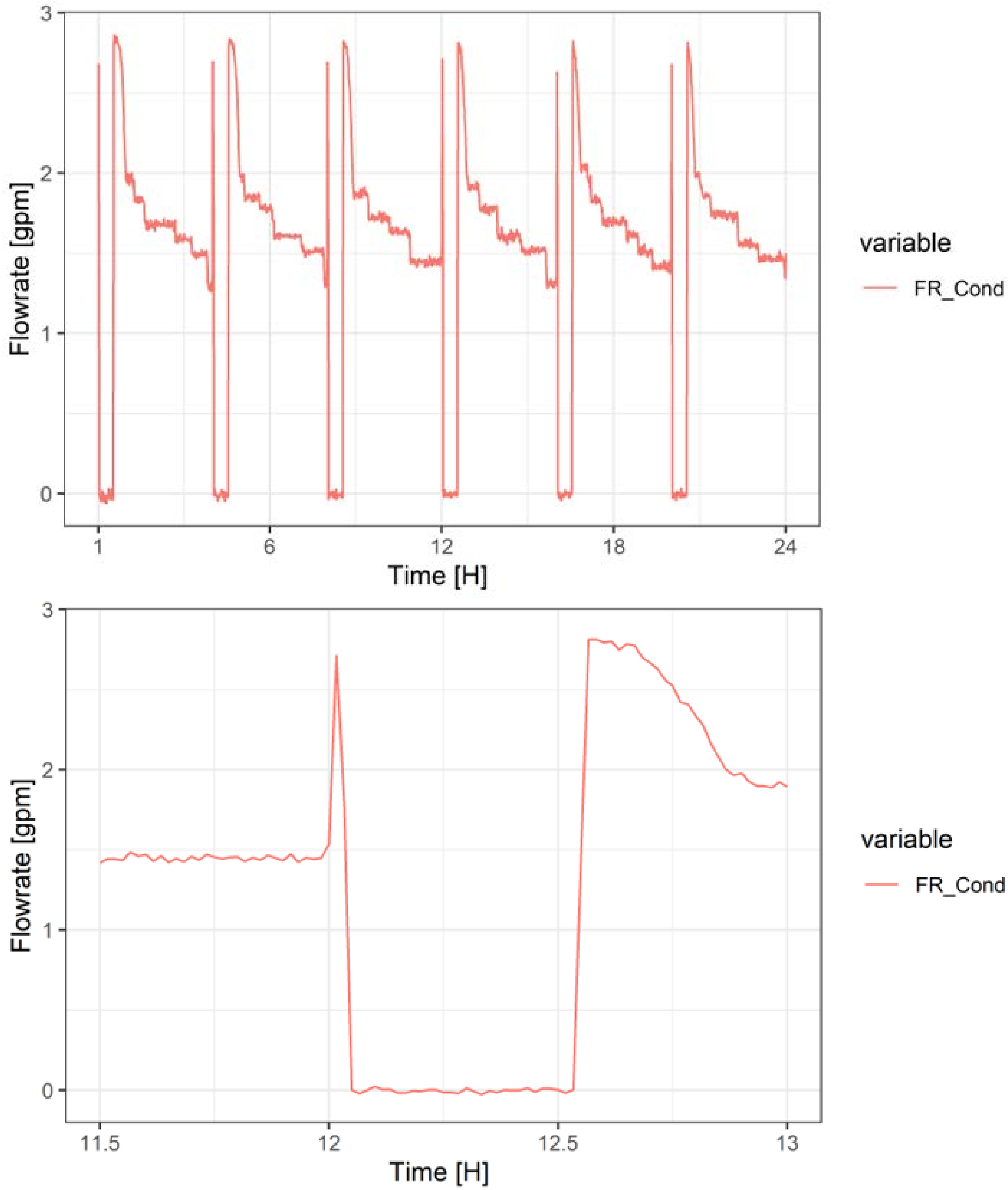

Figure 58. ECM Condenser Water Flowrate at 55 °F Setpoint Liquid Inlet Temperature

D.3 ECM Condenser Water Measurements – 80 °F Inlet Setpoint Temp

The ECM condenser liquid inlet and outlet temperatures, pressures, and inlet flowrate at the 80 °F inlet setpoint temperature are shown below in the following figures. Inlet pressure and flowrate were used to calculate hydraulic power across the condenser used to estimate pump power. Inlet and outlet temperatures are shown in Figure 59. Inlet and outlet pressure is shown in Figure 60, and flowrate is shown in Figure 61. Measurements are shown across the full 24-h cycle as well as across a zoomed-in 1.5-h period.

