End-Use Savings Shapes Measure Documentation:

Boiler Replacement with Air-Source Heat Pump Boiler and Electric Boiler Backup

Korbaga Woldekidan

National Renewable Energy Laboratory

NREL is a national laboratory of the U.S. Department of Energy
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Technical Report
NREL/TP-5500-86199
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Cover Photos by Dennis Schroeder: (clockwise, left to right) NREL 51934, NREL 45897, NREL 42160, NREL 45891, NREL 48097, NREL 46526.

List of Acronyms

ANSI	American National Standards Institute
ASHP	air-source heat pump
COP	coefficient of performance
DHL	design heating load
DOAS	dedicated outdoor air system
EIR	energy efficiency ratio
EULP	end-use load profiles
EUSS	end-use savings shape
HDT	heating design temperature
HVAC	heating, ventilating, and air conditioning
IES	Illuminating Engineering Society
LRMER	Long-Run Marginal Emissions Rate
NRCan	National Resources Canada
OAT	outdoor air temperature
PLR	part load ratio
PSZ-AC	packaged single zone air conditioner
PVAV	packaged variable air volume
VAV	variable air volume

Executive Summary

Building on the successfully completed effort to calibrate and validate the U.S. Department of Energy's ResStock™ and ComStock™ models over the past three years, the objective of this work is to produce national data sets that empower analysts working for federal, state, utility, city, and manufacturer stakeholders to answer a broad range of analysis questions.

The goal of this work is to develop energy efficiency, electrification, and demand flexibility end-use load shapes (electricity, gas, propane, or fuel oil) that cover most of the high-impact, market-ready (or nearly market-ready) measures. "Measures" refers to energy efficiency variables that can be applied to buildings during modeling.

An *end-use savings shape* is the difference in energy consumption between a baseline building and a building with an energy efficiency, electrification, or demand flexibility measure applied. It results in a time-series profile that is broken down by end use and fuel (electricity or on-site gas, propane, or fuel oil use) at each timestep.

ComStock is a highly granular, bottom-up model that uses multiple data sources, statistical sampling methods, and advanced building energy simulations to estimate the annual sub hourly energy consumption of the commercial building stock across the United States. The baseline model intends to represent the U.S. commercial building stock as it existed in 2018. The methodology and results of the baseline model are discussed in the final technical report of the End-Use Load Profiles project.

This document focuses on a single end-use savings shape measure—natural gas-fired boiler replaced with an air-source heat pump boiler. When the operating outdoor air temperature is below the cutoff temperature of the heat pump boiler, an electric boiler was used as a backup. Application of this measure helps to quantify the decarbonization as well as the energy savings potential from the replacement.

This measure is applicable for 33% of the U.S. commercial building stock modeled in ComStock and the following key observations were made.

- 7.5 % total site energy savings (348 TBtu)
- 63.1% heating natural gas savings (521.8 TBtu)
- 87.1% increase in heating electricity (172.1 TBtu).

Acknowledgments

The authors would like to acknowledge the valuable guidance and input provided by Shanti Pless and the ComStock team, particularly Andrew Parker and Christopher CaraDonna.

The author would also like to thank Bart Ransom from Colmac for providing heat pump performance data.

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1 Boiler Replacement with Air-Source Heat Pump Boiler with Electric Boiler Backup

1.1 Accessing Results

This documentation covers the "Replace Boiler by Air-Source Heat Pump Boiler" upgrade methodology and briefly discusses key results. Results can be accessed on the ComStock™ data lake at "end-use-load-profiles-for-us-building-stock" or via the Data Viewer at comstock.nrel.gov.

1.2 Measure Summary

Measure Title	Replace Boiler by Air-Source Heat Pump Boiler (replace_boiler_by_heatpump)
Measure Definition	This measure replaces a natural gas boiler by an air-source heat pump for space heating.
Applicability	Buildings that use natural gas boiler for HVAC system.
Not Applicable	Buildings that do not use natural gas boiler for space heating such as those with furnace, electric heaters, or district heat source. Boilers used for domestic hot water heating and supplemental heating in heat pump condenser loops will not be changed in this measure.
Release	2023 Release 1

2 Technology Summary

Air-source heat pump (ASHP) boilers are one of several candidate technologies for boiler electrification and achieving climate goals. This technology uses electricity to move heat from the surrounding air and transfer it at a higher temperature for space heating application. As indicated in Figure 1, ASHP boilers work as refrigeration in reverse and are usually two to three times more efficient than electric resistance heaters. Because the surrounding air serves as a heat source, the performance of ASHPs depends on the heat content (temperature) of the outdoor air. Heat pump equipment is often controlled to disable operation below a specified outdoor air temperature, often called the cutoff temperature or compressor lockout temperature. The specific value can vary by equipment type and manufacturer, and sometimes user is able to specify a value between a range limit. For operation below the cutoff temperature, backup boilers are used. Similar to heat pump performance and capacity retention, the compressor lockout temperature can improve with technology development.

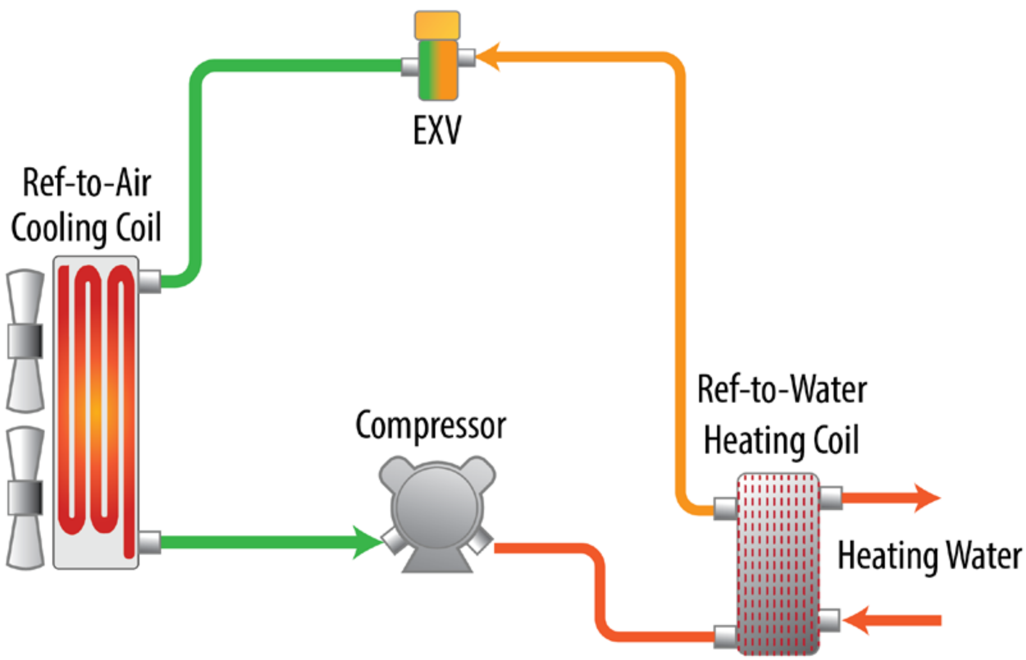

Figure 1. ASHP boiler operation

3 ComStock Baseline Approach

The current version of the boilers in ComStock are gas-fired, noncondensing boilers. Their efficiencies are determined using the U.S. Department of Energy's reference buildings templates and capacities. The values are summarized in Table 1 [3]. To adjust the boiler efficiency, three different cubic performance curves are utilized based on the part load ratio (PLR) as shown in Table 2. Figure 2 shows a graphical representation of the curves output. All ComStock boilers have a heating set point of 180°F with a capacity to modulate flow based on the heating load.

Table 1. Boiler Efficiency and Performance Curve Assignment

Template	Minimum Capacity (Btu/hr)	Maximum Capacity (Btu/hr)	Minimum Annual Fuel Utilization Efficiency (AFUE)	Minimum Thermal Efficiency (%)	Minimum Com-bustion Efficiency (%)	Efficiency Function of Part Load Ratio (EFFFPLR)	Notes
Pre-1980	-	299,999		0.73		Boiler Constant Efficiency Curve	From DOE Reference Buildings
Pre-1980	300,000	no max		0.74			
Pre-1980	250,000,000	249,999,999		0.76			
1980-2004	-	299,999	0.8				From 90.1-1989
1980-2004	300,000	249,999,999			0.8		
90.1-2004	-	299,999	0.8			Boiler with No Minimum Turndown	From 90.1-2004
90.1-2004	300,000	249,999,999		0.75			
90.1-2004	250,000,000	no max			0.8		
90.1-2007	-	299,999	0.8				From 90.1-2007
90.1-2007	300,000	249,999,999		0.8			
90.1-2007	250,000,000	no max			0.82		
90.1-2010	-	299,999	0.8				From 90.1-2010
90.1-2010	300,000	249,999,999		0.8			
90.1-2010	250,000,000	no max			0.82		
90.1-2013	-	299,999	0.82				From 90.1-2013 From 90.1-2016
90.1-2013	300,000	999,999		0.8			
90.1-2013	1,000,000	249,999,999		0.8		Boiler with Minimum Turndown	
90.1-2013	250,000,000	no max			0.82		
90.1-2016	-	299,999	0.82			Boiler with No Minimum Turndown	
90.1-2016	300,000	999,999		0.8			
90.1-2016	1,000,000	249,999,999		0.8		Boiler with Minimum Turndown	
90.1-2016	250,000,000	no max			0.82		
90.1-2019	-	299,999	0.84			Boiler with No Minimum Turndown	From 90.1-2019
90.1-2019	300,000	999,999		0.8			
90.1-2019	1,000,000	249,999,999		0.8		Boiler with Minimum Turndown	
90.1-2019	250,000,000	no max			0.82		

Table 2. Boiler Performance Curves

Name	Form	Dependent Variable	Independent Variable 1	coeff_1	coeff_2	coeff_3	coeff_4	Notes
Boiler Constant Efficiency Curve	Cubic	Efficiency Multiplier	Part Load Ratio	1	0	0	0	From DOE Reference Building
Boiler with Minimum Turndown				0.7791	1.4745	-2.5795	1.3467	From Regression of Prototype Building EMS
Boiler with No Minimum Turndown				0.7463	1.3196	-2.2154	1.1674	From Regression of Prototype Building EMS

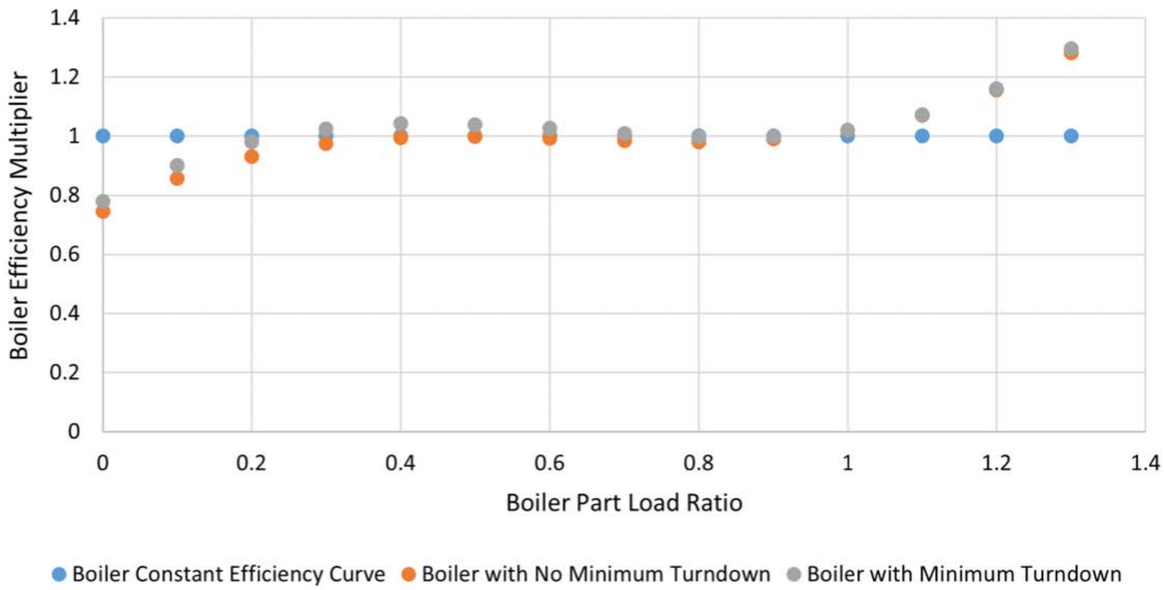

Figure 2. Boiler performance curves

4 Modeling Approach

According to the Commercial Buildings Energy Use Survey (CBECS), natural gas used by boilers and furnaces accounts for 73% of space heating energy consumption in U.S. commercial buildings [4]. Electrifying these heat sources would have a considerable impact in achieving climate coals. This measure replaces natural gas boilers for HVAC applications by heat pump boilers. Outputs from the simulation runs could be used to quantify the carbon reduction and energy impact of the electrification.

