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Impact Analysis of Transitioning to Heat Pump Rooftop Units for the U.S. Commercial Building Stock

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Abstract

Twenty percent (25%) of the energy consumed by the U.S. commercial building sector is from on-site combustion of fossil fuels for space heating. Part of decarbonizing U.S. energy systems to meet climate initiatives will require electrification of space heating equipment, often by transitioning to heat pumps. Rooftop units (RTU) are the most prominent commercial building HVAC system type and should therefore be prioritized for electrification solutions. However, there is limited understanding of the impact on emissions when considering regional electricity generation methods, as well as the impact of ambient temperature on capacity and efficiency, defrost operation, realistic sizing methodologies, and supplementary heating on overall heat pump performance. This study explores the effects of transitioning all installed, existing RTUs to high-performance heat pump RTUs for the U.S. commercial building stock. The analysis is performed using ComStock™, the U.S. Department of Energy's calibrated model of the U.S. commercial building stock. Results show 10% and 9% reductions in stock aggregate energy consumption and greenhouse gas emissions, respectively. This analysis will help inform the transition to heat pump RTUs for the U.S. commercial building stock.

Keywords: heat pump energy modeling; commercial building stock energy modeling; ComStock; heat pump rooftop unit modeling; commercial building electrification; commercial building HVAC modeling

1. Introduction

1.1. Decarbonizing the U.S. commercial building stock

Several simultaneous market transformations must occur to decarbonize the U.S. energy systems. These include renewable power supply and storage options, widespread adoption of energy efficiency and electric demand flexibility, as well as ending the burning of on-site fossil fuels such as natural gas, propane, fuel oil and others. Buildings burn fossil fuels on-site primarily for space heating, water heating, equipment loads and cooking. The combined on-site natural gas combustion of the residential and commercial sectors is estimated to contribute to 9% of all U.S. carbon dioxide (CO_2) emissions annually [1].

Natural gas is the primary space heating and water heating fuel source for half of all commercial buildings in the U.S. by number, representing almost 70% of commercial building floor space, with more than half of the natural gas consumed for space heating [2]. In addition to natural gas, almost 10% of commercial buildings report using fuel oils and propane for primary space heating purposes [2]. This leaves only 25% of all commercial buildings currently relying on electricity as their only source for space heating needs [2].

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1.2. Heat pump rooftop units as a decarbonization pathway

Many technologies are used to provide space heating in commercial building HVAC systems. Packaged rooftop units (RTUs) are currently used to heat 37% of commercial buildings in the U.S. (representing 50% of the total commercial floor space) [2]. Heat pumps currently provide space heating for only approximately 11% of commercial buildings (representing 15% of the total floor area) [2].

Heat pumps offer a high-performance electric option for commercial building space heating. Their use of electricity for heating enables pathways toward decarbonization with heat pumps delivering space heating 2-4 times more efficiently than electric resistance options. Based on the 2018 Commercial Buildings Energy Consumption Survey (CBECS) data, it is estimated that fewer than 15% of commercial buildings utilize heat pumps for space heating equipment, and when they are in use they are more commonly found in the warmer southern region of the U.S. [2].

Heat pump technologies are available on the market today to replace existing gas-fired or electric resistance RTU systems. Most manufacturers offer heat pump rooftop units (HP-RTU) with compressors capable of providing 105 kW or less of cooling capacity (30 tons). There is remarkable opportunity for growth and widespread adoption of this technology, and expansion of the field will have an extensive impact on electrification efforts.

1.3. ComStock energy modeling

ComStock™ is a physics-based model of the U.S. commercial building stock built from an ensemble of 350,000 OpenStudio® whole-building energy models using the EnergyPlus® simulation engine. As used in this analysis, ComStock represents 66% of the U.S. commercial building stock by floor area. Unlike analyses that use a small number of "typical" building energy models to represent the stock, ComStock includes a diverse range of building and HVAC system configurations, construction vintages, hours of operation, etc. The relative prevalence of these building characteristics is informed by a wide range of data sources, including commonly used datasets such as CBECS [3], proprietary data sources such as CoStar [4], and difficult-to-acquire data sources such as whole-building hourly electricity consumption data from a large sample of buildings. The diversity of building characteristics helps create more robust conclusions because the models reflect the diversity found in reality. ComStock has gone through extensive calibration and validation, particularly on the electricity consumption side, as documented in detail in [5].

For this paper, two of the most important ComStock inputs are the prevalence of HVAC system types and heating fuels. Both model inputs are derived from the CBECS 2012 microdata, in particular the HVAC-related questions [3]. Rather than assuming a uniform distribution of the prevalence of natural gas or electric heating across each census division, data from the residential American Communities Survey (ACS) [6] was used to supplement the CBECS data. The CBECS totals for each census division were maintained, while counties with more residential natural gas heating according to ACS were scaled to have a higher percentage of the commercial natural gas heating, and vice-versa. For a more detailed description of the ComStock model, including the data sources and assumptions referred to above, see the "ComStock Documentation" National Renewable Energy Laboratory (NREL) report expected to be published in March 2023.

2. Methodology

2.1. Analyzing measures with ComStock

ComStock uses a representative set of 350,000 unique OpenStudio energy models to represent the energy usage and behavior of the U.S. commercial building stock [7]. ComStock's bottom-up engineering model approach is especially well-suited for studying the impact of proposed changes to the existing building stock [8], [9]. For this workflow, the baseline set of ComStock OpenStudio models is modified to represent the proposed change to be studied. For example, if increasing the efficiency of cooling equipment in all commercial buildings by 10% was proposed, then the baseline (representing the building stock circa ~2018) set of OpenStudio energy models would be modified to have 10% higher cooling efficiency. The results of this modified run can then be compared back to the original ComStock baseline results to understand the various changes that occurred due to the modification.

In this study, the ComStock baseline is modified to represent replacing all RTUs in the existing building stock, either gas-fired or electric resistance, with high-performance HP-RTUs. This change affects HVAC equipment serving ~45% of the floor area in the ComStock baseline; the floor area representation of all HVAC system types in ComStock is shown in Fig. 1 [7]. High-performance HP-RTUs are defined to include the following features:

- Single-zone variable air volume (VAV) fan operation
- High-efficiency variable-speed compressor(s) for heating and cooling (>17 IEER)
- High-efficiency variable-speed fans
- Cold climate suitable with low temperature capabilities (minimum compressor operating temperature ≤ -17.8°C) and compressor defrost operation

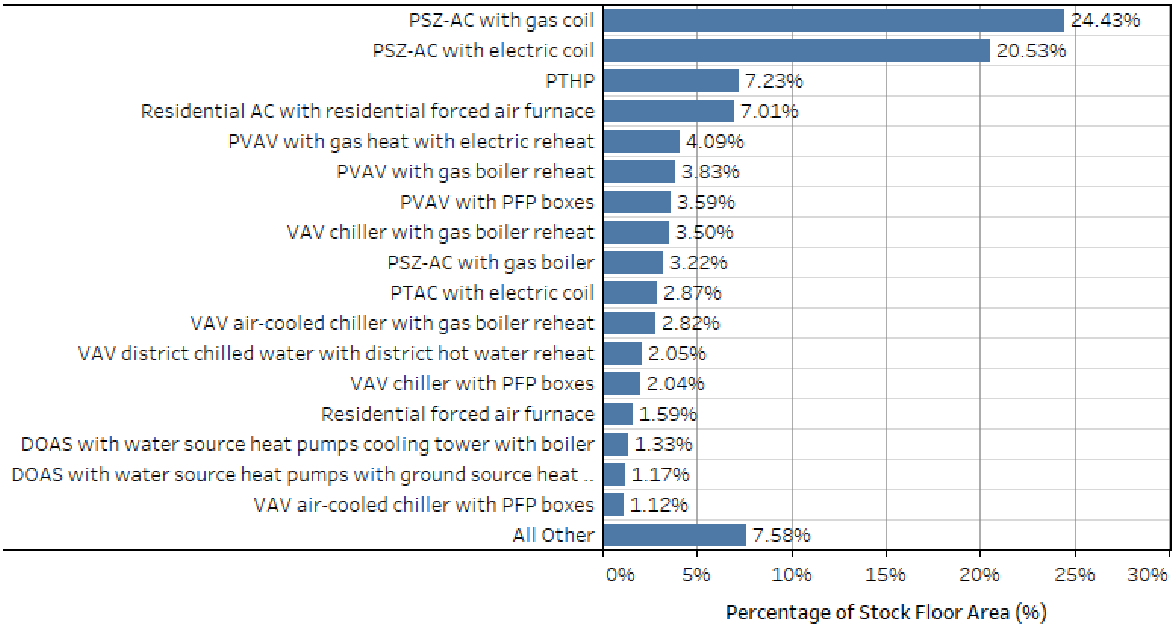

Fig. 1. ComStock HVAC system type prevalence by stock floor area.