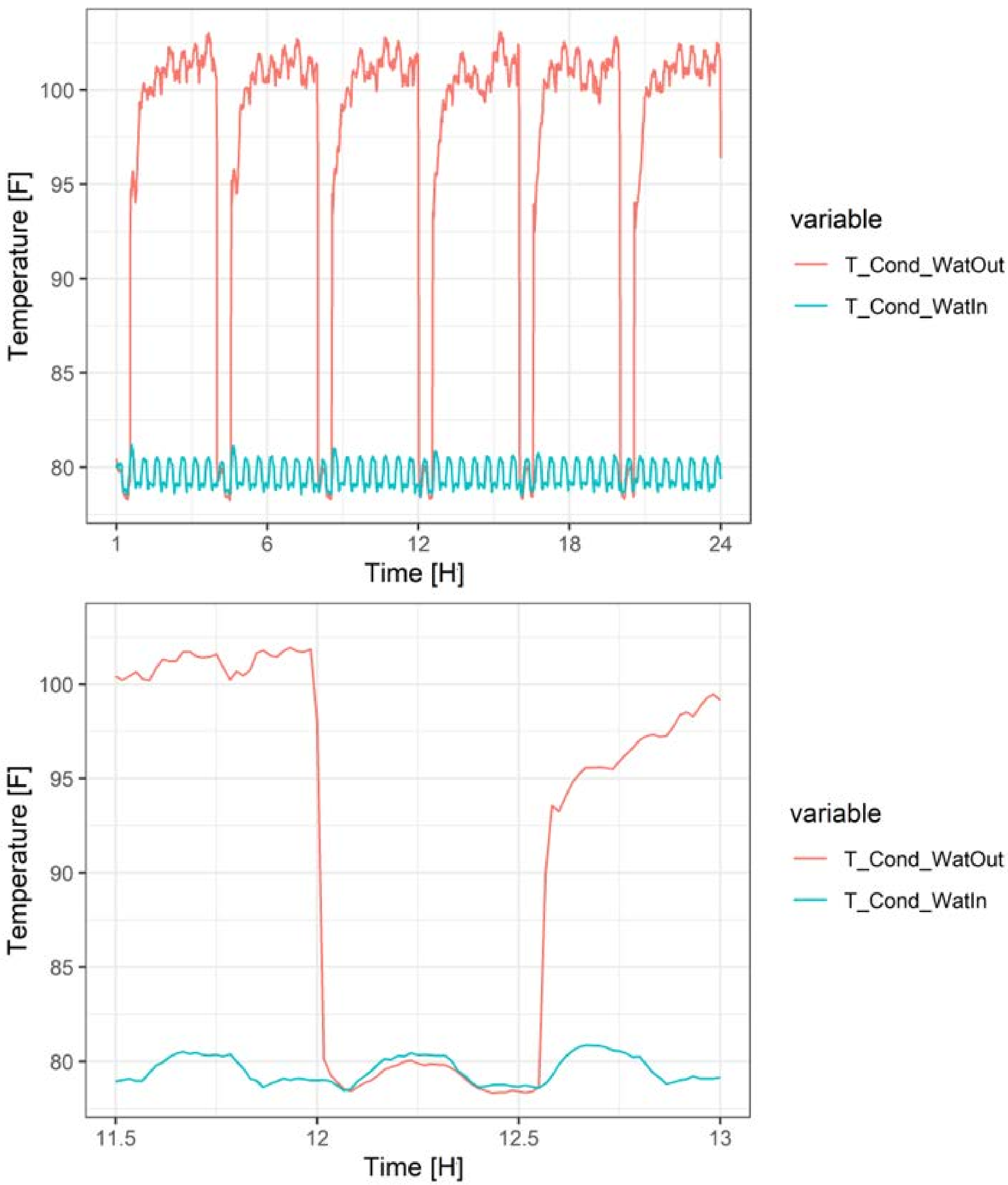

Figure 59. ECM Condenser Inlet and Outlet Water Temperature at 80 °F Setpoint.