The measure provides multiple options for the natural gas boiler replacement. Table 3 summarizes the measure inputs and their default values used in the simulation run.

Table 3. Measure Input Summary

Measure Inputs	Description	Default Value	Units
Keep_setpoint	Provides an option to keep the original hot water set point.	False	True/False
hw_setpoint	Provides a new hot water set point if user chooses to change the original value.	140	˚F
autosize_hc	Provides an opportunity to auto-size heating coils when a user provides a new hot water set point.	True	True/False
Sizing_method	Provides an option for sizing the heat pump. The two options are sizing based on “percentage of peak load” and on “outdoor air temperature.”	Outdoor air temperature	
hp_sizing_temp	Provides the outdoor air temperature on which to base ASHP sizing if user chooses the sizing method as “outdoor air temperature.”	17	˚F
hp_sizing_per	Provides the percentage of the peak heating load on which to base the sizing if user chooses the sizing method as “percentage of peak load.”	70	%
hp_des_cap	Maximum design heat pump heating capacity per unit. If the model requires a higher capacity, multiple units will be added in the loop.	40	kW
bu_type	Provides two options for backup heater: keeping the existing boiler or adding an electric resistance heater.	Electric resistance heater	
hpwh_cutoff_Temp	Provides the cutoff temperature for the heat pump boiler.	−5	˚F
hpwh_Design_OAT	Provides design outdoor air temperature for the heat pump boiler.	47	˚F
COP	Provides the design coefficient of performance (COP) at the design outdoor air temperature.	2.85	

4.1 Applicability

Figure 3 shows the heating type distribution among the ComStock buildings. This measure is applicable for buildings that use gas boiler as a heating source, which accounts for 33% of the ComStock baseline total floor area. The orange bar in the figure indicates portions for the applicable buildings. Out of the buildings with boiler, this measure is applicable for 94% of them; the rest are boilers used for supplemental heating for heat pump applications.

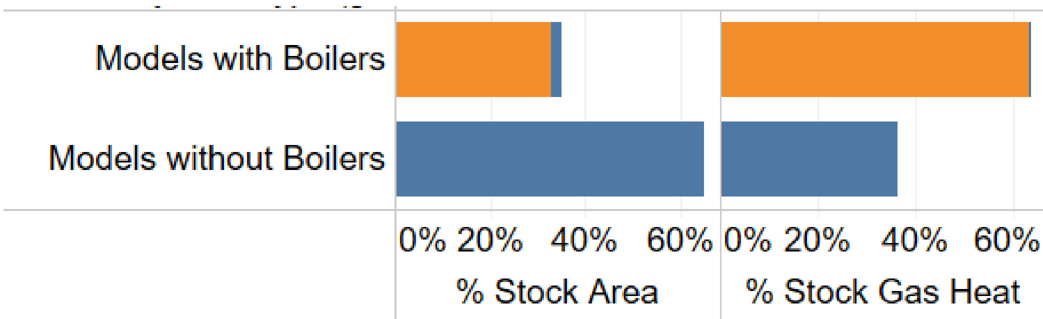

Figure 3. Measure applicability

This measure is applicable to the ComStock OpenStudio® models with the following heating, ventilating, and air conditioning (HVAC) system types (Table 4):

Table 4. Applicable HVAC System Types

	Applicable HVAC System Types
1	Dedicated outdoor air system (DOAS) with fan coil air-cooled chiller with boiler
2	DOAS with fan coil chiller with boiler
3	DOAS with fan coil district chilled water with boiler
4	DOAS with water-source heat pump cooling tower with boiler
5	Packaged single zone air conditioner (PSZ-AC) with gas boiler
6	Packaged variable air volume (PVAV) with gas boiler
7	PVAV with gas heat with electric reheat
8	Variable air volume air-cooled chiller with gas boiler reheat
9	Variable air volume chiller with gas boiler reheat
10	Variable air volume district chilled water with gas boiler reheat

4.2 ASHP Sizing

Heat pump sizing is one of the critical steps that needs to be addressed when retrofitting a boiler with an ASHP boiler. The sizing process requires consideration of several factors, such as: [2]

- Design heating water supply temperature
- Design heating outdoor air temperature
- Equipment cost

- Operating cost
- Electrical infrastructure cost to support the higher peak demand from switching to an electric heating source from a gas-fired heating source
- Carbon emission reduction.

Optimal sizing is a balance of the above factors and should be aligned with the priorities of the building owner. Most of the commercially available ASHP boilers are relatively small and require cascading for higher capacities. Apart from requiring more area for installation, cascading provides flexibility, improves efficiency during part load operation, and enhances system redundancy and resiliency.

4.2.1 Estimation of Design Heating Load

The design heating load (DHL) in a building can be estimated using energy audit load estimates, energy modeling of design loads, or existing equipment capacities. Because the objective of this measure is to evaluate the benefit of replacing an existing boiler, the DHL estimation is based on existing equipment capacity in the model. In practice, using existing capacity is acceptable if [6]:

- No historic comfort issue is reported.
- No major energy performance upgrades have been made since the most recent heating sizing was done.
- Evidence exists that reasonable sizing practices were followed during the design.

For the purpose of sizing, the heating load is assumed to linearly change from zero (at the heating-enable outdoor air temperature) to the design heating load (at the heating design temperature)[6], as shown in Figure 4. In most applications, heating is enabled at an outdoor air temperature of 60°F. The line connecting the zero-heating load and the design heating load is referred to as the heating load line.

The outdoor temperature for the design heating load depends on the climate condition. Table 5 summarizes the winter design day condition corresponding to the design heating load for the representative cities in each ASHRAE climate zone [7] .

Table 5. Winter Design Day Temperature per ASHRAE Climate Zone

Climate Zone	City	Winter Heating Design Day Temperature (˚F)
1A	Miami, FL	47.7
2A	Houston, TX	29.1
2B	Phoenix, AZ	38.7
3A	Atlanta, GA	20.7
3B-Coast	Los Angeles, CA	44.4
3B	Las Vegas, NV	30.6
3C	San Francisco, CA	38.8
4A	Baltimore, MD	12.9
4B	Albuquerque, NM	17.8
4C	Seattle, WA	24.4
5A	Chicago, IL	−4.0
5B	Boulder, CO	0.7
6A	Minneapolis, MN	−13.4
6B	Helena, MT	−15.3
7	Duluth, MN	−19.5
8	Fairbanks, Ak	−43.4

4.2.2 Sizing Options and Target Capacity Estimation

The sizing options depend on the client's expectations for the ASHP boiler installation. In general, the heat pump can be sized in two ways:

- Sizing the ASHP as the primary heating source for a fraction of the design load with the assumption that a backup/existing heating system will be run during the coldest periods (peak loads).
- Sizing the ASHP to provide all or nearly all the heating at the design condition with little or no use of backup heating.

In most scenarios, peaking of the system might only occur 1% of the time. Thus, for retrofit projects with boilers, it can be cost-effective to utilize the existing boilers to handle peaks and size heat pumps for an optimal balance between upfront capital investment costs and carbon emissions [8]. If the sizing is done with the assumption of a backup heater, the National Resources Canada (NRCan) ASHP Sizing and Selection Guide [6] recommends basing the target capacity corresponding to a heating load at 17°F. On the other hand, if the plan is to size the ASHP to provide all or nearly all the heating need, NRCan recommends the target capacity to be the same as the design heating load. Once the target capacity is estimated, an ASHP with a heating capacity close to the target capacity at 17°F (for sizing based on a backup system) or at the heating design temperature (for sizing with no backup system) should be selected.

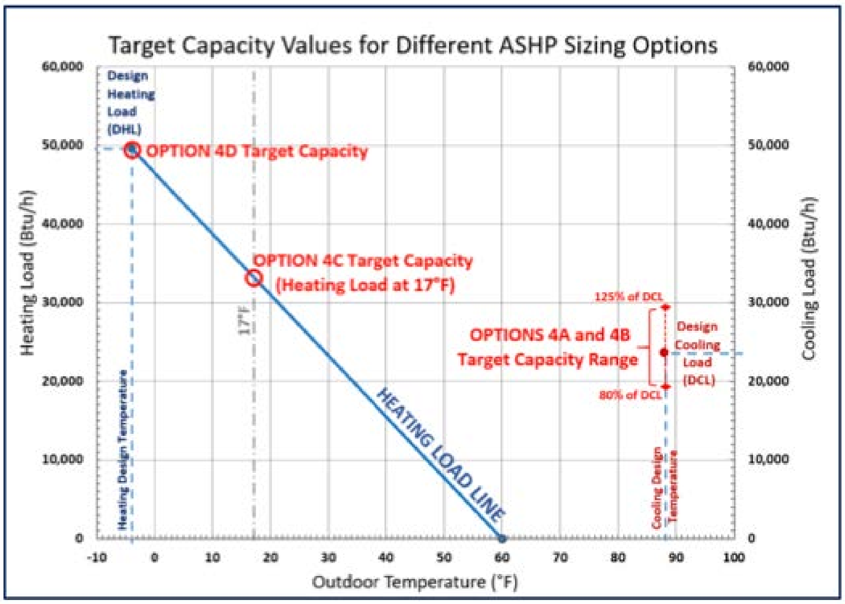

Figure 4. Target capacity estimation

Figure from [6]

Two options are provided for heat pump sizing in the measure, one based on percentage of the peak load and the other based on outdoor air temperature, with a gas or electric backup boiler system to address the remaining loads.

4.2.2.1 Sizing Based on Peak Load Percentage

In this approach, the target capacity is estimated as a percentage of the DHL. The corresponding outdoor air temperature (target OAT) that results in a heating demand equal to the target capacity is determined using the heating load line, as shown in Figure 5.

$$Target\ Capacity = Peak\ Percentage * Boiler\ Capacity$$

$$Target\ OAT = 16 - \frac{Target\ Capacity * (16 -\ HDT)}{DHL}$$

where 16°C (60°F) is the assumed outdoor air temperature to enable heating, HDT is the heating design temperature, and DHL is the design heating load (which is assumed to be equal to the heating capacity of the existing boiler in the model).

As shown in Figure 5, the heat pump's capacity drops as the outdoor air temperature decreases. Thus, the heat pump should be sized to provide the target capacity when operated at the target OAT. When the outdoor air temperature is lower than the target OAT, the backup heater will supplement the heat pump. Once the outdoor temperature is lower than the heat pump cutoff temperature, the heat pump will stop operating and the backup heater will be the only source of heating. For this reason, the backup heating must be sized to accommodate the full DHL. The shaded region in Figure 5 indicates the portion of the heating provided by the backup heater.

If the target OAT is lower than the cutoff temperature, the target capacity should be computed based on the cutoff temperature instead of the target OAT.