Note that PSZ-AC stands for packaged single-zone air conditioner, PTHP stands for packaged terminal heat pump, PVAV stands for packaged variable air volume, DOAS stands for dedicated outdoor air unit, and PFP stands for parallel fan-power.

2.2. Baseline building stock RTU modeling

The state of the existing RTUs in ComStock is based on a combination of when the buildings were built and how the equipment has been updated over time, described in detail in the "ComStock Documentation" NREL report expected to be published in March 2023. Equipment performance is assumed to meet the energy code requirements in force at the time and place of installation. For this reason, most of the existing RTUs are modeled as constant air volume with single speed compressors. This is influential to the results in this analysis since energy savings will be calculated based on the energy performance of the ComStock baseline models versus an updated version of the ComStock baseline that uses the proposed HP-RTUs.

The in force energy code for the ComStock baseline is shown as a percentage of applicable floor area in Fig 2. Applicable floor area for this analysis includes ComStock buildings with "PSZ-AC with gas coil" and "PSZ-AC with electric coil" HVAC system types. Most ComStock baseline RTUs follow energy code requirements from the early 2000s. Other energy efficiency features such as demand control ventilation, energy recovery, and economizer control are only applied to baseline ComStock RTUs if required by the in force energy code for the particular model. The ComStock workflow checks the necessary characteristics of each RTU to determine if the feature is required. Similarly, heating, cooling, and fan efficiencies are set based on the in force code year. For models with the "PSZ-AC with electric coil" HVAC system type, the ComStock baseline will use electric resistance coils with a coefficient of performance (COP) of 1. For models with the "PSZ-AC with gas coil" HVAC system type, the ComStock baseline will use a gas furnace efficiency of generally around 80%.

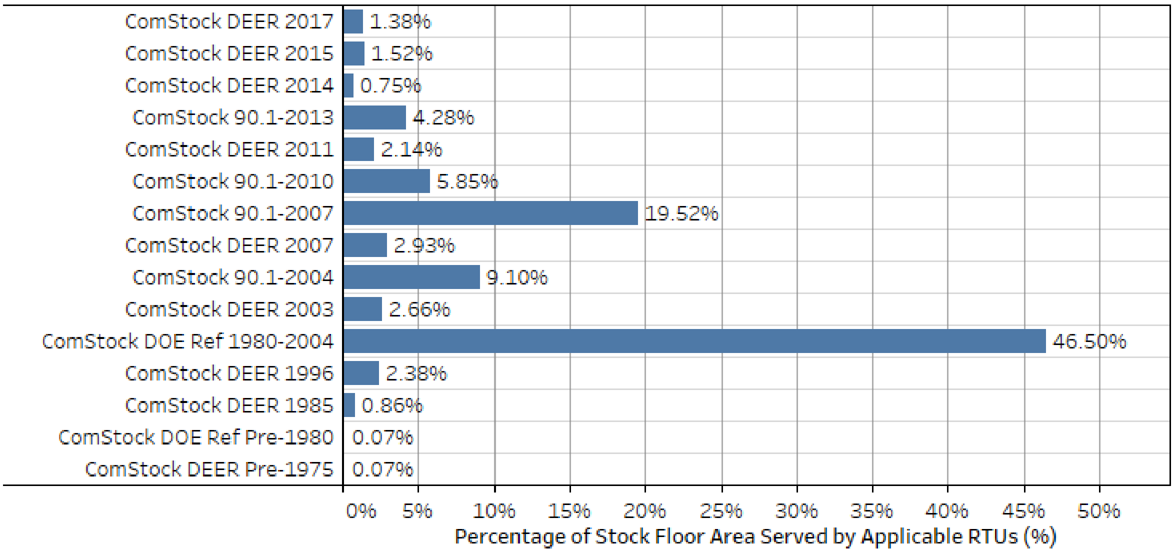

Fig 2: ComStock baseline in force energy code followed as a percentage of applicable floor area. Applicable floor area includes ComStock buildings with "PSZ-AC with gas coil" and "PSZ-AC with electric coil" HVAC system types.

2.3. Heat pump RTU modeling

The HP-RTUs are modeled using the EnergyPlus “AirloopHVAC:UnitarySystem” object [10], [11]. An OpenStudio measure is used in conjunction with the ComStock workflow to modify/remove any applicable RTUs in the ComStock baseline models (“PSZ-AC with gas coil” and “PSZ-AC with electric coil” in Fig. 1) and articulate the appropriate HP-RTU objects and settings. Non-applicable systems are not affected, nor are core operational parameters of systems such as schedules, thermostat setpoints, unoccupied operation behavior, and design outdoor airflow rates. Furthermore, energy-saving features found in applicable baseline RTUs such as airside heat/energy recovery, economizers, or demand control ventilation are preserved as-is for the new HP-RTU systems. This provides even comparability, noting that these features are feasible and available in HP-RTU systems. The modeling details of the HP-RTU system are described further in the following subsections.

2.3.1. Single-zone VAV operation

The modeled HP-RTUs utilize a single-zone VAV operation, which varies the supply airflow and discharge air temperature to efficiently maintain zone thermostat setpoints. During heating operation, as loads increase, first the supply air temperature is gradually raised until it hits a maximum threshold, and then supply airflow is increased until loads are met. During cooling operation, as loads increase, supply air temperature is gradually lowered until it meets a minimum threshold, and the supply airflow is increased until loads are met (Fig. 3) [12]. This is generally expected to provide fan energy savings during periods of reduced loads. The minimum supply airflow ratio modeled is 40%, which is common for single-zone RTUs [13]. The exception to the 40% minimum is when higher outdoor airflow rates are required to maintain ASHRAE Standard-62.1 minimum outdoor airflow rates: in these cases, the minimum flow rate to satisfy design outdoor air ventilation rates are modeled [14].

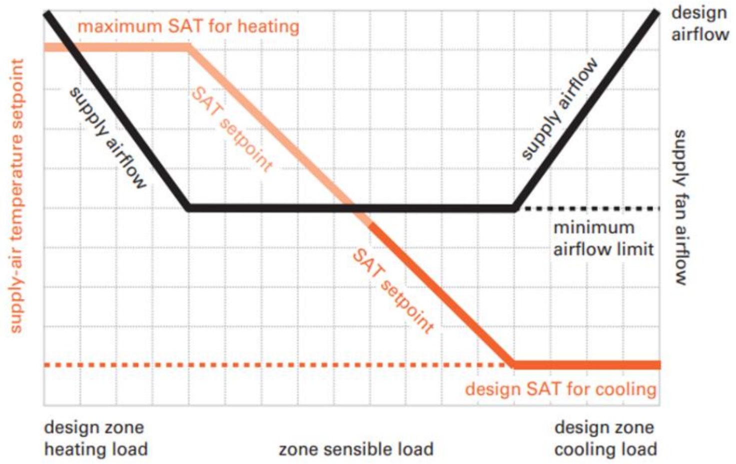

Fig. 3. Visual representation of single-zone VAV operation. Image from *[10]*.

2.3.2. Cooling performance

The variable-speed direct expansion (DX) cooling system in the proposed HP-RTUs is modeled using the EnergyPlus "Coil:Cooling:DXMultiSpeed" object using four speeds of cooling [10], [11]. The highest speed (speed 4) represents the cooling performance at rated conditions with the compressor fully loaded. The efficiency values used for this study are based on a 10-ton variable-speed RTU with a full-load COP of 3.6 at rated conditions with Integrated Energy Efficiency Ratio (IEER) above 17 [15]. Because the EnergyPlus COP input is compressor-only, and therefore removes supply fan energy, the modeling input is adjusted to 4.11 COP using the methodology from the Pacific Northwest National Laboratory's (PNNL's) Daikin Rebel study [15].