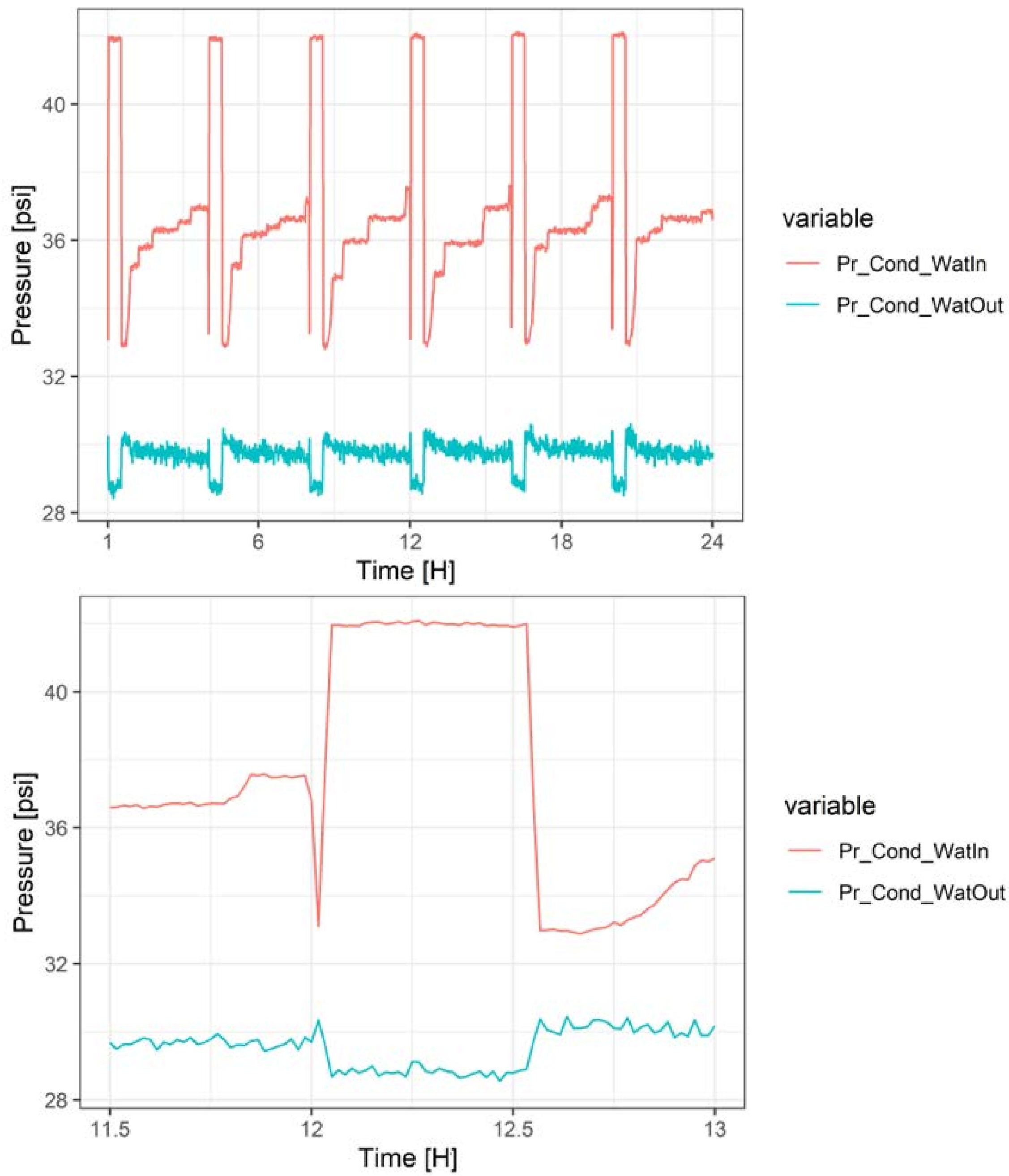

Figure 60. ECM Condenser Inlet and Outlet Water Pressure at 80 °F Setpoint Liquid Inlet Temperature.

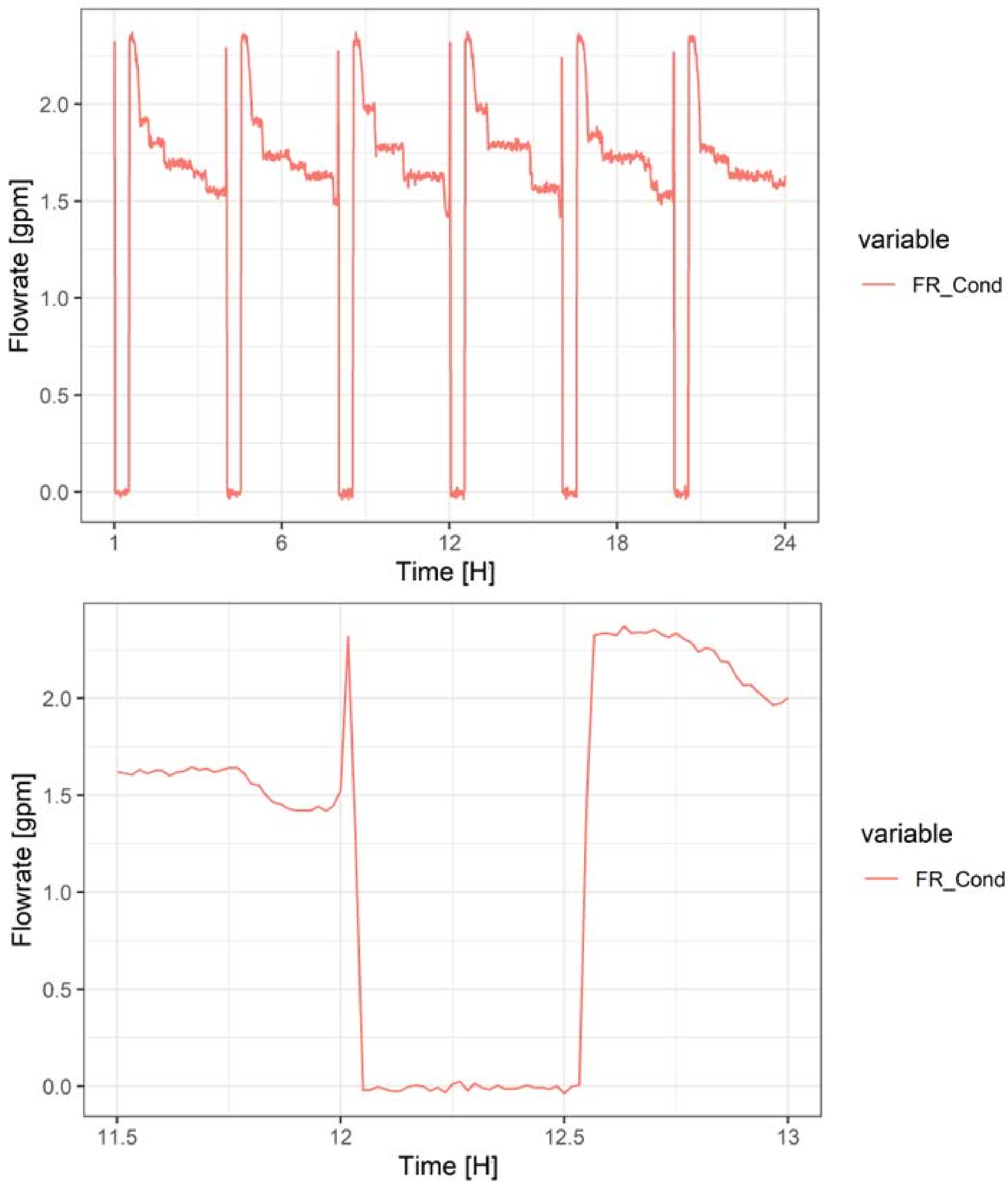

Figure 61. ECM Condenser Water Flowrate at 80 °F Setpoint Liquid Inlet Temperature