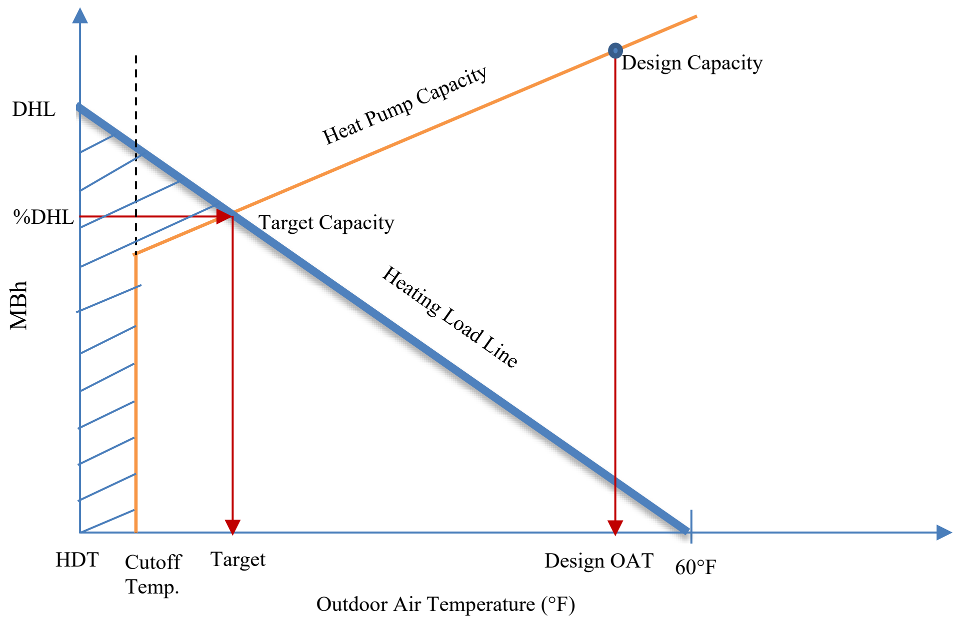

Figure 5. Heat pump sizing approach

4.2.2.2 Sizing Based on a Target Outdoor Air Temperature

In this approach, the heat pump is sized using the target capacity on the heat load line corresponding to the target outdoor air temperature provided by the user. The target capacity is calculated as:

$$Target\ Capacity = \frac{(16 - (Target\ OAT)) * DHL}{(16 - HDT)}$$

In both methods, if either the HDT or the cutoff temperature is greater than the target OAT, the sizing should be done according to the maximum of the two instead of the target OAT. For example, if the cutoff temperature of the heat pump is 0°F and the user picks a target OAT of 17°F, it doesn't make sense to size the heat pump corresponding to 17°F, as the heat pump won't be operating below 0°F.

4.2.3 Defrosting Consideration

ASHP boilers require occasional defrosting when operating at an outdoor air temperature below 47°F. Commercial ASHP boilers with multiple compressors reduce the impact of defrosting by limiting the defrosting to only one circuit at a time. The frequency of defrosting depends on the operating conditions. Table 6 summarizes the suggested capacity derate factors for sizing application [2].

Table 6. Heat Pump Defrost Capacity Derate Factors

Outdoor Air Temperature, °F	Capacity Derate Factor
>47	1 (no derate)
35–47	0.95 to 0.98
20–34	0.90 to 0.95
5–20	0.85 to 0.90
0–5	0.80 to 0.85

The target capacity estimated in Section 4.2.2 is updated using the "Capacity Derate Factor" to account for the impact of defrosting.

$$Updated\ Target\ Capcity = \frac{Target\ Capcity}{Derate\ Factor}$$

4.2.4 Estimation of Rated Heat Pump Capacity

Most heat pump manufacturers provide the heat pump rated capacity at a specific condition, usually at an outdoor air temperature of 47°F. The updated target capacity estimated in Section 4.2.3 needs to be converted to the required capacity at the design condition. To estimate the required rated capacity of the heat pump at the design outdoor air temperature, we used a performance curve called $CapFT$ [9] that captures the variation of a heat pump's capacity with outdoor air temperature and hot water set point. Heat pump performance curves are discussed in depth in Section 4.3 and Appendix A. The target capacity at the design outdoor air temperature (Target Capacity @ Design OAT) is estimated as:

$$Target\ Capacity\ @\ Design\ OAT = \frac{Updated\ Target\ Capcity}{CapFT\ @\ Target\ OAT}$$

$$CapFT@Target\ OAT = a + b * T_{cond_{out}} + c * T_{cond_{out}}^2 + d * Tareget\ OAT + e * Target\ OAT^2 + f * T_{cond_{out}} * Target\ OAT$$

where a, b, c, d, e, and f are $CapFT$ performance curve coefficients and $Tcond_{out}$ is the hot water temperature at the condenser outlet of the heat pump (which is equivalent to the hot water heating set point).

4.2.5 Use of Multiple Heat Pumps

In practice, it is more common to have multiple heat pumps of medium size than to have one big heat pump in a building. It creates redundancy and increases the energy efficiency of the system by providing the flexibility to run a few heat pumps at a higher part load ratio and with less cycling than running a big chiller. This measure allows users to provide the rated capacity of the heat pump they would like to use. The default heating capacity per unit used in the measure is 40 kW (136.5 MBH) and is based on Mitsubishi's Ecodan ASHP [10]. The measure adds multiple heat pumps in parallel when the estimated rated capacity is greater than the assumed rated capacity per unit. The number of ASHPs is estimated as:

$$Number\ of\ ASHPs = Roundup(\frac{Estimated\ Rated\ Capacity}{Rated\ Capcity\ per\ Unit})$$

where “Roundup” is used to convert a fraction to the closest higher integer value.

If the estimated rated capacity is lower than the assumed rated capacity per unit, only one heat pump with the estimated rated capacity will be used in the model.

4.2.6 Heat Pump Set Point

Heat pumps have a lower hot water temperature output than boilers. There are some CO_2 refrigerant heat pump water heaters that supply hot water up to 180°F [11], but most of the commercially available heat pumps have a hot water supply temperature capped at around 140°F. The hot water supply temperature they generate also depends on the outdoor air temperature. Figure 6 and Figure 7 show operation maps of heat pumps by Trane and Mitsubishi. For the Trane unit, the hot water leaving temperature drops from 140°F at an outdoor air temperature of 70°F to 100°F at an outdoor air temperature of 0°F. The Mitsubishi unit maintains a hot water set point even at colder temperatures; it supplies 158°F hot water at an outdoor air temperature as low as −4°F and drops to 150°F at −13°F.

Taking the performance of the Mitsubishi unit into consideration, the measure assumed the heat pump can provide the requested hot water supply temperature all the way to the cutoff temperature.

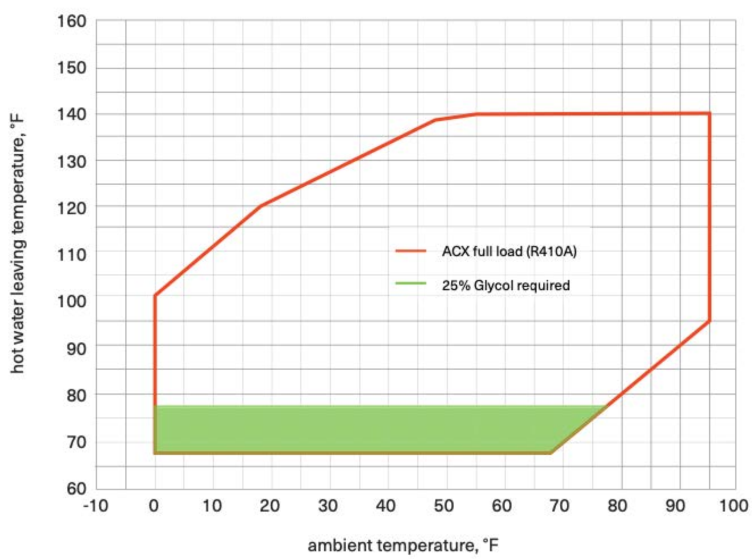

Figure 6. Operating map for ACX heat pump by Trane

Figure from [2]

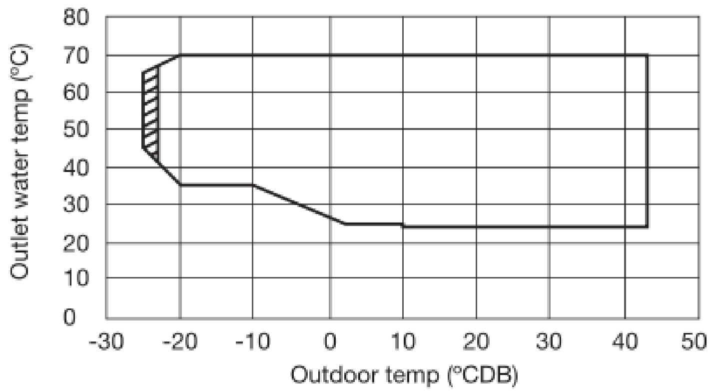

Figure 7. Operation map for Mitsubishi Ecodan ASHP

Figure from [10]

From an energy use perspective, if the existing coil sizes are big enough, using a lower hot water set point is recommended. Figure 8 shows ASHRAE's recommended minimum COPs for different hot water set points [12].

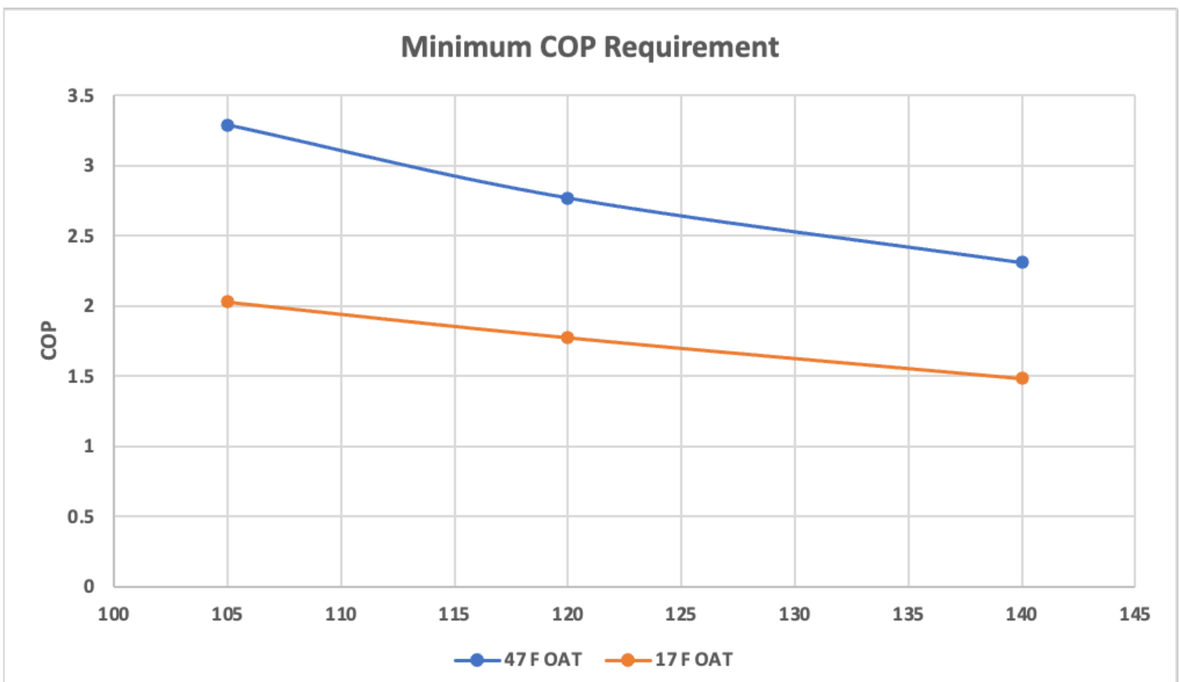

Figure 8. American National Standards Institute (ANSI)/ASHRAE/Illuminating Engineering Society (IES) 90.1-2021 minimum heating COPs

We have provided two options in the measure for assigning a hot water set point: original set point or new set point.

- Original set point: The existing hot water loop set point will be used as a set point for the heat pump and the hot water loop.

- New set point: The user-assigned new set point value will be used for the heat pump and hot water loop.

When the set point is changed, it will have an impact on the downstream coils connected to the heat pump. More flow needs to be supplied by the pump to handle the same heating load with a lower set point. To accommodate this change, the measure provides an option to auto-size the heating coil. If users choose to auto-size, the measure will auto-size the overall heat transfer coefficient and maximum water flow rate of the values of the coil.

4.3 Modeling ASHP Boilers in OpenStudio

During the measure development, we considered two heat pump models in OpenStudio; pumped condenser and plant loop EIR heat pump. The pumped condenser heat pump model is a combination of multiple objects, including a fan, a water tank, and an air-to-water heat pump coil object that is composed of a heat pump and a water circulation pump between the heat pump and the water tank [9]. This model doesn't allow a cutoff temperature below 23°F and this is way higher than the practical cutoff temperature by commercial heat pump boilers which could go as low as -25oF, especially the CO_2-based heat pumps [1]. Because of this limitation, this model is not used in this version of the measure but could be considered in the future once the cutoff temperature limit is relaxed to a lower value.