The other speed levels (speeds 1 through 3) represent lower compressor speeds, which would occur when the required load to be met is less than the full capacity of the unit. Each speed corresponds to a fraction of the rated capacity, a rated COP, and a rated airflow. Lower compressor speeds generally show higher COP values, which allows for higher efficiencies during these periods of partial loading. For instance, a PNNL lab testing and modeling study showed 20-50% annual cooling energy savings for variable speed RTUs over conventional RTU cooling systems [15]. The capacity fractions and COPs for the different compressor speeds were determined using NREL lab testing data for three variable-speed central ducted AC systems. Because the testing is based on residential central AC units rather than commercial RTUs, the values derived from the testing are normalized to the rated COP of 4.11 to better represent a commercially available HP-RTU for this study (Table 1) [15]. Variable speed HP-RTUs are capable of modulating to the specified fractions, but they may not do so in the same manner as the residential units the performance parameters are based on [13].

Table 1. Multispeed cooling coil performance parameters. Units with high outdoor air fraction may not achieve lower compressor speeds if it violates ventilation requirements.

Compressor Speed Level	**Capacity Fraction of Rated**	**COP Fraction of Rated**	**Applied HP-RTU COP**	**Sensible Heat Ratio Fraction**
Rated	1.00	1.00	4.11	1.00
4	1.00	1.00	4.11	1.00
3	0.67	1.08	4.44	1.01
2	0.51	1.11	4.56	1.03
1	0.36	1.07	4.40	1.11

Five performance curve modifiers are used for modeling the DX multispeed cooling objects. The performance curves were derived from separate work that used NREL lab testing data of three variable speed central ducted AC systems where values representative of the three systems are used. For multispeed objects, these modifier curves are specific to the compressor speed to which they are applied, so each speed will have its own set of curves. They are described as follows.

1. **Energy input ratio (EIR) as a function of part load ratio** – uses the calculated part load ratio to determine an EIR modifier from compressor cycling, which is multiplied against the full-load EIR for the stage (Fig. 4). Note that EIR is the inverse of COP, so decreasing the EIR increases the realized efficiency. For the multispeed units modeled in this work, this curve is only used for the lowest compressor speed where cycling losses may occur.
2. **Capacity as a function of temperature** – uses outdoor drybulb and indoor wetbulb temperatures to predict a capacity modifying factor that is multiplied against the rated capacity for each stage (Fig. 5). For heat pumps, the available capacity decreases with temperature.
3. **EIR as a function of temperature** – uses outdoor and indoor drybulb temperatures to determine an EIR (1/COP) modifying factor that is multiplied against the rated EIR for each stage for the time step (Fig. 6). Note that other modifier functions can also affect the final COP.
4. **Capacity as a function of flow** – modifies capacity based on the determined flow rate for a time step. This curve is not used since capacity is already accounted for in the speed level.
5. **EIR as a function of flow** – modifies EIR based on the determined flow rate for a time step. This curve is not used since the EIR is already accounted for in the speed level.

The cooling performance maps for EIR as a function of part load ratio, COP as a function of temperature, and capacity as a function of temperature are shown Fig. 4, Fig. 5, and Fig. 6, respectively.

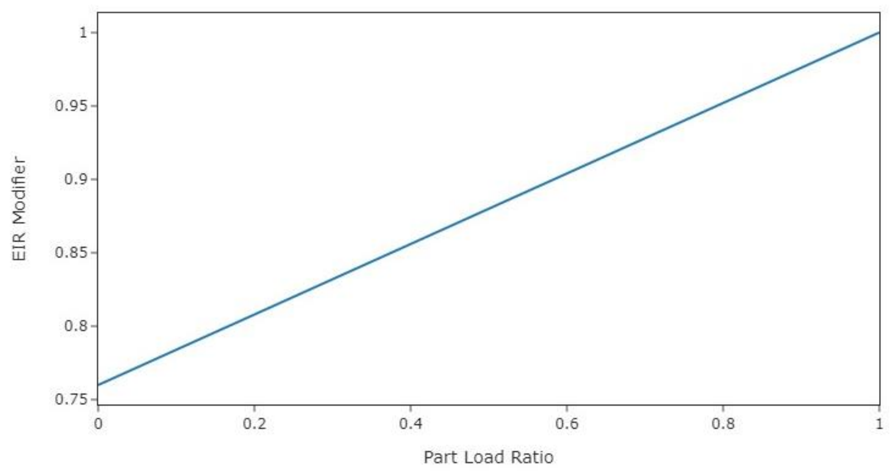

Fig. 4. Heating and cooling energy input ratio modifier as a function of part load ratio for all speed levels. This curve primarily captures losses to due part-load cycling at compressor speed 1. This value is divided by the EIR for the time step, which effectively decreases efficiency at lower part load ratios.

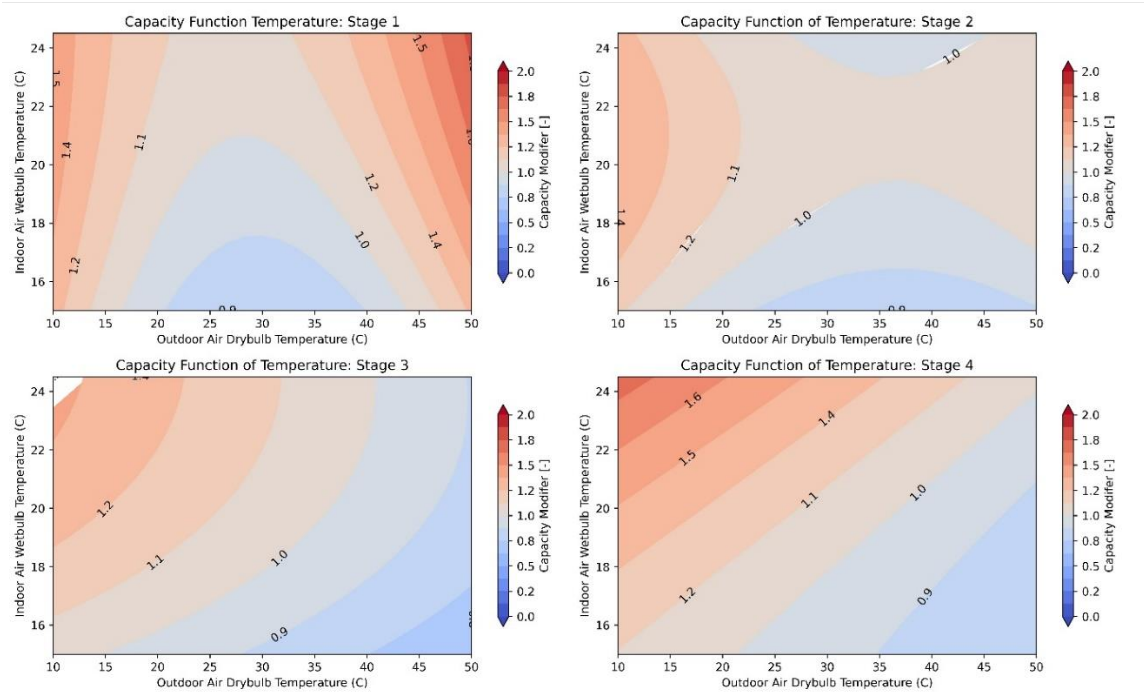

Fig. 5. Capacity as a function of temperature performance map for the four stages of cooling. This value is multiplied by the nominal capacity for each time step to determine the actual available capacity for the time step.