D.4 ECM Condenser Water Measurements – 108 °F Inlet Setpoint Temp

The ECM condenser liquid inlet and outlet temperatures, pressures, and inlet flowrate at the 108 °F inlet setpoint temperature are shown below in the following figures. Inlet pressure and flowrate were used to calculate hydraulic power across the condenser used to estimate pump power. Inlet and outlet temperatures are shown in Figure 62. Inlet and outlet pressure is shown in Figure 63, and flowrate is shown in Figure 64. Measurements are shown across the full 24-h cycle as well as across a zoomed-in 1.5-h period.

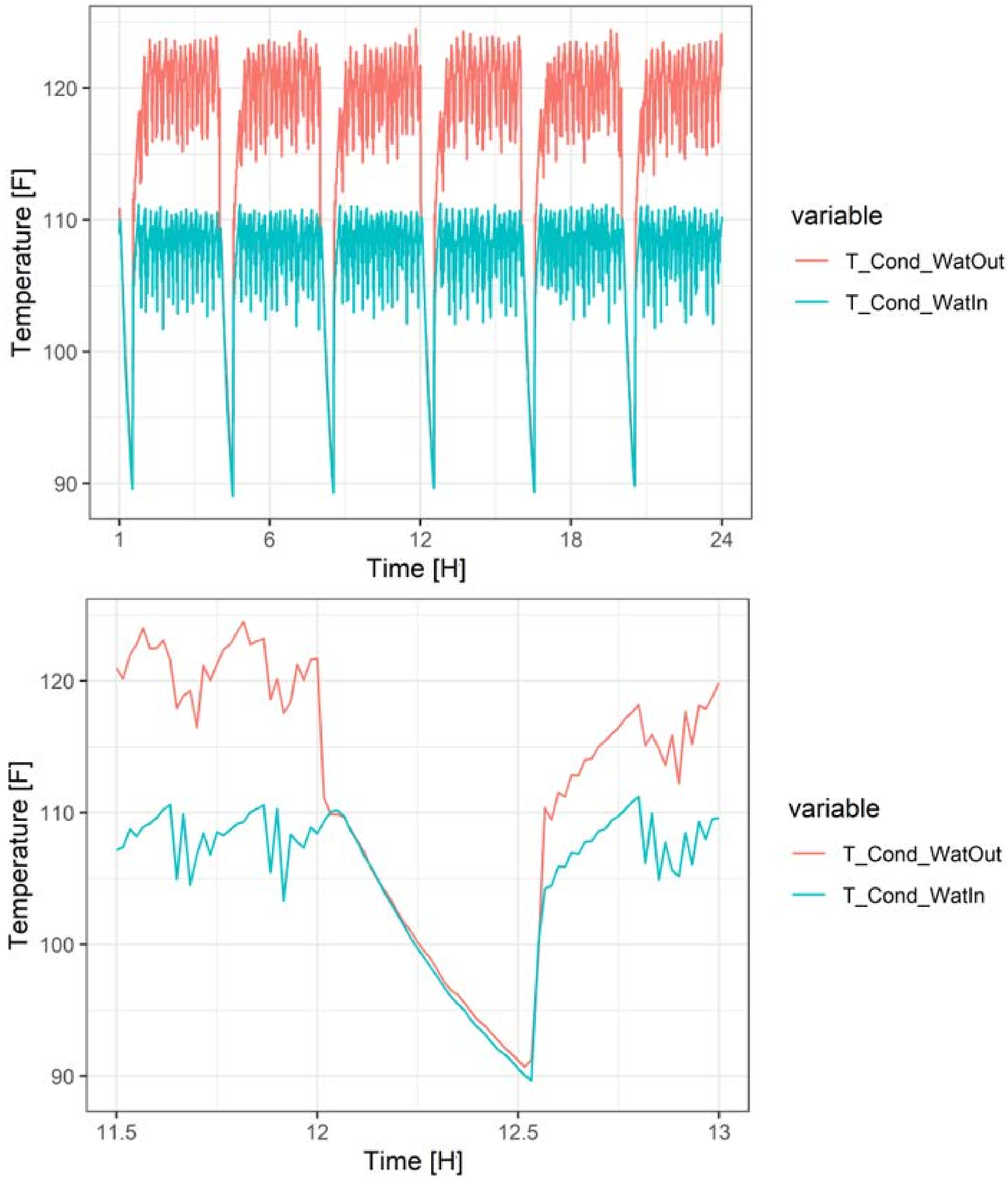

Figure 62. ECM Condenser Inlet and Outlet Water Temperature at 108 °F Setpoint.