The second option considered—and the model selected for this measure—is a plant loop heat pump energy efficiency ratio (EIR) heating model. The plant loop EIR heating heat pump object is a recently added object for modeling a heat pump. Unlike the pumped condenser heat pump, this object doesn't have a water tank that helps differentiate the heat pump from the main hot water loop. In addition, the current version is a constant flow model that requests full design flow from the plant [9]. Because of this limitation, this object could not be directly added to a hot water loop with a variable speed pump. To circumvent this, and to provide the necessary separation between the hot water loop and the heat pump, we added a heat pump loop (as shown in Figure 9) to the existing building model. The heat pump loop has a heat pump on the supply side and a fluid-to-fluid heat exchanger on the demand side. The same heat exchanger is connected in series to the existing boiler. As indicated in Figure 9, the heat exchanger is added before the boiler so that it will be the primary heating source while the boiler handles the rest. For multiple heat pumps in the heat pump loop, we used a "sequentialLoad" control scheme, in which heat pumps are fired sequentially until the heating load is met. To avoid system inefficiency due to the addition of a heat exchanger, we used an "ideal" heat exchanger (i.e., the effectiveness of the heat exchanger was assumed to be 1). We also used an "UncontrolledOn" control scheme for the heat exchanger, which allows the heat exchanger to run whenever there is a nonzero flow in the main hot water loop.

Another limitation of this object is that it doesn't have a cutoff temperature. We used "AvailabilityManagerLowTemperatureTurnOff" in the heat pump loop, which allows the loop to be disabled when the outdoor air temperature is below the cutoff temperature.

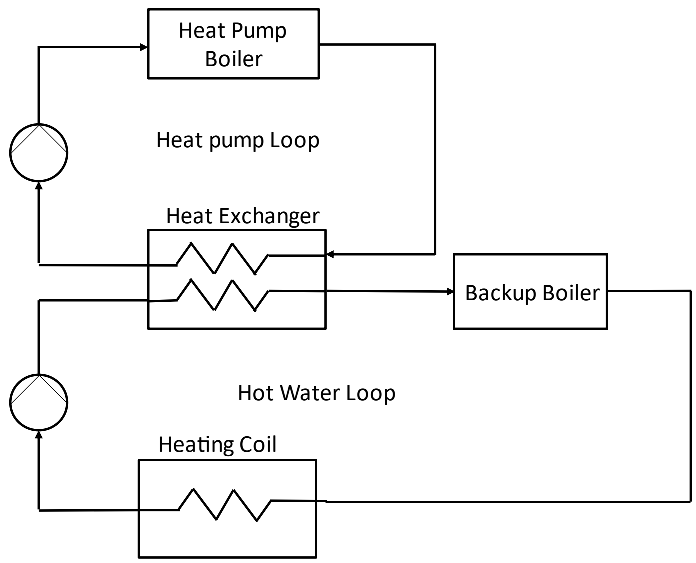

Figure 9. Configuration of heat pump and hot water loops

In this configuration, the heat pump will first attempt to lift the water temperature to the requested set point, followed by a backup heating element in the tank to address any remaining load not met by the heat pump.

The plant loop heat pump EIR heating model uses three performance curves—CapFTemp, EIRFTemp, and EIRPLR—to capture the impact of operating conditions on capacity and performance.

CapFTemp modifies the capacity of the heat pump based on the outdoor air and heat pump condenser outlet temperatures:

$$CapFTemp = a_1 + b_1 (T_{cond,out}) + c_1 (T_{cond,out})^2 + d_1 (T_{air,in}) + e_1 (T_{air,in})^2 + f_1 (T_{air,in}) (T_{cond,out})$$

EIRFTemp modifies the EIR, which is the inverse of the coefficient of performance (COP), of the heat pump based on outdoor and heat pump condenser outlet temperatures:

$$EIRFTemp = a_2 + b_2 (T_{cond,out}) + c_2 (T_{cond,out})^2 + d_2 (T_{air,in}) + e_2(T_{air,in})^2 + f_2 (T_{air,in}) (T_{cond,out})$$

EIRPLR modifies the EIR of the heat pump based on the part load ratio (PLR) and captures efficiency loss from compressor cycling:

$$EIRPLR = a_3 + b_3 PLR + c_3 PLR^2$$

We used data provided by Colmac [13] to generate the CapFTemp and EIRFTemp performance curves. During the measure development, we were not able to find performance data for EIRPLR. Thus, we assumed a linear variation between EIR and PLR that resulted in a 0% reduction in EIR at 1 PLR and a 25% reduction in EIR for a PLR close to zero. More detail about the performance curves is given in Appendix A.

Figure 10 and Figure 11 show how the CAPFT and EIRFT curve output values change with outdoor air temperature and hot water leaving temperature. As shown in Figure 10, the CAPFT value increases as the outdoor air temperature and condenser leaving water temperature increase. The EIRFT curve shown in Figure 11 shows a decrease in EIR (improvement in COP) as the outdoor temperature increases and the condenser leaving water temperature decreases.

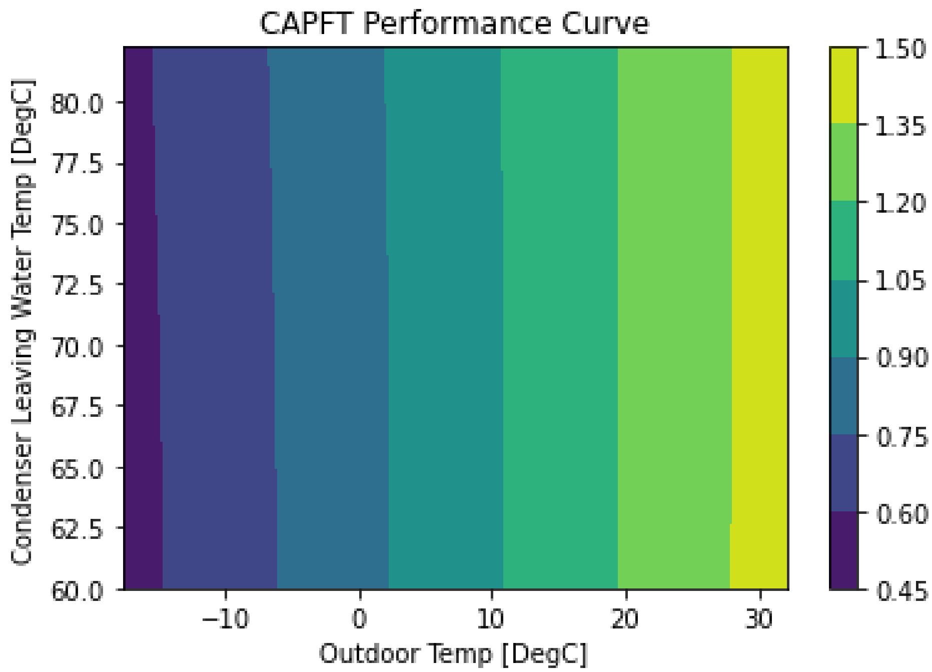

Figure 10. CAPFT performance curve output

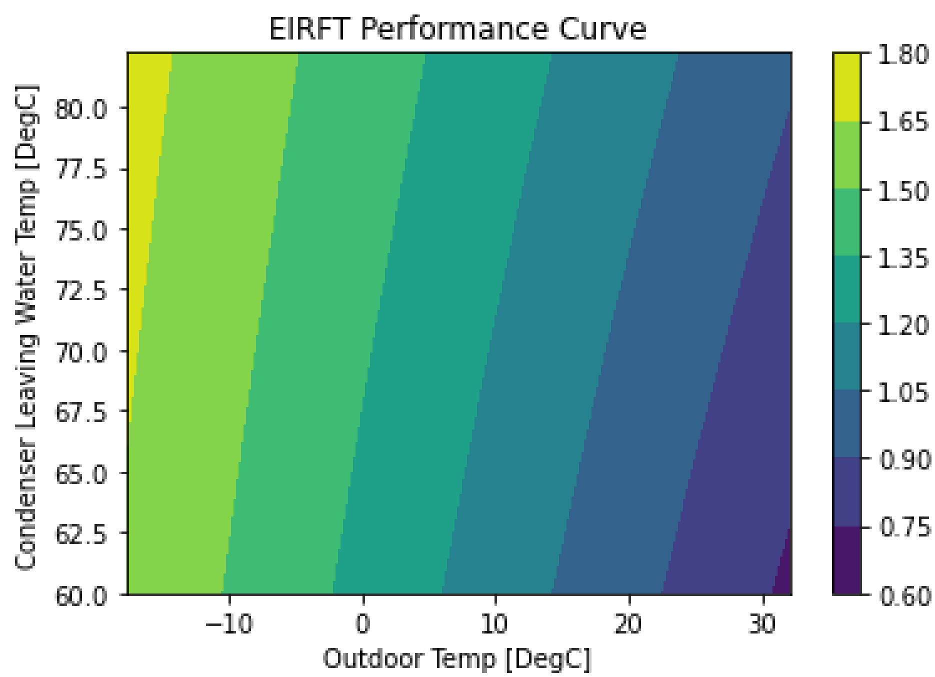

Figure 11. EIRFT performance curve output

4.3.1 Heat Pump Backup System

The measure provides two options for a backup: keep the existing boiler or use a new electric resistance element. If users choose to use a new electric resistance element, the measure uses the existing boiler as a backup heater after updating the fuel type to electricity and the thermal efficiency to 100%.

4.4 Limitations and Concerns

As indicated in Section 3, some of the boilers in the baseline ComStock models don't have minimum load turndown control. This allows small flow with insignificant heating in the hot water loop. This had a negative impact during the application of this measure, as the measure introduces a heat pump loop that is triggered by a nonzero flow in the hot water loop. The small flow in the hot water loop forces the heat pump to cycle frequently and eventually affects the expected savings from the application of this measure. This issue should be addressed in the next version of the ComStock models.

The heat pump object used in this measure—the plant loop EIR heating heat pump—is a constant flow model that requires full design flow from the plant. This model assumption imposes limitations on modeling variable speed heat pumps and introduces frequent cycling of the heat pump, leading to inefficiencies to the system. This measure could be updated in the future once the updated version of the heat pump object with a variable speed option is available.

5 Output Variables

Table 7 includes a list of output variables that are calculated in ComStock. These variables are important in terms of understanding the differences between buildings with and without the replace_boiler_by_heatpump measure applied. These output variables can also be used for understanding the economics of the upgrade (e.g., return on investment) if cost information (i.e., material, labor, and maintenance costs for technology implementation) is available.

Table 7. Output Variables Calculated From the Measure Application

Variable Name	Description
Heat pump capacity weighted design COP	COP of the heat pump at the rated design conditions
Heat pump average COP	Average heat pump COP
Heat pump total load	Total heating provided by heat pump
Boiler total load	Total heating provided by boiler
Heat pump total electricity	Total electricity consumption by heat pump
Boiler total electricity	Total electricity consumption by boiler
Heat pump capacity kbtuh	Heat pump capacity
Count heat pumps	Count of heat pumps
Count heat pumps 0-300 kbtuh	Count of heat pumps in the range of 0–300 kbtuh capacity
Count heat pumps 300-2500 kbtuh	Count of heat pumps in the range of 300–2500 kbtuh capacity
Count heat pumps 2500 plus kbtuh	Count of heat pumps with more than 2500 kbtuh capacity
Hot water loop total load	Total heating load in the hot water loop
Hot water loop boiler fraction	Fraction of heating load provided by boiler
Hot water loop heat pump fraction	Fraction of heating load provided by heat pump

6 Results

6.1 Single Building Model Example

Table 8 shows an end-use energy consumption comparison for a 75,000-square-foot hospital building model in Gallatin Field, MT before and after application of the default measure inputs. The two categories that are significantly affected by this measure are the heating and pump energy end uses. The electricity consumption for heating is 2.3 times lower than the natural gas consumption in the baseline. This is equivalent to an overall COP of 2.3 by the heat pump assuming a 100% efficiency boiler in the baseline model. The rated COP of the heat pump used is 2.85, and the observed reduction in the overall COP is due to a lower operating outdoor air temperature than the rated outdoor air temperature of 47°F and the use of electric resistance heater during outdoor air temperatures below the heat pump boiler cutoff temperature. The increment in the pump energy consumption is due to the addition of a constant speed circulation pump in the heat pump loop.