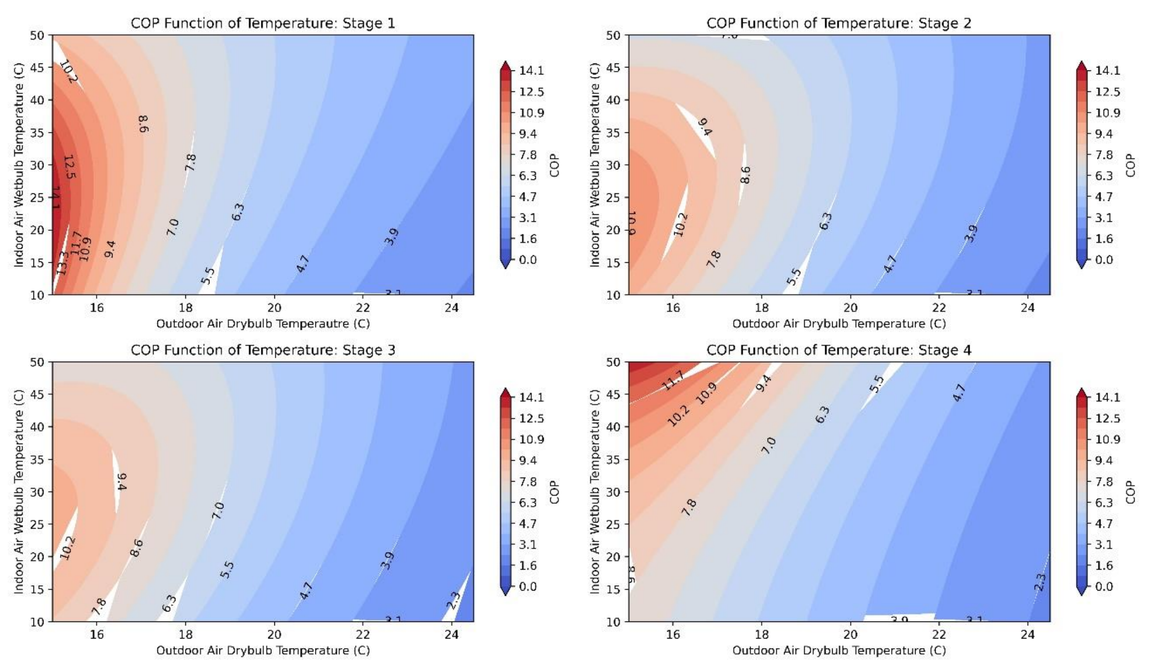

Fig. 6. COP as a function of temperature performance map for the four stages of cooling. Note that these COP values are for the compressor only – adding in supply fan energy would decrease the values presented.

2.3.3. *Heat pump heating performance*

The variable speed heat pump heating in the proposed HP-RTUs is modeled using the EnergyPlus "Coil:Heating:DXMultiSpeed" object using four speeds of heating [10], [11]. This object performs similarly to the "Coil:Cooling:DXMultiSpeed" object described previously. The rated efficiency values used for this study are based on a 10-ton variable-speed RTU with a full-load COP of 3.42 at rated conditions (8.3°C outdoor air temperature entering the condenser and 21.1°C drybulb indoor air temperature entering the coil) [15]. Because the EnergyPlus COP input is compressor-only, and therefore removes supply fan energy, the model input is adjusted to 3.8 COP, using the methodology from PNNL's study [15]. The parameters for each stage of heating are shown in Table 2. The capacity and COP fractions for each speed level were determined using manufacturer-provided data for a variable-speed central ducted forced air heat pump system (Table 2). The data is roughly 10 years old but is expected to be a reasonable representation of a variable speed system. The heating COP increases with lower speed levels similarly to what was described for the cooling COPs. Because the testing is based on residential central HP units rather than commercial RTUs, the values derived from the testing are normalized to the rated COP of 3.8 to better represent a commercially available HP-RTU for this study (Table 1) [15]. Variable speed HP-RTUs are capable of modulating to the specified fractions, but they may not do so in the same manner as the residential units the performance parameters are based on which emphasizes the need for additional research in this area [13]. The minimum operating temperature for the heat pumps is modeled at -17.8°C, which is the default setting for some manufacturers. The compressor will lock out below this temperature and only backup heat will be available.

Table 2. Multispeed heating coil performance parameters. COP values are at rated conditions and vary based on temperature.

Compressor Speed Level	**Capacity Fraction of Rated**	**COP Fraction of Rated**	**Applied HP-RTU COP**
Rated	1.00	1.00	3.80
4	1.00	1.00	3.80
3	0.85	1.05	3.98
2	0.48	1.24	4.71
1	0.28	1.45	5.51

Similar to the DX multispeed cooling objects, five performance curve modifier types are used for modeling the DX multispeed heating objects. The descriptions of these are discussed in the cooling performance section of this document, with the only difference being that the heating coils use indoor air drybulb temperature as opposed to indoor air wetbulb temperature used for the cooling coils. The performance curves were derived from manufacturer-provided data for a central ducted variable speed heat pump system. The resulting performance maps for all speed levels are shown in Fig. 4, Fig. 7 and Fig. 8 for EIR as a function of part load ratio, COP, and capacity retention, respectively. As expected with heat pumps, heating COP and capacity generally reduce with outdoor air drybulb temperature.

Heat pump performance maps are especially impactful due to the general reduction of capacity and efficiency at lower outdoor air temperatures where increased heating loads often occur. This study attempts to utilize the best available data, as described previously, as this will notably impact the results. However, it should be emphasized that complete heat pump performance data is still scarcely available at the time of this study, especially for variable speed commercial RTUs and low temperature operation. This adds limitations to the understanding of heat pump performance and operation in this analysis. Further research on heat pump performance could increase confidence in heat pump modeling, and this study may be updated as more data becomes available.

Comparisons of modeled performance data versus alternative data sources were made, where possible, for validation that the performance data used is reasonable. Table 3 and Table 4 compare some key points on the modeled heat pump performance maps for COP and capacity retention as a function of outdoor air temperature, respectively, with other available data sources for validation. The first data source is for the variable speed Daikin Rebel HP-RTU, with specification sheet data specifying capacity and COP at 8.3°C (rated) and -8.3°C [11]. The second source is a Rheem two-stage HP-RTU with heating performance data at various outdoor air temperatures [16]. The last data source is from a study that performed lab testing on a Carrier cold climate variable speed HP-RTU, which provides COP values at various outdoor air temperatures [17].

The modeled HP-RTU outperforms the capacity retention of the reference units by 5 to 9%, with the largest difference occurring with the Rheem unit at -17.8°C (Table 3). For COP retention, the modeled HP-RTU outperforms the reference units by 3% to 14%, with the largest difference occurring with the Rheem unit at -17.8°C (Table 4). Although there are some notable differences between the modeled and reference unit performance, and the modeled HP-RTU outperforms the reference units in all cases, these comparisons still suggest the modeled HP-RTU performance is reasonably appropriate compared to other available data, especially considering they are different units from different data sources. Note that no alternative data sources were found for comparing part-load performance or the impacts of cycling on variable speed heat pump units, further emphasizing the need for more research in this space to increase modeling confidence.

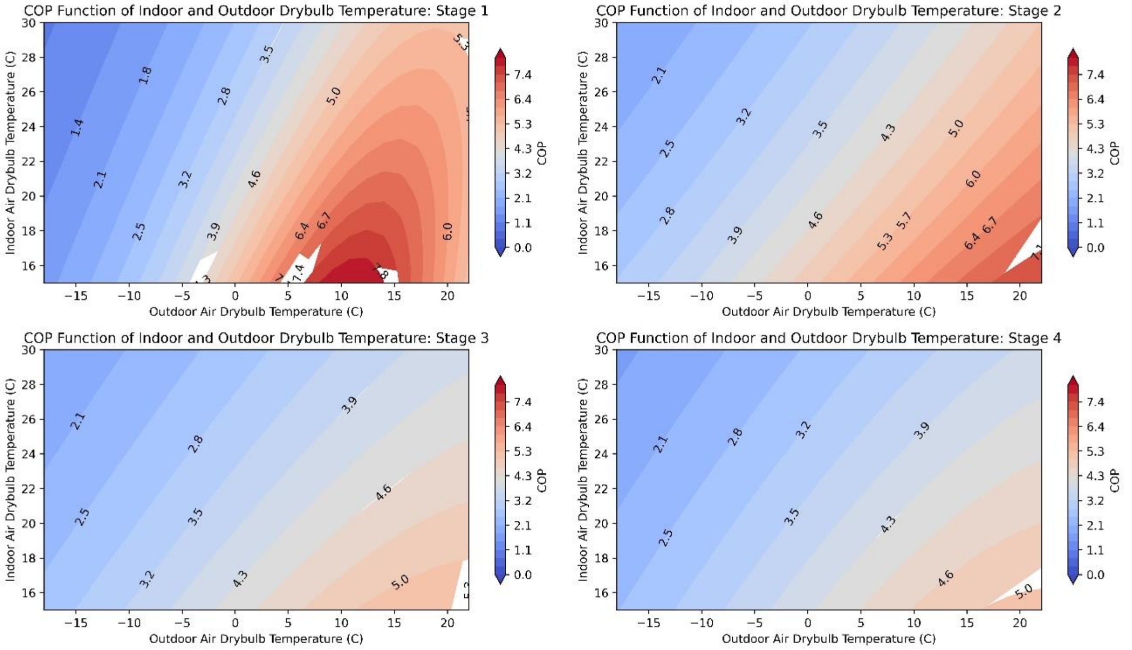

Fig. 7. COP as a function of temperature performance map for the four stages of heating. Note that these COP values are for the compressor only – adding in supply fan energy would decrease the values presented.