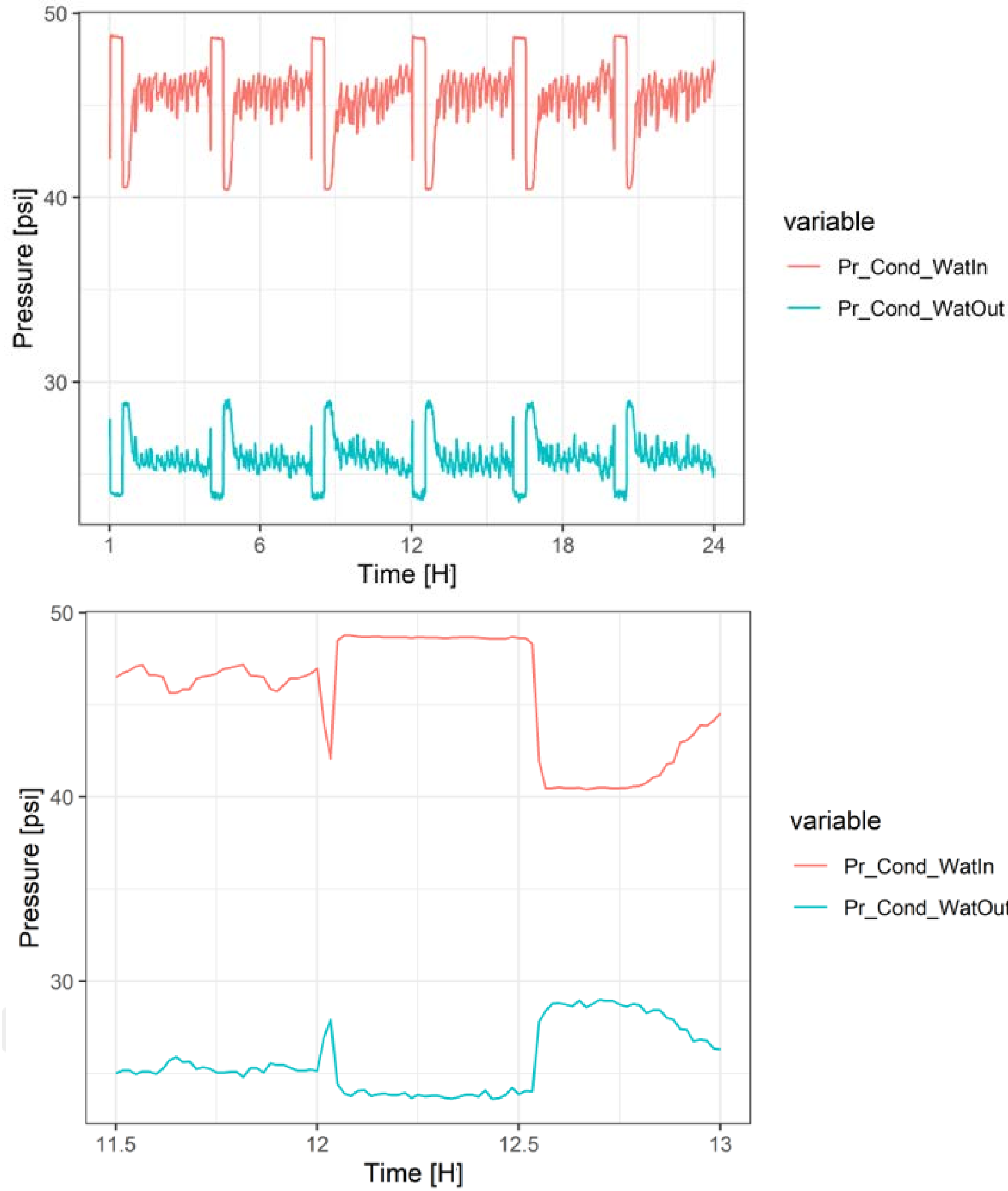

Figure 63. ECM Condenser Inlet and Outlet Water Pressure at 108 °F Setpoint Liquid Inlet Temperature.