Table 8. End-Use Energy Consumption Comparison

End Use	Baseline		Updated	
	Electricity [GJ]	**Natural Gas [GJ]**	**Electricity [GJ]**	**Natural Gas [GJ]**
Heating	0	6864.46	2976.96	0
Cooling	681.2	0	681.25	0
Interior Lighting	872.41	0	872.41	0
Exterior Lighting	256.9	0	256.9	0
Interior Equipment	1860.28	214.09	1860.28	214.09
Exterior Equipment	0	0	0	0
Fans	1188.21	0	1188.54	0
Pumps	155.45	0	173.07	0
Heat Rejection	0	0	0	0
Humidification	0	0	0	0
Heat Recovery	0	0	0	0
Water Systems	285.01	405.9	285.02	405.9
Refrigeration	126.52	0	126.5	0
Generators	0	0	0	0
Total End Uses	**5425.99**	**7484.45**	**8420.95**	**619.99**

We checked the sequencing between the heat pump and the boiler during low-temperature operation using an example cutoff temperature of 25°F (−4°C). Figure 12 shows the operation pattern. As expected, the heat pump handled most of the load at a higher temperature. The boiler starts supplementing the ASHP as the temperatures drops and takes over all the heating when the temperature is below the cutoff temperature.

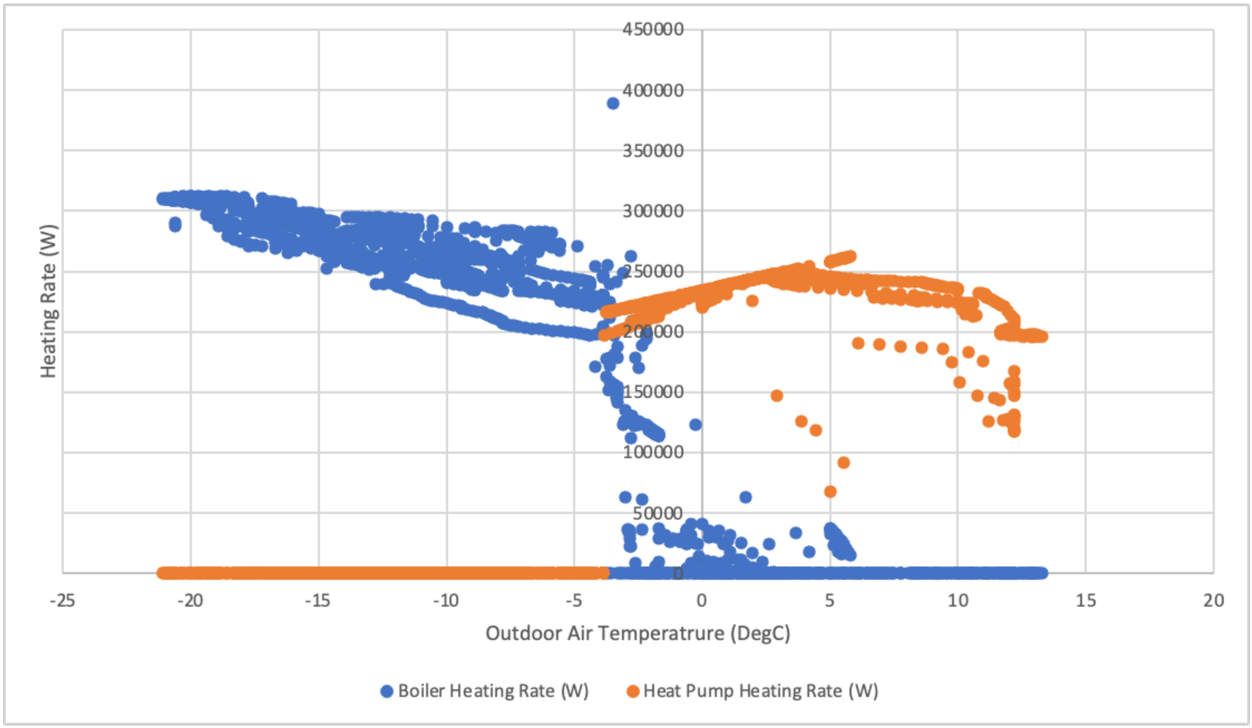

Figure 12. Sequencing between ASHP and boiler

6.2 Stock Energy Impact

This measure is applicable for a natural gas boiler heating type, which accounts for 33% of the floor area of the U.S. commercial building stock modeled in ComStock (Figure 3). The following key observations are made from application of this measure.

- 7.5 % total site energy savings (348TBtu)
- 42% total natural gas energy savings (521.8TBtu)
- 5.6% increase in total electricity (172.1TBtu)
- 63.1% heating natural gas savings
- 87.1% increase in heating electricity.

As indicated in Figure 13, the measure has negligible impacts in most of the end-use categories except heating electricity, heating natural gas, and pump electricity. The remaining gas heating in applicable buildings comes from a small subset of non-applicable systems (e.g., boilers that serve condenser water loops).

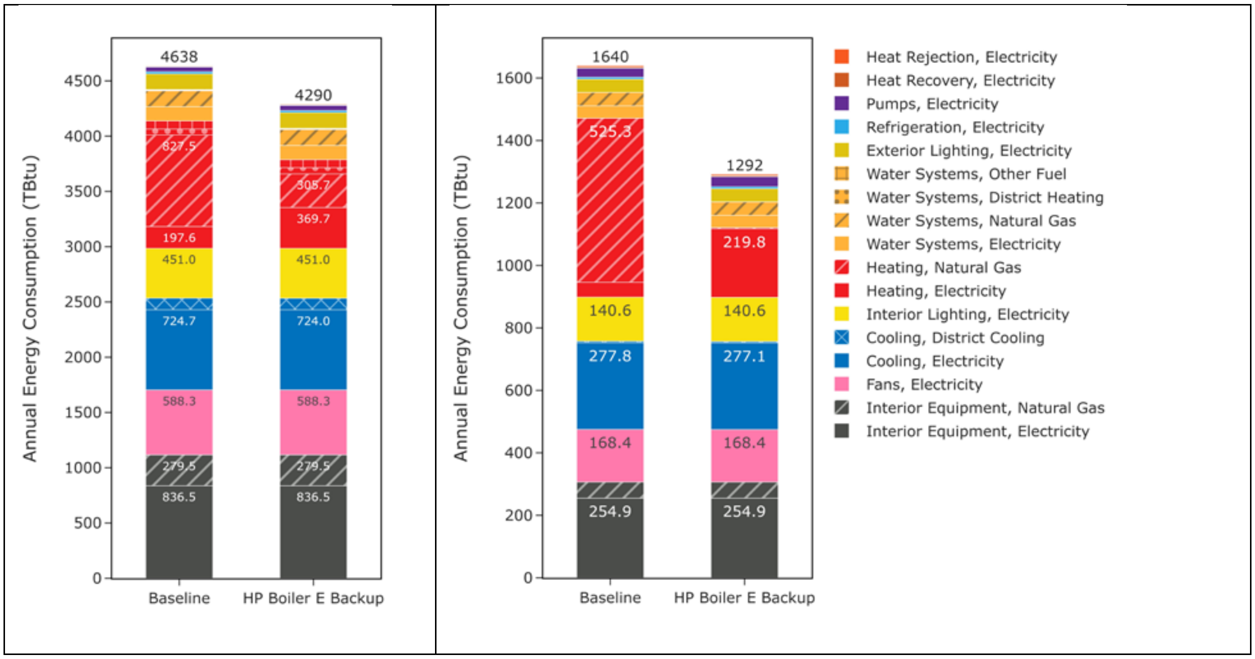

Figure 13. End-use energy consumption comparison for (a) all ComStock models and (b) for applicable buildings only

6.3 Stock Peak Impacts

Figure 14 shows the impact of the upgrade on the average daily maximum peak demand per square foot across the ComStock building models. There was no increase in summer, while 0.6% and 5.8 % increases were observed during the shoulder and winter seasons. This is expected, as the heat pump boiler mainly operates during the winter and sometimes during shoulder season. Heat pump peak demand is dependent on many factors including the climate, cutoff temperature, heat pump capacity, heat pump type (constant speed, variable speed), back up heating fuel type and capacity, unit efficiency and operating characteristics (e.g., thermostat schedule).

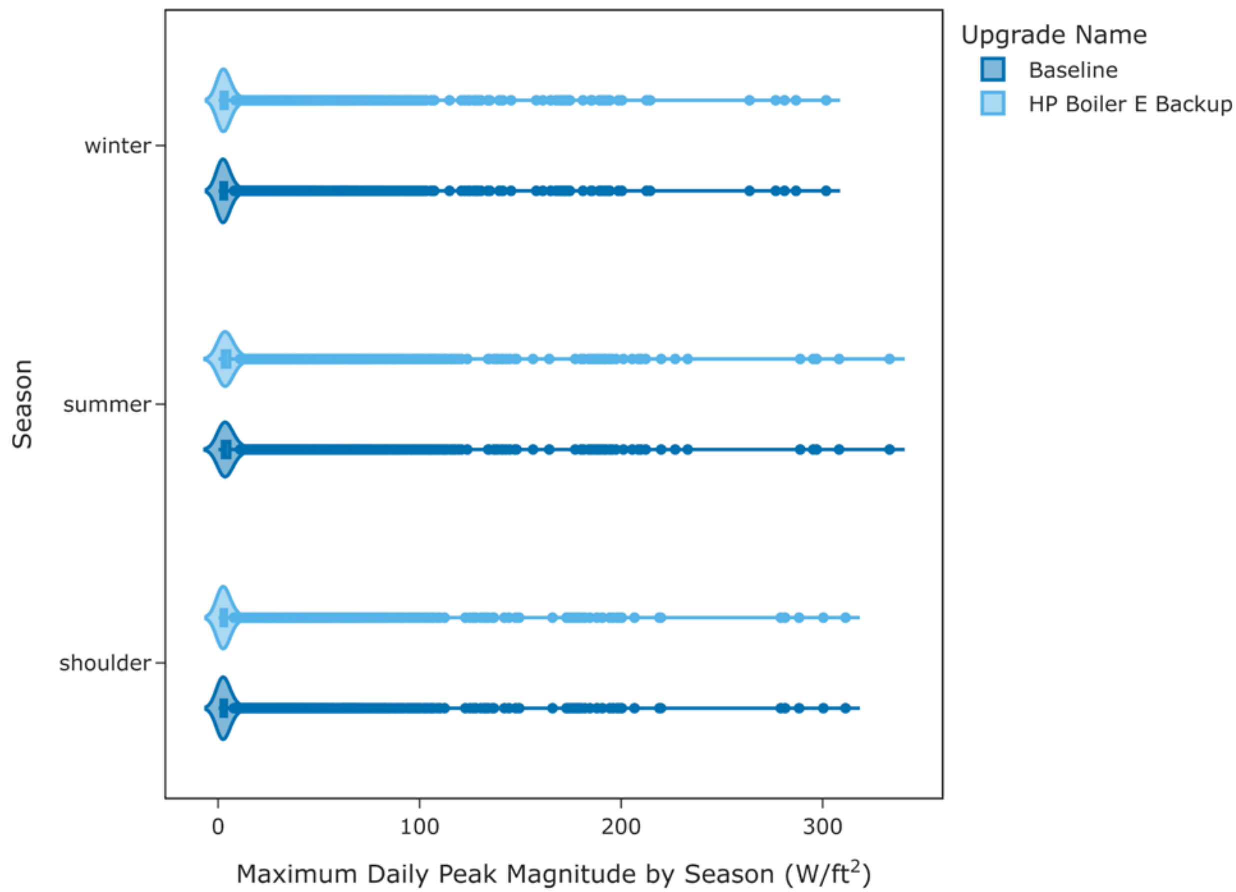

Figure 14. Average daily maximum peak demand comparison

6.4 Measure Impact by HVAC System Type

Figure 15 shows the percent site energy savings by different types of HVAC system. DOAS with water-source heat pump cooling tower with boiler and PVAV with gas heat with electric reheat exhibited the least savings. The primary reason for the limited savings observed in the DOAS with water-source heat pump cooling tower with boiler system is that heat pumps predominantly fulfill the heating load while the boiler solely provides heating for the DOAS units. In the case of the PVAV with gas heat with electric reheat system, the reduced savings are attributed to the presence of electric reheat coils in the VAVs which handle majority of the heating. On the other hand, the VAV systems with boiler reheat demonstrate higher savings. This is mainly due to the relatively higher floor areas they serve, makes them well suited for application of this measure.