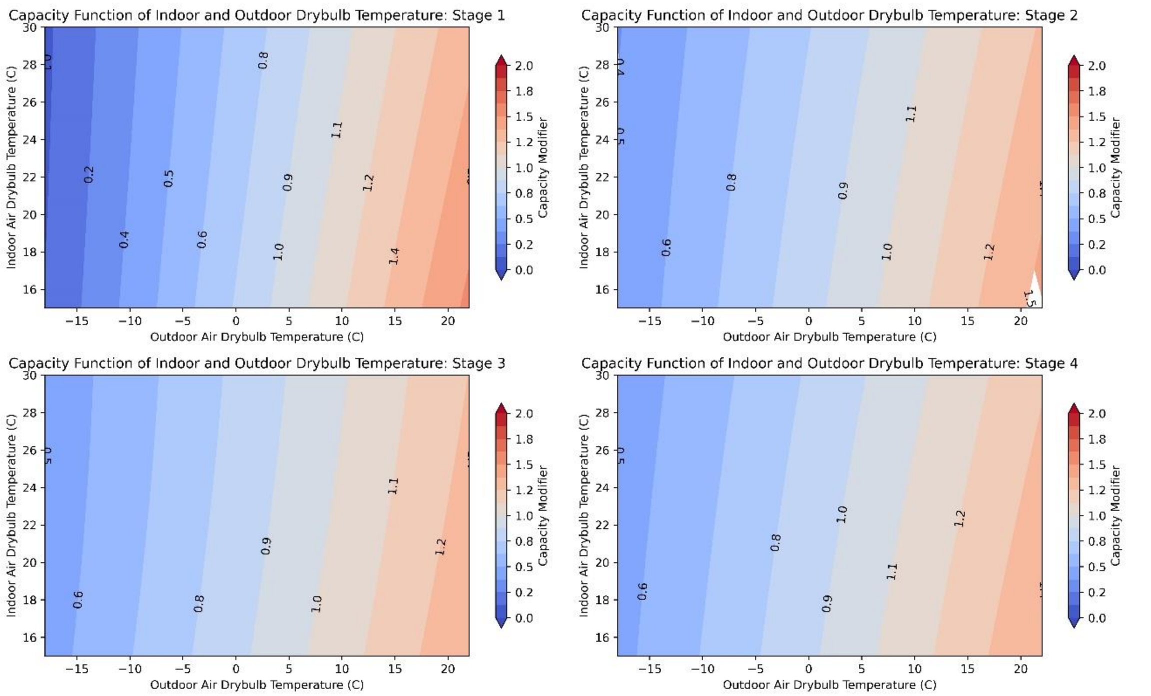

Fig. 8. Capacity as a function of temperature performance map for the four stages of heating. This value is multiplied by the nominal capacity for each time step to determine the actual available capacity for the time step.

Table 3. Capacity retention as a function of outdoor air temperature comparison for Daikin Rebel, Rheem Renaissance, and the modeled HP-RTU performance curves.

Reference Temperature, °C	**8.3°C**	**-8.3°C**	**-17.8°C**
Modeled HP-RTU Capacity Fraction	1	0.64	0.45
Daikin Rebel Capacity (kW)	30.8	18.0	-
Daikin Rebel Capacity Fraction	1	0.59	-
% Diff. Modeled HP-RTU vs. Daikin Capacity Fraction	-	7.80%	-
Rheem Renaissance Capacity (KW)	31.5	19.1	12.9
Rheem Renaissance Capacity Fraction	1	0.61	0.41
% Diff. Modeled HP-RTU vs. Rheem Renaissance Capacity Fraction	-	5.47%	9.33%

Table 4. COP comparison of the modeled HP-RTU, the Daikin Rebel, and a lab-tested Carrier unit. Note that the COPs associated with the modeled HP-RTU and Rheem unit are compressor only while the other include the supply fan. Including the supply fan in the calculation will decrease the COP.

Reference Temperature, °C	**8.3°C**	**-8.3°C**	**-17.8°C**
Modeled HP-RTU COP (compressor only)	3.80 (speed 4)	2.66 (speed 4)	2.11 (speed 4)
Modeled HP-RTU COP Fraction (compressor only)	1	0.70	0.55
Daikin Rebel COP	3.42	2.38	-
Daikin Rebel COP Fraction	1	0.70	-
% Diff Modeled vs. Daikin COP Fraction	-	0%	-
Carrier COP Estimate	3.1	2.1	1.62
Carrier COP Fraction	1	0.68	0.52
% Diff Modeled vs. Carrier COP Fraction	-	2.9%	5.5%
Rheem Renaissance COP (compressor only)	4.2	2.77	1.98
Rheem Renaissance COP Fraction (compressor only)	1	0.66	0.47
% Diff Modeled HP-RTU vs. Rheem Renaissance COP Fraction	-	5.8%	14.3%

2.3.4. Heat pump sizing & backup heating

The sizing of heat pumps is non-trivial since the same system is used for both heating and cooling. Heat pumps in colder climates usually require a source of supplemental heat, which today is often sized to meet the entirety of the heating load. This is because heat pump capacity is reduced as outdoor ambient temperatures decrease, which generally corresponds to the highest heating loads for the building. Furthermore, compressor lockout controls are often implemented in heat pump systems, which disable heat pump operation below a certain temperature [13]. This would require the supplemental heat source to be sized to meet loads below this temperature. Because the supplemental heat source in colder climates is then often sized to meet the design heating load, the system can then be sized based on the required cooling capacity with the assumption that the supplemental heat source will address any heating load beyond the corresponding capacity of the heat pump, avoiding the need to purchase a larger capacity unit. Supplemental heat is less of a concern in warmer climates where the design cooling load exceeds the design heating load, even when accounting for heat pump capacity degradation at lower temperatures, and where the design heating temperature is well above any minimum compressor lockout temperature.

The supplemental heat source is often electric resistance, which has an effective site COP of 1, while the heat pump system will often demonstrate a site COP much higher than this even at temperatures down to -17.8°C. Sizing heat pump systems to address more of the heating load is sometimes suggested since the heat pump heating is more efficient than electric resistance, so long as the sizing of heat pump system still enables effective operation for both heating and cooling [18], [19]. However, this analysis simply sizes the heat pumps based on cooling load, and reserves studying the impact of other sizing approaches for future analyses. The minimum outdoor air temperature for heat pump operation is modeled as -17.8°C, which aligns with the default minimum temperature for some manufacturers, noting that this default value can change between manufacturers and can be overridden, which would impact performance [13].

2.3.5. Defrost operation

Frost formation can occur on the outdoor unit during heat pump heating operation due to humidity in the outdoor air condensing and freezing on the cold outdoor coil. Frost needs to be periodically removed so the coil can function properly. This is generally done using either an electric resistance coil or by reversing the operating of the heat pump to remove the frost buildup, both of which result in additional energy consumption. This analysis uses reverse cycle as it is common in practice and does not require additional heating coils.

Reverse cycle defrost inhibits the heating capacity of the heat pump system which may require the use of lower-efficiency supplemental heating during these times. Additionally, reversing the cycle of the heat pump causes additional heating load in the RTU since the system is essentially in cooling mode, which EnergyPlus adds to the total effective heating load [10], [11].

Control of the defrost cycle can also vary. Some units use a set time fraction, where the unit operates in defrost mode for a specified time when outdoor air temperatures are below a specified temperature. This analysis uses EnergyPlus' "on-demand" defrost operation, which estimates the amount of time needed for defrost based on a set of empirical calculations dependent on outdoor air wetbulb temperature, coil temperature, and other parameters: these calculations are described in detail in the EnergyPlus Documentation [10], [11].