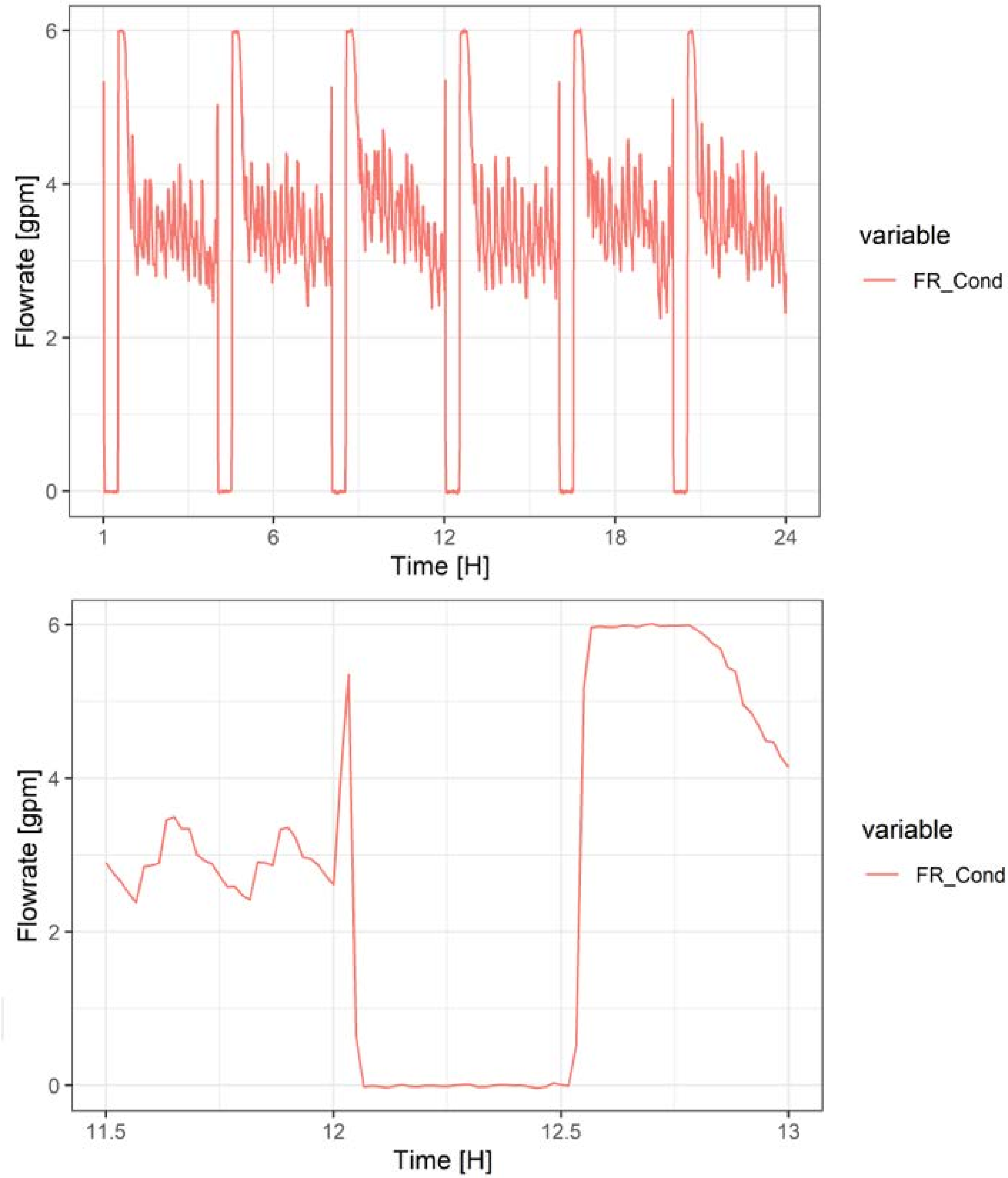

Figure 64. ECM Condenser Water Flowrate at 108 °F Setpoint Liquid Inlet Temperature

Appendix E. Results and Conditions Summary

A summary of the resultant daily energy and mean power consumption, as well as a summary of environmental chamber and case conditions is provided in this appendix. Complete tables show total and component energy and power data during compressor cycling, off-cycles, and during defrost, as well as calculated hydraulic power across the condenser. Environmental chamber conditions, air and liquid condenser conditions, and case air temperatures are provided as averages isolated only to periods of compressor cycling, off-cycling, and defrost as well.

E.1 Results Summary

The summary of results for the baseline evaluation are shown in Table 11. The summary of results for the ECM evaluations are shown in Table 12, Table 13, and Table 14 for condenser liquid inlet temperatures of 55 °F, 80 °F, and 108 °F, respectively.

Table 11. Summary of Results from Baseline Evaluation During Compressor Cycling, Off-Cycling, and Defrost

	Total Energy Consumption (kWh)	Total Compressor Energy (kWh)	Total Condenser Fan Energy (kWh)	Total Evaporator Fan Energy (kWh)	Total Lighting Energy (kWh)	Mean Controller Energy (kWh)
Compressor On-Cycle	46.4	41.5	2.5	0.9	0.9	0.0
Compressor Off-Cycle	0.4	0.0	0.0	0.1	0.1	0.0
Defrost Cycle	0.2	0.0	0.0	0.1	0.1	0.0
Full Evaluation	47.0	41.6	2.5	1.2	1.1	0.1

	Mean Total Power (W)	Mean Component Sum Power (W)	Mean Compressor Power (W)	Mean Condenser Fan Power (W)	Mean Evaporator Fan Power (W)	Mean Lighting Power (W)	Mean Controller Power (W)
Compressor On-Cycle	2461.7	2436.2	2206.2	133.3	48.7	45.5	2.6
Compressor Off-Cycle	120.5	112.4	15.0	1.2	48.7	45.5	2.0
Defrost Cycle	107.5	101.4	4.8	0.4	48.2	45.5	2.5
Full Evaluation	1956.5	1935.0	1733.5	104.8	48.6	45.5	2.5

Table 12. Summary of Results from ECM Evaluation at 55 °F Condenser Inlet Temperature During Compressor Cycling, Off-Cycling, and Defrost

	Total Energy Consumption (kWh)	Total Compressor Energy (kWh)	Total Evaporator Fan Energy (kWh)	Total Lighting Energy (kWh)	Mean Controller Energy (kWh)	Condenser Hydraulic Energy (kWh)	Est. Total Energy w/ Cond Pump (kWh)
Compressor On-Cycle	21.1	18.4	1.0	0.9	0.1	0.1	21.2
Defrost Cycle	0.4	0.0	0.2	0.2	0.0	0.0	0.4
Full Evaluation	21.6	18.4	1.2	1.1	0.1	0.1	21.7
	Mean Total Power (W)	Mean Component Sum Power (W)	Mean Compressor Power (W)	Mean Evaporator Fan Power (W)	Mean Lighting Power (W)	Mean Controller Power (W)	Mean Condenser Hydraulic Power (W)
Compressor On-Cycle	1021.4	987.1	889.9	48.2	45.5	3.5	7.0
Defrost Cycle	134.4	104.0	4.9	48.1	45.5	5.5	0.5
Full Evaluation	899.4	865.7	768.2	48.2	45.5	3.8	6.1