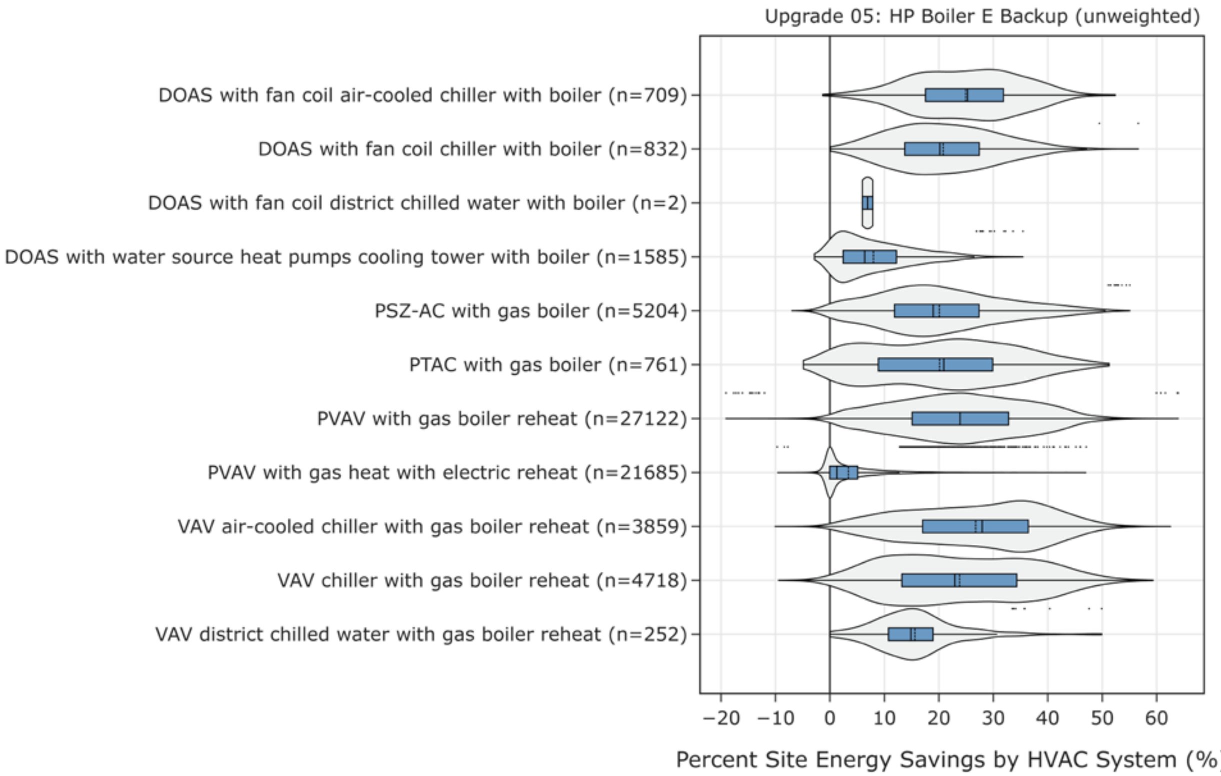

Figure 15. Percent savings by HVAC system type

In each category, a few buildings exhibited negative savings due to their very small heating loads. In the baseline case, these loads were managed by modulating the boiler. However, in the updated case, the small nonzero flow in the hot water loop triggered the heat exchanger to request flow in the heat pump loop. However, the heat pump model used only supports constant flow, leading to frequent cycling of the heat pumps, as depicted in Figure 16. Consequently, this resulted in higher energy usage compared to the baseline. Only a very small fraction of the total buildings demonstrated this behavior, rendering its impact on the overall result to be minimal, and the absolute energy impact was small.

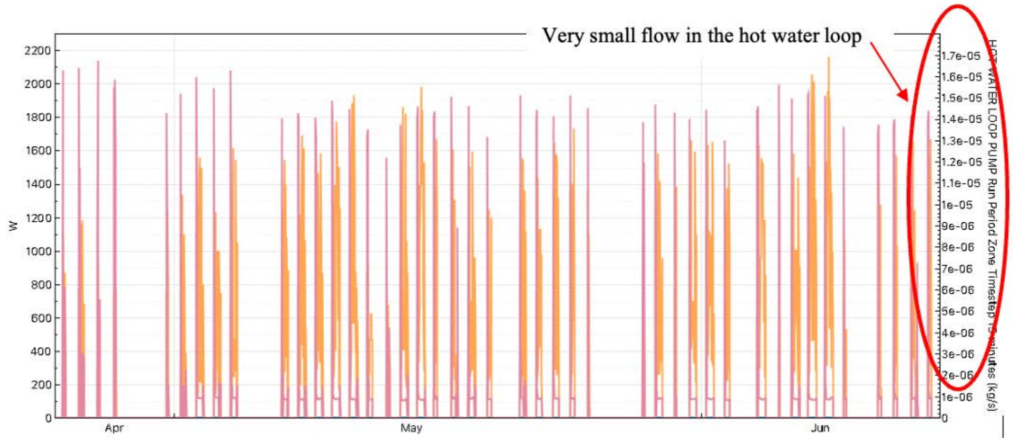

Figure 16. Flow in the hot water loop that triggers heat pump operation

6.5 Measure Impact by End Use

The impact of the measure on end-use energy consumption is indicated in Figure 17. As expected, the end uses that are affected most are natural gas heating, electricity pumps and electricity heating. As expected, nearly 100% of the natural gas used for heating is replaced by electricity. The increase in electricity consumption due to addition of a heat pump boiler with a backup electric boiler is illustrated by a negative percent savings in electricity heating and electricity pumps category. The increase in electricity pump energy use is due to the addition of a circulation pump in the heat pump loop. Very few buildings exhibited an unexpected reduction in pump electricity use. This is for buildings with significantly low heating load in the hot water loop that is prohibiting the heat exchanger in the loop from activating. This causes both the hot water loop and heat pump loop pumps not to run and resulted in pump energy savings. In these buildings, the electric reheat coils are operating in the absence of a heat source from the hot water loop to maintain the required zone temperature and no increase in unmet hours was observed. The minor changes in the other end uses are due to the change in the hot water set point from 180°F for the baseline to 140°F.

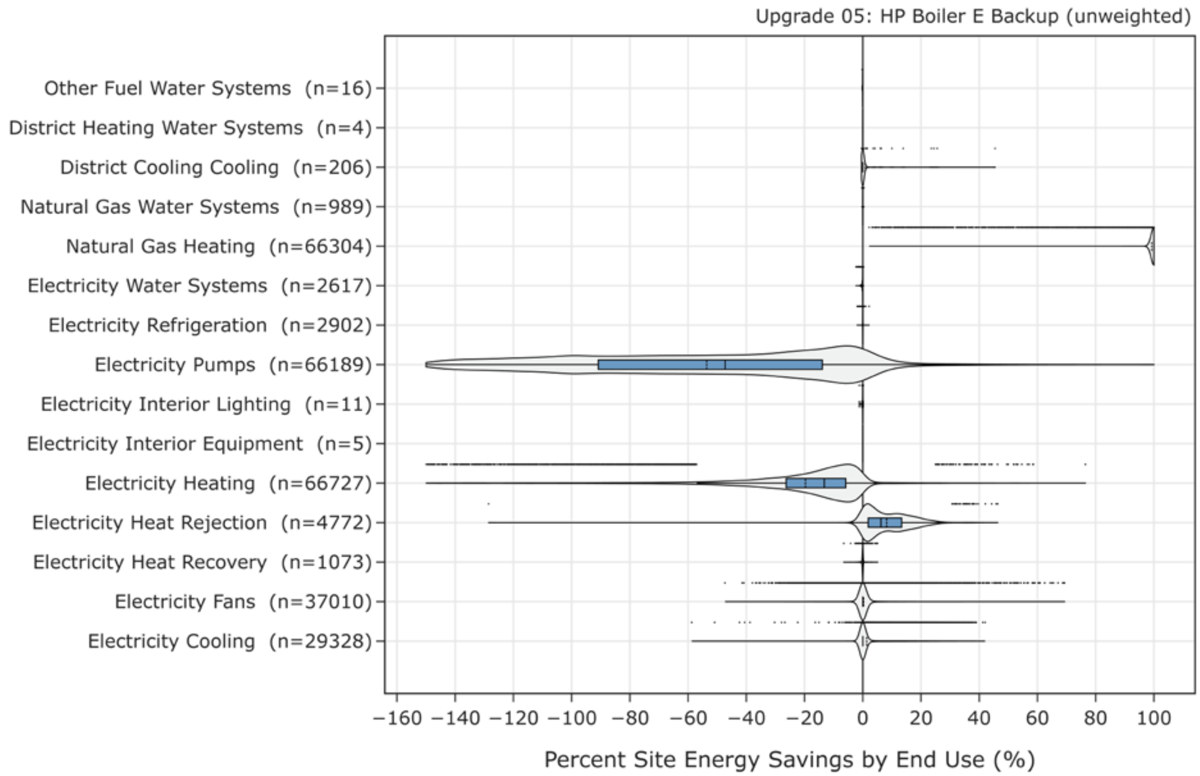

Figure 17. Percent savings by end use

6.6 Measure Impact by Building Type

Figure 18 shows the impact of the measure by building type. Hospitals, primary schools, and secondary schools are the three building types with the most site energy savings, whereas quick-service restaurants exhibit the least. The higher savings in these three buildings is attributed to their HVAC system type as well as their relatively higher heating loads. Only ~1% of the ComStock quick-service restaurant buildings use boiler and the rest use furnace, electric resistance and ASHPs, which resulted in lower savings from application of this measure. In contrast, 63% of hospitals, primary schools and secondary schools use boiler which attributed to the observed higher site energy savings.