2.3.6. Greenhouse gas emissions

Greenhouse gas (GHG) emissions savings are estimated in this work using similar methods to those presented in [9]. For direct fuel combustion, the impact of GHG emissions was estimated using emissions factors published by the ANSI/RESNET/ICCC: natural gas = 63.27 kg/GJ (147.3 lb/MMBtu), propane = 76.4 kg/GJ (177.8 lb/MMBtu), and fuel oil = (84.15 kg/GJ) 195.9 lb/MMBtu [20], [21].

Estimating GHG emission impacts due to electricity consumption is more complicated. It requires projecting the resource composition of the electric grid as a result of load increase, which may be substantial when electrifying commercial building HVAC equipment as is done in this study. This is important since the prevalence of cleaner grid resources will directly impact the GHG emissions per kWh of electricity used. Furthermore, the resources producing electricity on the electric grid vary based on location and time of day, which can impact GHG estimates if not accounted for properly [22], [23].

The choice of grid emissions scenario will impact emissions factors and therefore analysis results [24]. This analysis uses NREL's Cambium dataset. The Cambium dataset provides emissions factors that vary based on time of day and U.S. grid region to capture geospatial and temporal variation in converting consumed kWh to GHG emissions [22], [23]. The Cambium dataset includes various grid scenarios to choose from: this analysis uses Cambium long-run marginal emissions rate "low renewable energy cost 15-year" scenario data [22], [23]. The timeseries grid emission factors in this dataset are coupled with ComStock timeseries electricity consumption outputs for each model based on time of day and the location-based Cambium grid region to estimate the GHG emissions impacts of transitioning to HP-RTUs. The published emissions values presented represent a single year of emissions, which are calculated using a weighted average year over the levelization period.

2.3.7. Heat pump annual performance

For this analysis, annual effective heating COP is calculated for the ComStock simulations to understand the performance of the heat pump heating system, as shown in Equation 1.

$$COP_{effective} = \frac{q_{hp} + q_{supp}}{E_{hp} + E_{supp} + E_{defrost}} \tag{1}$$

$q_{hp} = annual\ heating\ output\ energy\ from\ heat\ pump$
$q_{supp} = annual\ heating\ output\ from\ supplemental\ heating\ coil$
$E_{hp} = annual\ heat\ pump\ heating\ electricity\ input$
$E_{supp} = annual\ supplemental\ heating\ electricity\ input$

$E_{defrost} = annual\ reverse\ cycle\ defrost\ electricity\ input$

The effective annual COP calculation in Equation 1 includes performance and capacity degradation with temperature, heat pump sizing limitations, and heat pump defrost cycles and the associated supplemental heating coil operation. This calculation does not include supply fan energy, which is commonly included in product specification sheet values for rated COP values. It is also important to note that natural gas furnaces and boilers are often rated on thermal efficiency and exclude the delivery system (fans or pumps). Including supply fan energy in the COP calculation could lower the values presented in this study by 20-40%.

3. Results and Discussions

3.1. Annual energy and GHG savings

Ten percent (10%) total annual site energy savings (130 TWh) can be achieved for the modeled U.S. commercial buildings, including savings of 8% for electricity, 17% for natural gas, and 50% for other fuels (fuel oil and propane). Savings numbers reflect the 66% of the U.S. commercial building stock currently modeled in ComStock at the time of this analysis. The total site energy savings for just the 45% of buildings with RTUs is 21% (Fig. 9b). The site energy savings for electricity are primarily from fan energy savings from the single-zone VAV operation with high-efficiency fans and cooling savings from high-efficiency variable-speed compressor operation. Notably, these fan and cooling energy savings could also be achieved with high-performance non-heat-pump RTUs. The median fan and cooling energy savings are approximately 35% and 40%, respectively, which align with the results from a lab testing and modeling study performed by PNNL on variable-speed RTUs [15]. There are also substantial electricity savings in the heating end use from converting electric resistance RTUs ("PSZ-AC with electric coil") in the baseline to HP-RTUs which have higher heating COPs. The electricity savings presented in this study would be reduced, possibly causing a net electricity penalty, if replacement of baseline electric resistance RTUs was omitted from the analysis.

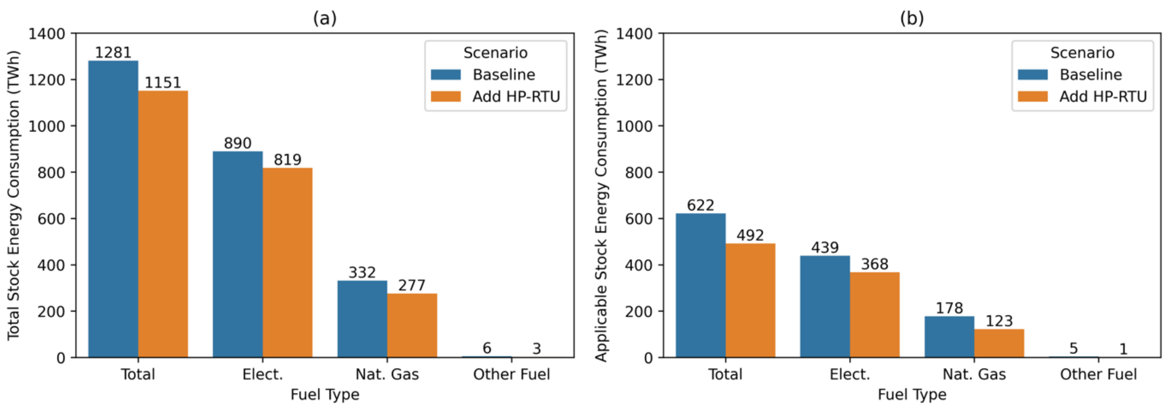

Fig. 9. Aggregate annual site energy consumption by fuel type for (a) the total U.S. commercial building stock modeled by ComStock and (b) the U.S. commercial building stock applicable to the HP-RTU scenario.

The site energy savings from the combustion fuels (natural gas and other) can be attributed to transitioning the baseline gas-fired RTUs to electric heat pump heating. Switching from a gas heated system to an electric heated system will add electricity consumption to the building, although some of the increase in electricity may be offset by fan and cooling savings. HP-RTU replacement alone may not completely electrify applicable buildings if combustion fuels are still used for water heating, appliances, and non-RTU HVAC systems.

Nine percent (9%) total annual GHG emissions savings (32.4 MMT) can be achieved for the modeled U.S. commercial buildings by replacing all existing RTUs with HP-RTUs (Fig 10a). This represents 6.6% GHG emissions reduction for electricity consumed by these buildings, 16.6% GHG emissions reduction for natural gas, 66.7% for fuel oil, and 50% for propane. Similar to the site energy savings projections, these savings numbers reflect the 66% of the U.S. commercial building stock currently modeled in ComStock at the time of this analysis. The reductions in GHG emissions are primarily caused by net reduced energy consumption across all fuel types (Fig. 9). However, emissions for electricity are also impacted by temporal and geospatial variation in grid emissions factors informed by the Cambium dataset. Electricity consumption may be shifted to times with higher or lower grid emissions factors based on the predicted grid resources for that time. As previously mentioned, this study chose a single Cambium scenario for grid emissions factors. The choice of scenario impacts emissions results [24]. Electrification will become more attractive from a GHG emissions standpoint as the grid continues to generate more electricity from lower GHG sources, effectively lowering grid emissions factors. Geospatial variation in emissions factors between grid regions will impact the GHG reductions from electrification in the particular region [22], [23].

The baseline ComStock model currently underestimates natural gas consumption by around 30% for the modeled buildings, as shown in Figure 193 of the technical report on calibration of ComStock [5]. The source of this error is unknown but may be attributable to some combination of underestimation of heating load, overestimation of primary combustion equipment efficiencies, or misrepresentation of inefficiencies in controlling or maintaining HVAC systems. Because the baseline natural gas consumption is used as a starting point for comparison, the savings potential for HP-RTUs may be higher than the results shown described above.