Table 13. Summary of Results from ECM Evaluation at 80 °F Condenser Inlet Temperature During Compressor Cycling, Off-Cycling, and Defrost

	Total Energy Consumption (kWh)	Total Compressor Energy (kWh)	Total Evaporator Fan Energy (kWh)	Total Lighting Energy (kWh)	Mean Controller Energy (kWh)	Condenser Hydraulic Energy (kWh)	Est. Total Energy w/ Cond Pump (kWh)
Compressor On-Cycle	30.2	27.4	1.0	0.9	0.1	0.1	30.3
Defrost Cycle	0.4	0.0	0.2	0.2	0.0	0.0	0.4
Full Evaluation	30.6	27.4	1.2	1.1	0.1	0.1	30.7
	Mean Total Power (W)	Mean Component Sum Power (W)	Mean Compressor Power (W)	Mean Evaporator Fan Power (W)	Mean Lighting Power (W)	Mean Controller Power (W)	Mean Condenser Hydraulic Power (W)
Compressor On-Cycle	1457.7	1419.8	1322.7	48.1	45.5	3.5	4.6
Defrost Cycle	134.8	104.4	5.3	48.1	45.5	5.4	0.3
Full Evaluation	1275.8	1238.9	1141.5	48.1	45.5	3.8	4.0

Table 14. Summary of Results from ECM Evaluation at 108 °F Condenser Inlet Temperature During Compressor Cycling, Off-Cycling, and Defrost

	Total Energy Consumption (kWh)	Total Compressor Energy (kWh)	Total Evaporator Fan Energy (kWh)	Total Lighting Energy (kWh)	Mean Controller Energy (kWh)	Condenser Hydraulic Energy (kWh)	Est. Total Energy w/ Cond Pump (kWh)
Compressor On-Cycle	47.6	44.7	1.0	0.9	0.1	0.1	48.2
Defrost Cycle	0.4	0.0	0.2	0.1	0.0	0.0	0.4
Full Evaluation	48.0	44.7	1.2	1.1	0.1	0.1	48.7
	Mean Total Power (W)	Mean Component Sum Power (W)	Mean Compressor Power (W)	Mean Evaporator Fan Power (W)	Mean Lighting Power (W)	Mean Controller Power (W)	Mean Condenser Hydraulic Power (W)
Compressor On-Cycle	2288.7	2150.0	1322.7	48.6	45.5	3.6	29.8
Defrost Cycle	136.3	104.4	7.3	48.7	45.5	5.5	2.3
Full Evaluation	2001.7	1238.9	1864.3	48.6	45.5	3.8	26.2

E.2 Conditions Summary

The summary of conditions in the environmental chamber and in the refrigerator case for the baseline evaluation are shown in Table 15. The summary of conditions for the ECM evaluations are shown in Table 16, Table 17, and Table 18 for condenser liquid inlet temperatures of 55 °F, 80 °F, and 108 °F, respectively. Mean condenser heat rejection was not calculated for the baseline as with the ECM because only air temperature and not airflow were measured through the condenser, whereas water flowrate was measured through the ECM condenser.

Table 15. Summary of Case and Environmental Chamber Conditions from Baseline Evaluation During Compressor Cycling, Off-Cycling, and Defrost

	Total Time (min)	Average Cycle Length (min)	Mean T_A Dry Bulb (°F)	Mean T_A Dew Point (°F)	Mean T_B Dry Bulb (°F)
Compressor On-Cycle	1130	37.1	75.5	58.2	69.5
Compressor Off-Cycle	178	4.4	73.7	57.1	67.1
Defrost Cycle	132	30.7	74.2	59.8	69.2
Full Evaluation	1440	NA	75.1	58.2	69.2

	Max Average Product Simulator Temp (°F)	Min Average Product Simulator Temp (°F)	Mean Average Product Simulator Temp (°F)	Mean Curtain Discharge Temp (°F)	Mean Curtain Return Temp (°F)
Compressor On-Cycle	39.9	37.5	38.4	34.6	51.1
Compressor Off-Cycle	39.4	37.5	38.1	36.9	52.0
Defrost Cycle	38.5	37.5	37.9	48.1	57.7
Full Evaluation	39.9	37.5	38.3	36.1	51.8

	Mean Condenser Saturation Temp (°F)	Mean Evaporator Saturation Temp (°F)	Mean Condenser Subcool Temp (°F)	Mean Evaporator Superheat Temp (°F)
Compressor On-Cycle	122.8	23.9	1.3	10.4
Compressor Off-Cycle	102.5	34.6	-0.7	2.1
Defrost Cycle	89.2	45.1	0.3	0.4
Full Evaluation	117.2	27.1	1.1	10.1

	Mean Condenser Water Inlet Temperature (°F)	Condensate Produced (lbs)	Mean Condenser Water Inlet Flowrate (gpm)	Mean Condenser Heat Rejection (W)
Compressor On-Cycle	NA	NA	NA	NA
Compressor Off-Cycle	NA	NA	NA	NA
Defrost Cycle	NA	NA	NA	NA
Full Evaluation	NA	110.9	NA	NA

Table 16. Summary of Case and Environmental Chamber Conditions from ECM Evaluation at 55 °F Condenser Inlet Temperature During Compressor Cycling, Off-Cycling, and Defrost