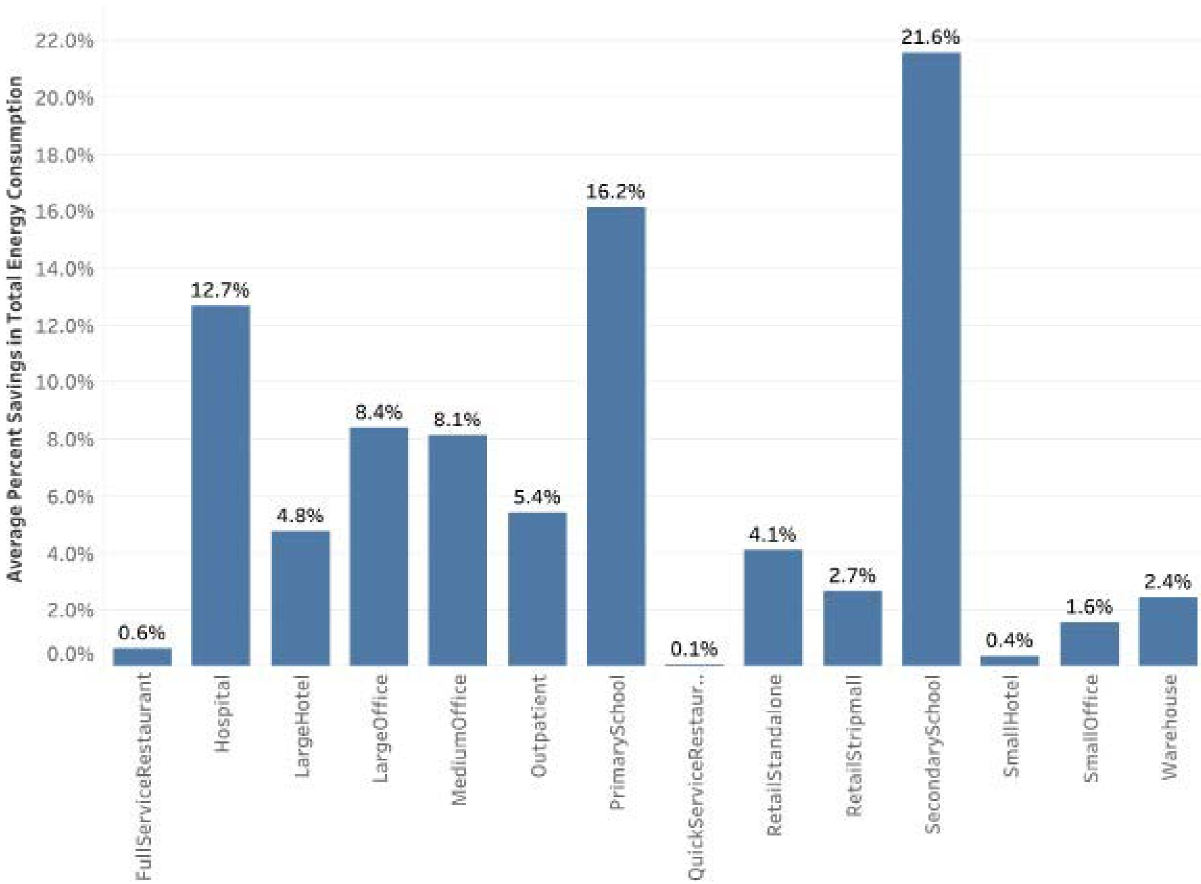

Figure 18. Site energy savings by building type

6.7 Measure Impact by State

Figure 18 and Figure 19 show a state-by-state percentage natural gas and electricity savings distributions across the entire building stock. As expected, cold regions exhibited higher savings in natural gas consumption and a higher increase in electricity energy use. The state level average increase in electricity consumptions is indicated by negative savings. Please note that the savings are also influenced by buildings type distribution and total floor area across the states. Besides the need for more electricity by the heat pump boilers in the colder regions, the high electricity demand is also in part due to the lower efficiency of the heat pump boilers in these regions. The unexpected electricity saving observed in Florida is due to cooling energy savings in some buildings due to the model autosizing limitation discussed in Section 4.4. As indicated in Figure 13, no significant stock level cooling electricity energy consumption difference is observed due to the upgrade, confirming only very few buildings exhibited cooling energy savings and the impact is negligible. Figure 21 shows the COP variations across the states. In general, heat pump boilers operate at a higher efficiency in hot climates compared to cold ones.

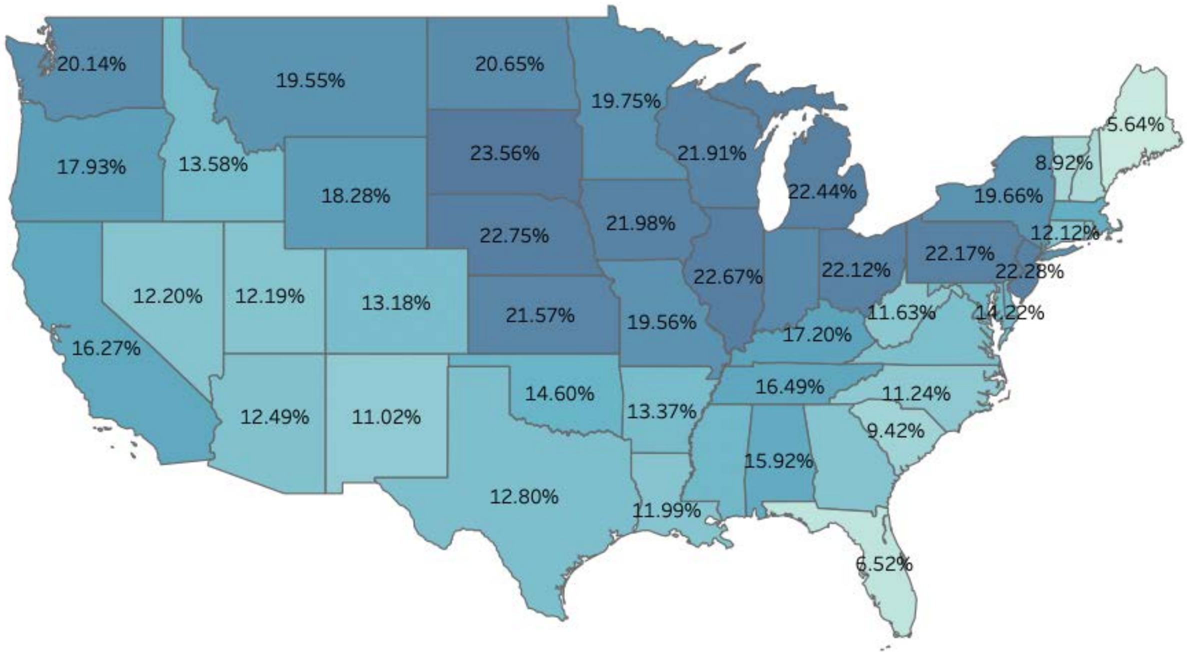

Figure 19. percentage savings in natural gas

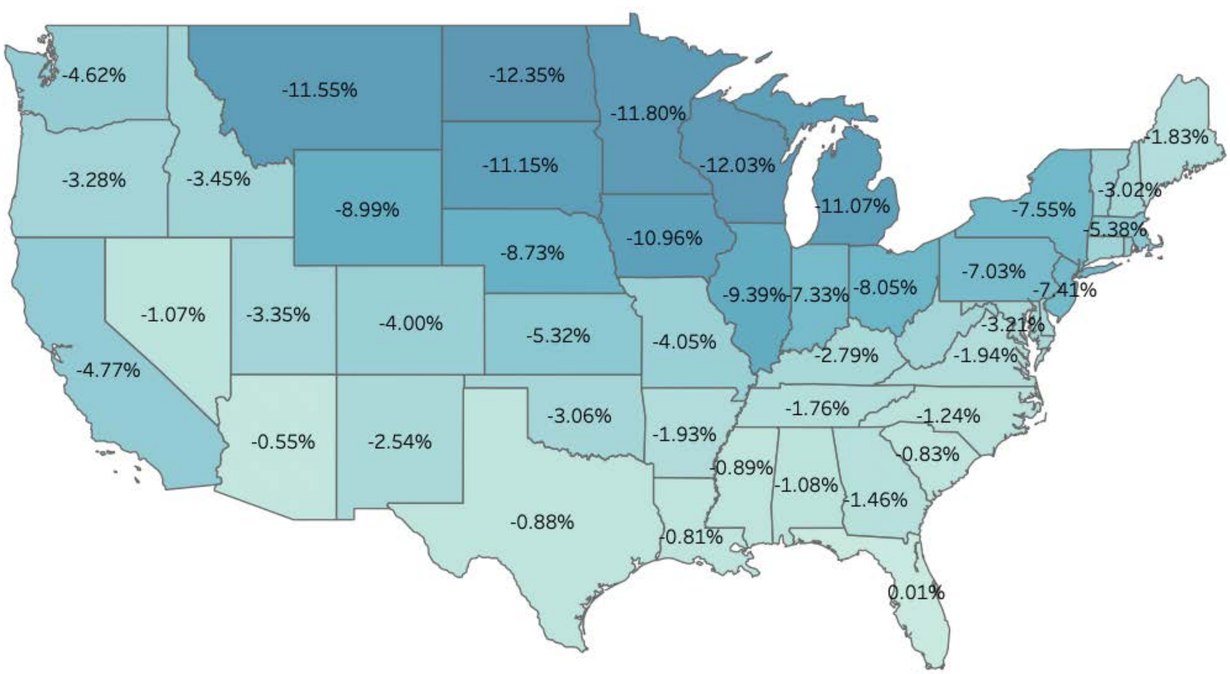

Figure 20. Percentage savings in electricity consumptions

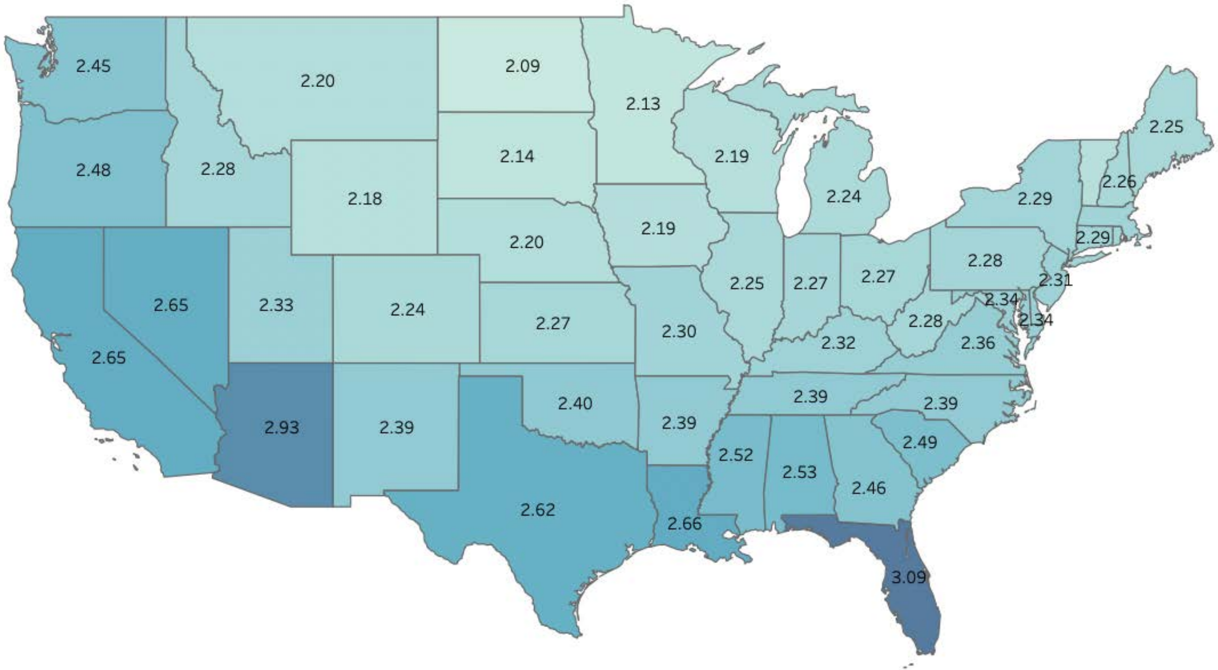

Figure 21. Heat pump average COP

6.8 Measure Impact on Greenhouse Gas Emissions

Figure 22 illustrates the annual greenhouse gas emissions comparison between the baseline and the upgrade. As expected, emissions from natural gas decreased while emission from electricity increased. The overall reduction in greenhouse gas emissions depends on the emission source. Three sources of electricity are considered for comparison: Cambium Long-Run Marginal Emissions Rate (LRMER) High Renewable Energy (RE) Cost 15-Year, Cambium LRMER Low RE Cost 15-Year, and emission and generation resource integrated database (eGRID). The percentage values in the figure indicate percentage increase or decrease in emissions compared to the baseline. All upgrade scenarios resulted in net savings in combined emissions from electricity and natural gas, with comparison using LRMER Low RE Cost scenario resulting the highest net savings of 22.8 MMT (9.3%).