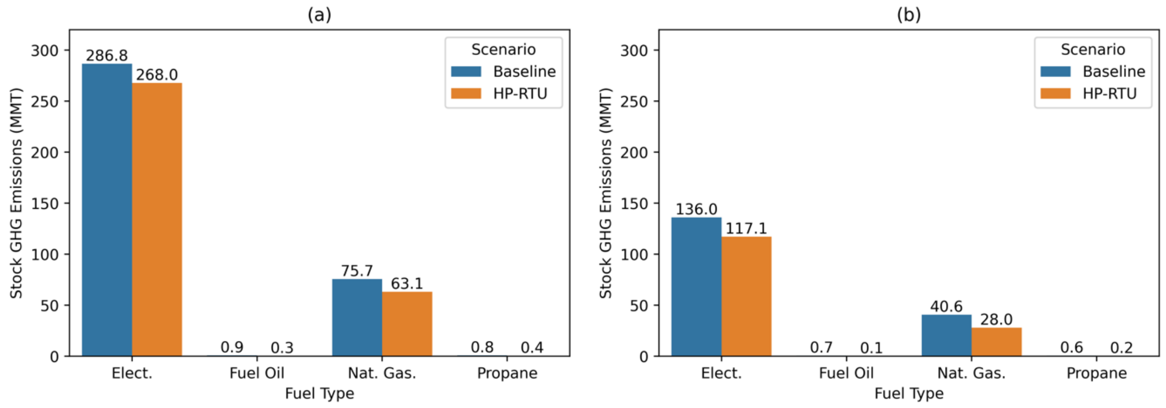

Fig 10. Aggregate GHG emissions by fuel type for (a) the total U.S. commercial building stock and (b) the U.S. commercial building stock applicable to the HP-RTU scenario.

3.2. Heat pump annual performance

Simulation results show variation by state in average annual effective heating COP of the HP-RTUs, with lower values around 2 COP and the highest values above 5 COP (Fig. 11). This annual average heating COP includes performance and capacity degradation, heat pump sizing limitations, and heat pump defrost operation and associated supplemental heating coil operation, but does not include supply fan energy which would lower the COP. Note that some of these higher COPs are attributed to operating the HP-RTUs at lower compressor speeds during part-load conditions (Table 2; Fig. 7), which is possible with the variable speed units modeled in this study. These average COPs would likely be reduced with constant speed HP-RTUs that cycle the full compressor capacity to meet loads.

States with warmer climates generally show higher heating COPs for HP-RTUs than states with colder climates (Fig. 11). This behavior is expected since heat pumps have better performance in warmer conditions. Additionally, in warmer climates the design heating load is generally closer to or below the design cooling load. Since this study sizes heat pumps based on the cooling load, the heat pump heating capacity in warmer climates will naturally meet a larger portion of the design heating load compared to cooler climates, leading to higher annual average heating COP values. However, the heating energy use intensity (annual energy used for heating divided by floor area) in colder states can be more than 10 times higher than in the heating intensity in warmer climates, which stresses the importance of cold climate performance (Fig. 12).

State average percentage of total heating electricity used by the supplementary system ranges from 6% to 56% (Fig. 13). This is due to the reduced capacity of heat pumps under cold ambient conditions, as well as the fact that the heat pumps are generally being sized to a smaller fraction of the design heating load (when sized to design cooling load, as is done in this study). States with higher fraction of supplementary heating generally correspond to lower COPs as expected (Fig. 11 & Fig. 13). Note that supplementary heating can also be induced by reverse-cycle defrost operation which temporarily disables heat pump from heating.

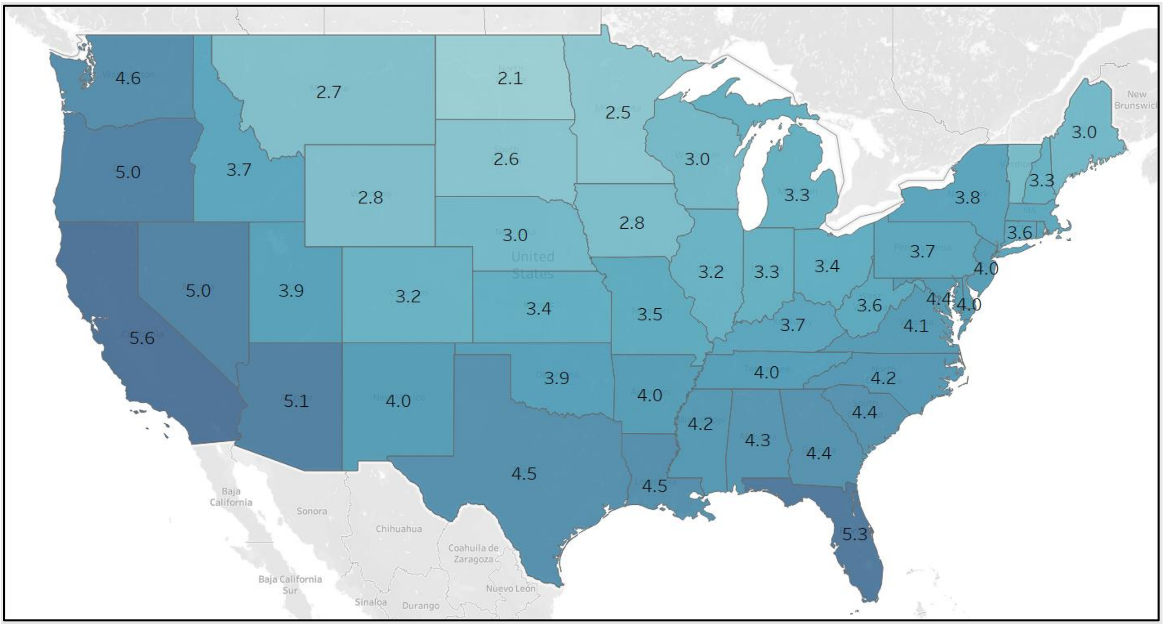

Fig. 11. Stock annual average effective heating COP by state. Effective heating COP is the total heating energy output divided by total heating energy input. Heating energy output includes heating from the heat pump and supplemental heating. Heating energy input includes heat pump compressor and outdoor fans, supplemental heating, and defrost. The heating energy input does not include associated supply fan energy use. Including supply fan energy use would reduce COPs.

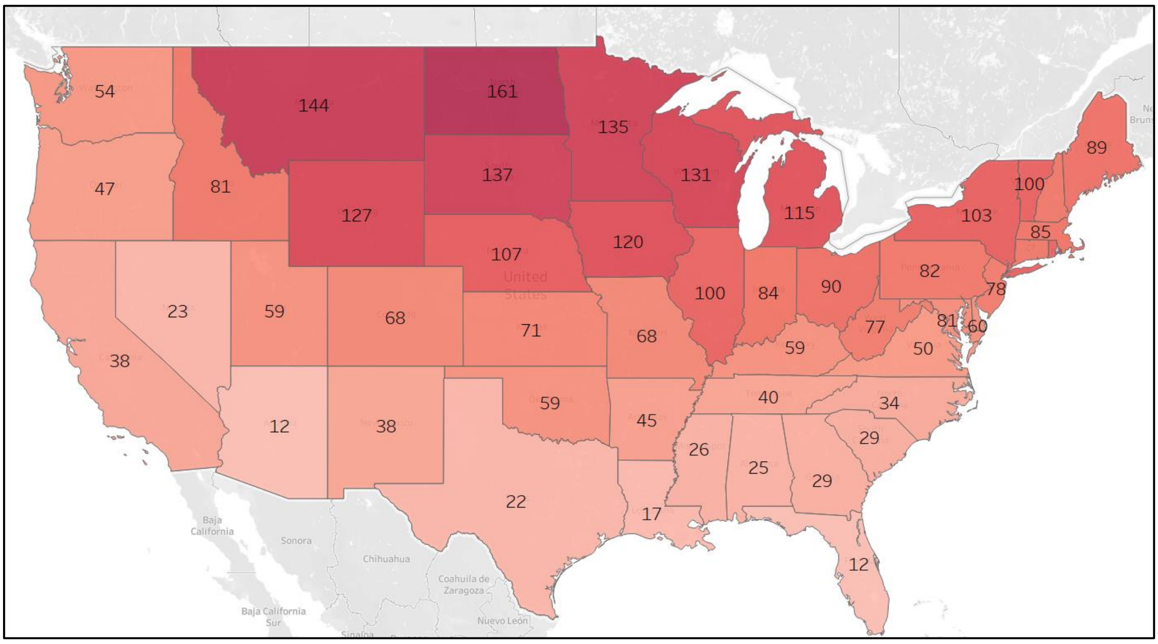

Fig. 12. ComStock baseline stock annual average heating energy use intensity (EUI; kWh/m^2/year) by state.