	Total Time (min)	Average Defrost Cycle Length (min)	Mean T_A Dry Bulb (°F)	Mean T_A Dew Point (°F)	Mean T_B Dry Bulb (°F)
Compressor On-Cycle	1242	207.0	75.2	58.4	70.6
Defrost Cycle	198	33.0	75.2	58.9	70.9
Full Evaluation	1440	NA	75.2	58.5	70.6

	Max Average Product Simulator Temp (°F)	Min Average Product Simulator Temp (°F)	Mean Average Product Simulator Temp (°F)	Mean Curtain Discharge Temp (°F)	Mean Curtain Return Temp (°F)
Compressor On-Cycle	38.7	36.5	37.6	33.1	50.0
Defrost Cycle	37.5	36.5	36.8	43.4	54.7
Full Evaluation	38.7	36.5	37.4	34.5	50.6

	Mean Condenser Saturation Temp (°F)	Mean Evaporator Saturation Temp (°F)	Mean Condenser Subcool Temp (°F)	Mean Evaporator Superheat Temp (°F)
Compressor On-Cycle	76.3	24.4	2.6	6.9
Defrost Cycle	68.9	43.6	-10.9	-6.1
Full Evaluation	75.3	27.0	2.6	6.5

	Mean Condenser Water Inlet Temperature (°F)	Condensate Produced (lbs)	Mean Condenser Water Inlet Flowrate (gpm)	Mean Condenser Heat Rejection (W)
Compressor On-Cycle	55.2	NA	1.7	17919.5
Defrost Cycle	55.3	NA	0.2	286.0
Full Evaluation	55.2	105.8	1.5	15494.8

Table 17. Summary of Case and Environmental Chamber Conditions from ECM Evaluation at 80 °F Condenser Inlet Temperature During Compressor Cycling, Off-Cycling, and Defrost

	Total Time (min)	Average Defrost Cycle Length (min)	Mean T_A Dry Bulb (°F)	Mean T_A Dew Point (°F)	Mean T_B Dry Bulb (°F)
Compressor On-Cycle	1242	207.0	75.2	58.1	70.5
Defrost Cycle	198	33.0	75.2	58.7	70.8
Full Evaluation	1440	NA	75.2	58.2	70.5

	Max Average Product Simulator Temp (°F)	Min Average Product Simulator Temp (°F)	Mean Average Product Simulator Temp (°F)	Mean Curtain Discharge Temp (°F)	Mean Curtain Return Temp (°F)
Compressor On-Cycle	39.1	37.0	38.2	33.7	50.6
Defrost Cycle	37.8	37.0	37.3	43.9	55.0
Full Evaluation	39.1	37.0	38.1	35.1	51.2

	Mean Condenser Saturation Temp (°F)	Mean Evaporator Saturation Temp (°F)	Mean Condenser Subcool Temp (°F)	Mean Evaporator Superheat Temp (°F)
Compressor On-Cycle	99.6	25.2	2.1	6.7
Defrost Cycle	83.8	44.1	-2.6	-5.5
Full Evaluation	97.5	27.8	2.1	6.3

	Mean Condenser Water Inlet Temperature (°F)	Condensate Produced (lbs)	Mean Condenser Water Inlet Flowrate (gpm)	Mean Condenser Heat Rejection (W)
Compressor On-Cycle	79.6	NA	1.8	18103.5
Defrost Cycle	79.4	NA	0.1	35.4
Full Evaluation	79.6	113.5	1.5	15619.1

Table 18. Summary of Case and Environmental Chamber Conditions from ECM Evaluation at 108 °F Condenser Inlet Temperature During Compressor Cycling, Off-Cycling, and Defrost

	Total Time (min)	Average Defrost Cycle Length (min)	Mean T_A Dry Bulb (°F)	Mean T_A Dew Point (°F)	Mean T_B Dry Bulb (°F)
Compressor On-Cycle	1248	208.0	75.0	58.8	72.5
Defrost Cycle	192	32.0	75.0	59.3	72.7
Full Evaluation	1440	NA	75.2	58.9	72.6
	Max Average Product Simulator Temp (°F)	**Min Average Product Simulator Temp (°F)**	**Mean Average Product Simulator Temp (°F)**	**Mean Curtain Discharge Temp (°F)**	**Mean Curtain Return Temp (°F)**
Compressor On-Cycle	38.9	37.0	38.0	33.6	51.5
Defrost Cycle	37.6	36.9	37.2	43.5	55.3
Full Evaluation	38.9	36.9	37.9	35.0	52.0
	Mean Condenser Saturation Temp (°F)	**Mean Evaporator Saturation Temp (°F)**	**Mean Condenser Subcool Temp (°F)**	**Mean Evaporator Superheat Temp (°F)**	
Compressor On-Cycle	122.7	24.1	1.6	6.5	
Defrost Cycle	103.4	43.6	0.5	-6.6	
Full Evaluation	120.1	26.7	1.4	6.0	
	Mean Condenser Water Inlet Temperature (°F)	**Condensate Produced (lbs)**	**Mean Condenser Water Inlet Flowrate (gpm)**	**Mean Condenser Heat Rejection (W)**	
Compressor On-Cycle	108.1	NA	3.7	20475.0	
Defrost Cycle	98.9	NA	0.3	69.2	
Full Evaluation	106.9	118.9	3.3	17754.2	