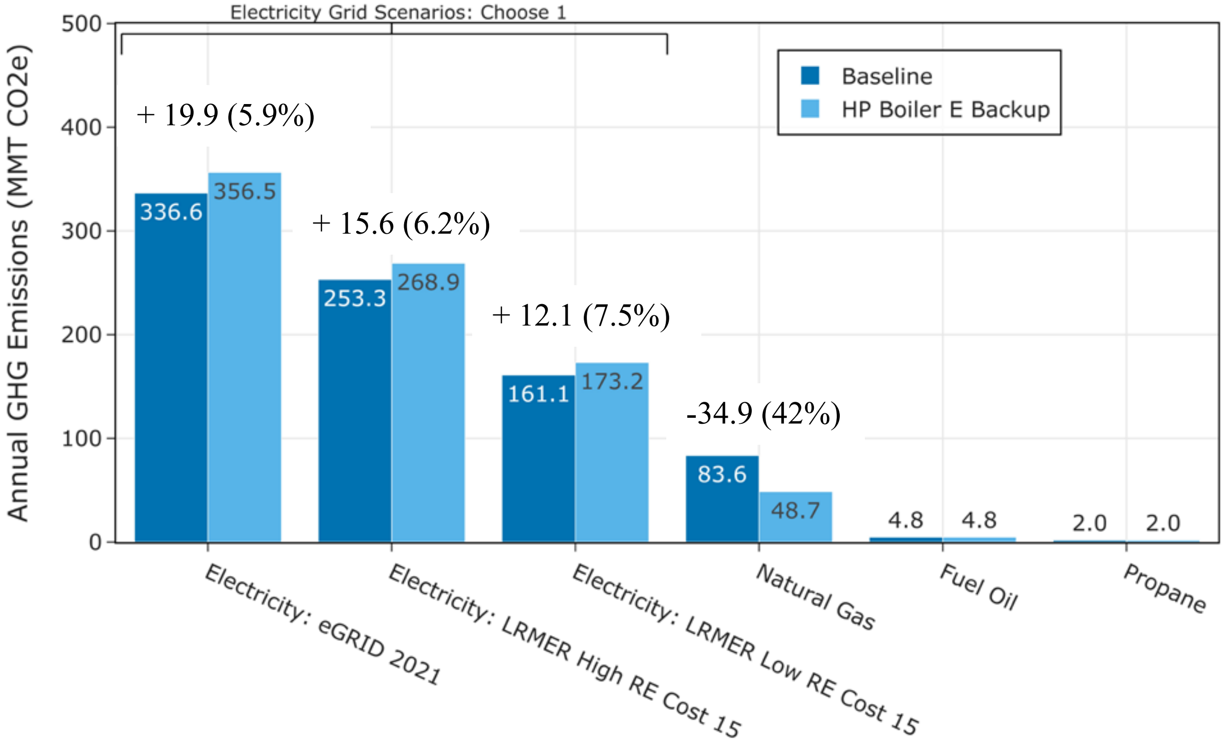

Figure 22. Annual greenhouse gas emission comparison

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Appendix A

A.1 Appendix ASHP Boiler Performance Curve Generation

As discussed in Section 4.3, the heat pump model used in this measure has three performance curves for capturing the dependency of the heat pump performance on the operating conditions. Two of the curves, capacity as a function of temperature (CapFTemp) and energy input ratio (EIR) as a function of temperature (EIRFTemp) capture the dependency of heat pump capacity and efficiency on outdoor air temperature and hot water supply temperature. EIR as a function of part load ratio (EIRPLR) captures the dependency of the heat pump efficiency on heat pump loading and cycling.

$$Q_{Available} = Q_{Reference} \text{ x } CapFTemp$$

$$P = P_{Reference} * EIRFTemp * EIRPLR$$

$$CapFTemp = a1 + b1\,(T_{cond,out}) + c1\,(T_{cond,out})^2 + d1\,(T_{air,in}) + e1\,(T_{air,in})^2 + f1\,(T_{air,in})\,(T_{cond,out})$$

$$EIRFTemp = a2 + b2\,(T_{cond,out}) + c2(T_{cond,out})^2 + d2\,(T_{air,\,in}) + e2(T_{air,in})^2 + f2\,(T_{air,in})\,(T_{cond,out})$$

$$EIRPLR = a3 + b3\ PLR + C3\ PLR^2$$

Where $Q_{Reference}$ is the design heating capacity of the heat pump, $P_{Reference}$ is the design power demand of the heat pump, $Q_{Available}$ is the adjusted heating capacity, P is the adjusted power demand, $T_{cond,out}$ is the condenser outlet water temperature, $T_{air,in}$ is the ambient air temperature, PLR is heat pump part load ratio, a1, b1, c1, d1, e1 … c3 are performance curve coefficients that need to be extracted from operational data.

The two temperature-dependent performance curves, CapFTemp and EIRFTemp, were generated using performance data provided by Colmac. It is important to note that the efficiency of the heat pump is influenced not only by the operating temperature conditions but also by factors such as the load on the heat pump and its cycling frequency. During the development of the measurement, we encountered a challenge in obtaining manufacturer data to account for this dependency. As a result, we assumed a linear variation between EIR and PLR. According to this assumption, there is no reduction in efficiency when the PLR is one (indicating full load), and a 25% reduction in efficiency when the PLR is close to zero (indicating low load).

In order to evaluate the accuracy of the Colmac performance data, a comparison was made with data from Trane and Mitsubishi. Unfortunately, detailed data for the two units from Trane and Mitsubishi was not available. However, the capacity drop with outdoor air temperature observed in the Colmac unit appeared to be consistent with the data from Trane and Mitsubishi.

The results of the comparison are summarized in Table A-1 and Table A-2. According to the data, the capacity reductions for the Trane and Colmac units were found to be 44% and 50%, respectively, as the temperature decreased from 47°F to 0°F.on the other hand, the Mitsubishi data indicates a 31% capacity reduction as the outdoor air temperature decreases from 45°F to

20°F, while the Colmac unit shows a slightly lower reduction of 25% within the same temperature range.

Table A-1. Capacity Reduction With OAT for Trane and Colmac Units

	Capacity at 50°F (Btu/hr)	Capacity at 0°F (Btu/hr)	% Capacity Reduction From 50°F to 0°F	COP
Colmac[14]	57,600	31500	**45%**	2.70 @ 50°F
Trane [2]			**50%**	2.70 @ 47°F

Table A-2. Capacity Reduction Comparison Between Mitsubishi and Colmac Units

	Capacity at 45°F (Btu/hr)	Capacity at 20°F (Btu/hr)	% Capacity Reduction From 50°F to 20°F	COP
Colmac[14]	54,750	42,200	**23%**	2.70 @ 50°F
Mitsubishi [10]	140,400	117,234	**25%**	2.85 @ 45°F

Table A-3 summarizes the performance curve coefficients that are estimated using the Colmac data. The curve outputs for different combination of condenser water leaving temperature and outdoor air temperature are indicated in Figure A- 1 and Figure A-2.

Table A-3. Performance Curve Coefficients

	CAPFT	EIRFT	EIRPLR
a	0.88302749	0.84177647	1.25
b	−0.0016513	0.00648504	−0.25
c	1.44E−05	−8.68E−06	0
d	0.01833385	−0.0273677	
e	3.6396E−05	0.00018754	
f	−2.04E−05	0.0001082	

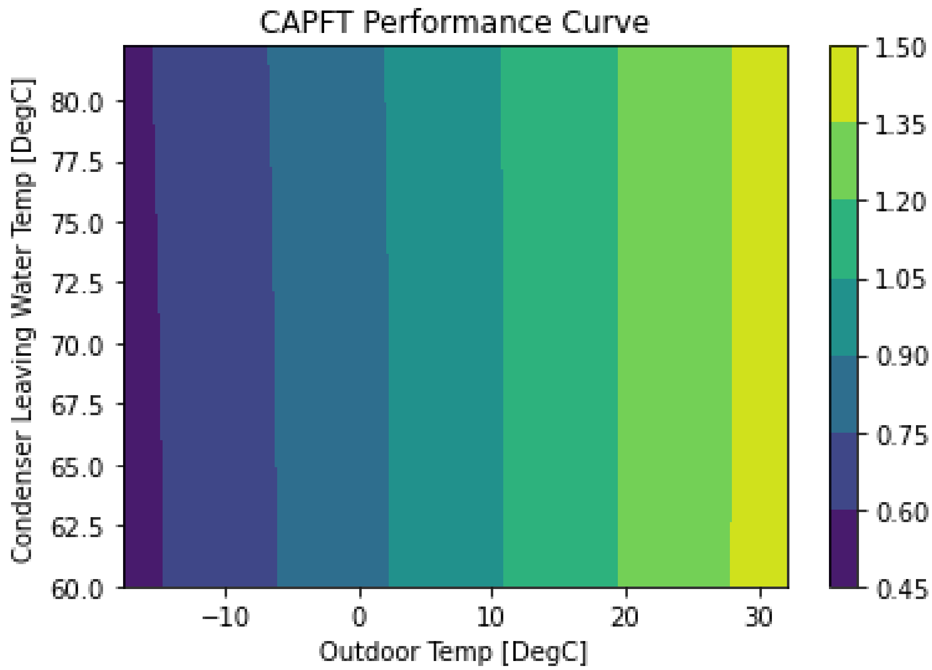

Figure A-1. CAPFT performance curve output

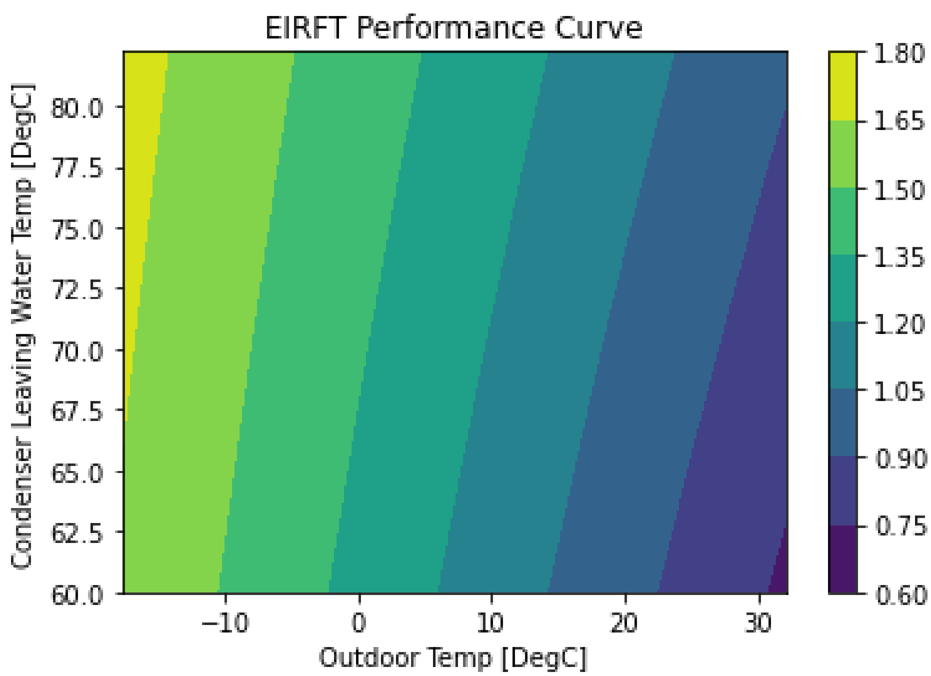

Figure A-2. EIRFT performance curve output

Figure A-3 illustrates the comparison between the actual heating capacity and EIR with the estimated values obtained using the performance curves for various combinations of outdoor air temperature and condenser water leaving temperature (CWLT). The figure demonstrates a satisfactory agreement between the two, providing assurance that we can confidently utilize the performance curves for modeling ASHP boilers.

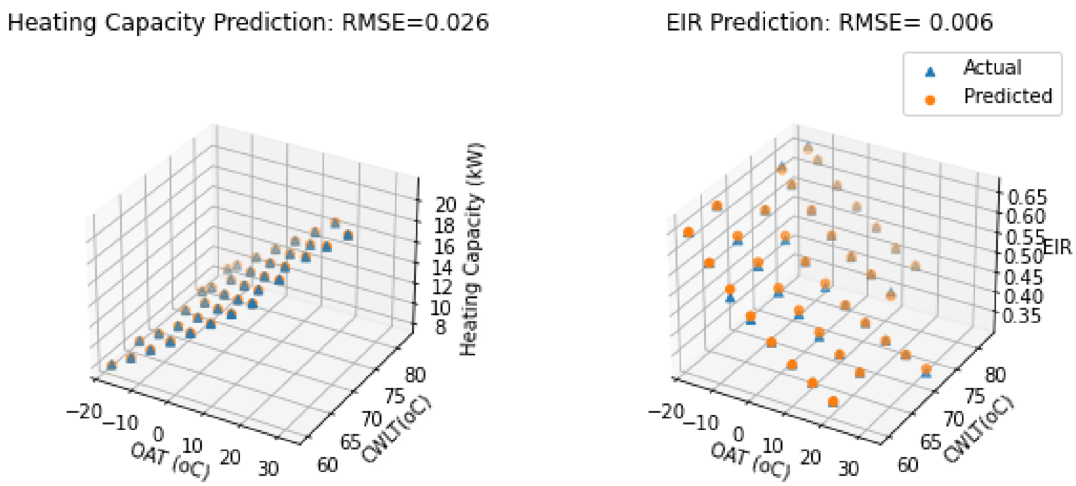

Figure A-3. Comparisons of predicted and actual capacity and EIR