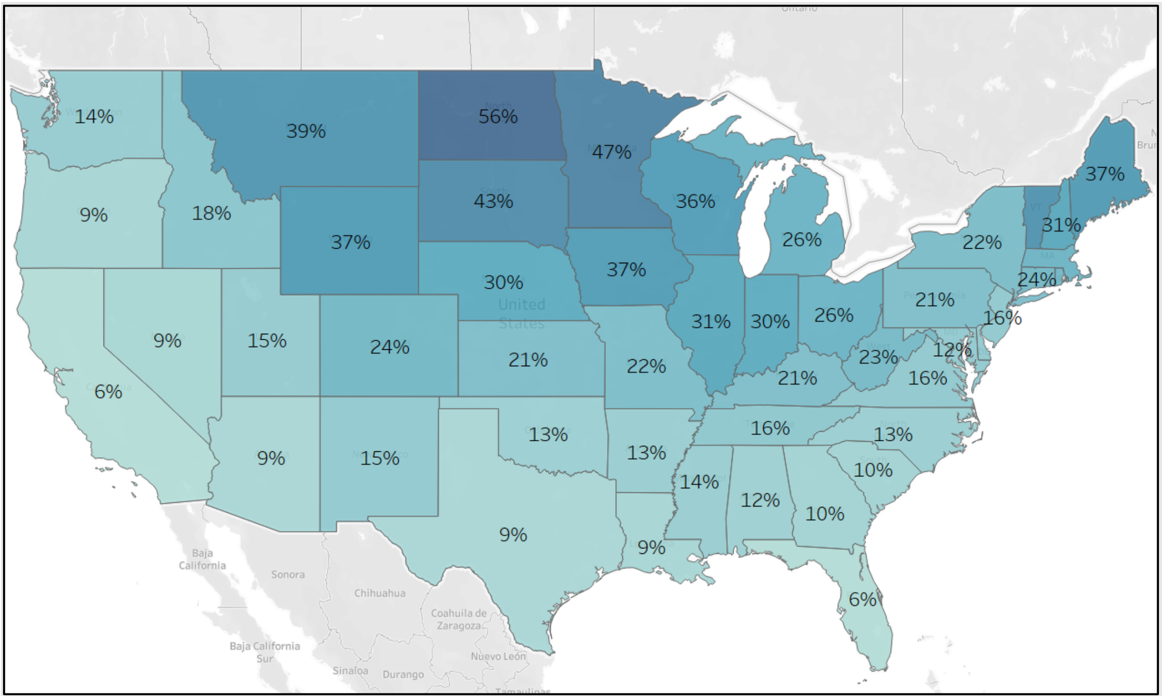

Fig. 13. Stock annual average percent heating electricity input used for supplement heating by state. Note that supplemental heating occurs due to insufficient heating capacity of the heat pump which can be further exacerbated from defrost operation.

Most states have multiple climate zones represented and therefore state average values do not tell the whole story. Fig. 14 shows the distribution of annual heating COPs for two different states. The average for South Carolina is 4.2, but the distribution extends from a COP of 3.1 up to 5.7, while the distribution for Michigan is more closely distributed to its average heating COP of 3.2. One question is why the range of annual heating COPs is so large in a single climate zone. A full analysis and discussion of the reasons for this range is outside the scope of this paper, but generally includes sizing considerations, particularly for buildings with either very unbalanced heating and cooling capacity requirements, heating energy consumption, or both. In some cases, climate can vary substantially within it state which could also cause wider COP distributions.

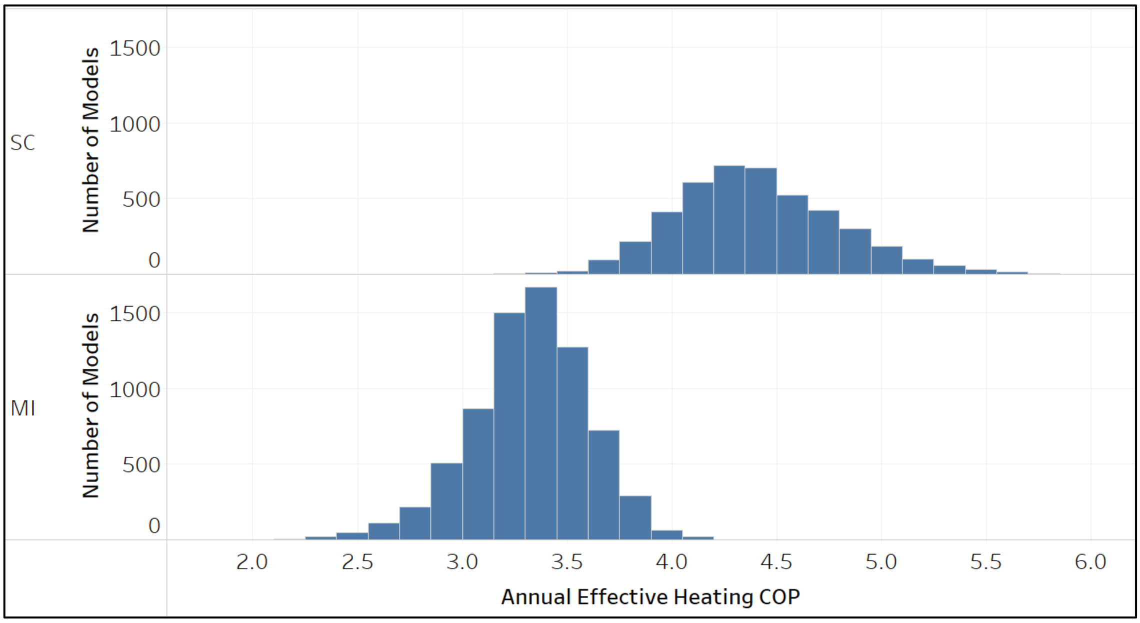

Fig. 14. Distribution of annual effective heating COPs for models in South Carolina (SC) and Michigan (MI)

4. Conclusion

Transitioning the existing RTUs (gas fired and electric resistance) in the U.S. commercial stock to high-performance HP-RTUs is a promising pathway toward electrification and reducing GHG emissions. Comstock energy modeling results show 10% total site energy savings (130 TWh) for the modeled building stock from this transition, noting that ComStock currently models 66% of the building stock at this time. GHG emissions also showed savings totaling around 9% (32.4 MMT CO_2) for the modeled building stock, noting that this analysis uses a single reference scenario from the Cambium dataset for grid emissions factors, the choice of which can impact these estimates. GHG and site energy and savings are achieved by transitioning from gas furnace and electric resistance heating to high-efficiency heat pump heating, and from fan and cooling savings by converting the older, less efficient RTUs currently in the U.S. commercial stock to high-performance variable speed RTUs. Note that the fan and cooling savings could also be achieved with high-performance non-heat-pump RTUs. GHG savings would be expected to increase from electrification pathways as the U.S. electric grid becomes less carbon intensive, therefore reducing the carbon contribution per kWh of electricity.

This analysis also presents average annual heating system COPs by state from the ComStock simulations, which demonstrates the effective COP that can be expected when factoring in impactful heat pump considerations such as supplementary heat, defrost cycles, realistic/economical sizing practices, and efficiency/capacity degradation at lower temperatures. ComStock modeling results suggest annual effective heating COP averages vary considerably by state, with a low of 2.1 and a high of 5.6, and even within a state. As expected, states with warmer climates generally show higher COPs, noting that states in colder climates generally require substantially more energy for heating, which increases the importance of cold climate performance.

Future research could analyze the impact of transitioning existing RTUs to HP-RTUs on electric peak demand, as well as the impact of adding air side energy recovery or utilizing different fuel sources for supplemental heat. Furthermore, although this study attempts to utilize the best available heat pump performance data and makes comparisons to alternative data sources where possible, limitations exist due to the small amount of performance data available for commercial variable speed HP-RTUs. Heat pump energy modeling can be sensitive to performance assumptions such as the variation in COP and available capacity under different operating conditions, which can impact analysis results. Future research could expand this body of knowledge through additional lab testing and field evaluation of HP-RTUs to increase confidence in heat pump modeling, which this work could be updated to include.

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