White Paper

Savings Calculations for Residential Air Source Heat Pumps

The Basis for Modifying EFLH and Seasonal Efficiency Factors for "Whole House" and "Displacement" Applications

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Prepared for

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Executive Summary

This white paper uses a temperature bin analysis to predict the seasonal annual efficiency for several types of cold climate Air Source Heat Pumps (ASHPs). The analysis also predicts the fraction of the seasonal space heating load that can be met by the ASHP unit. Several scenarios were modeled for whole house and displacement applications using various ASHP systems with different control options. The database from the NEEP Cold Climate Heat Pump List was used to analyze more than 4000 products. That data was used to evaluate the performance of various scenarios. The results were averaged and fit to linear regression models for the various scenarios and climates so that average factors could be developed for use in the TRM. The TRM calculation process requires basic information to predict energy savings:

- Design load (Manual J or other approved method) for the house
- Performance data about the ASHP (HSPF and other data from the NEEP database)
- The type of ASHP unit (e.g., central, ducted; mini-split ductless, multi-split, compact ducted)
- Other information about the application and controls

Appendix B includes the factors required for the TRM calculation process for a range of scenarios and climates.

Overview

This white paper is intended to provide the background and basis for energy savings calculations for air source heat pumps in different applications:

- A “Whole House” or “Full Load” heating application where the Air Source Heat Pump (ASHP) is sized to meet 90% or more of the design building load. In some cases, an integrated backup heating source (electric or fossil fuel) meets the remainder of the load, especially on the coldest days.
- A “Displacement” or “Part Load” heating application where the ASHP is only sized to displace a portion of the heating load. The original heating system remains in place to meet heating load not provided by the heat pump.

This paper develops a bin analysis method that determines the portion of the total heating load that can be attributed to the ASHP in both the residential applications described above. The total heating load for the home will be determined by the BEFLH method consistent with what is described in the GSHP White Paper (Henderson 2020) – this will ensure that the building load assumptions are consistent for all heating technologies.

Rated performance data for ASHPs are determined according to the ANSI/AHRI Standard 210/240. In addition, the Northeast Energy Efficiency Partnership (NEEP) has developed a Cold Climate Specification. To meet the NEEP specification, ASHPs must be variable speed and must provide a heating COP ≥ 1.75 at an outdoor temperature of 5°F at maximum speed. In addition, they must have an HSPF ≥ 10 for ductless systems, an HSPF ≥ 9 for ducted systems, and a SEER of ≥ 15 for all systems. Manufacturers are required to provide additional performance data (non-rated engineering data) for products to be included in the NEEP Cold-Climate Heat Pump database. This AHRI and NEEP published data is the basis for determining the heat pump performance with the bin analysis.

The results of the bin analysis for all the ASHPs in the database informs the development of factors for use in the TRM calculations. Factors are developed for various types of heat pumps (central-ducted, ductless, multi-splits, etc.) in the different applications (Whole House, Displacement). The factors determine:

- The portion of the annual space heating load met by the ASHP (i.e., fraction of total BEFLH)
- The seasonal heating efficiency of the heat pump operating to meet that load (i.e., as a fraction of the rated efficiency)

Published ASHP Performance Data

Published performance data for ASHP equipment includes the Heating Season Performance Factor (HSPF) and the rated heating capacity at 17°F and 47°F which are defined in AHRI Standard 210/240. This data is available in the AHRI Directory. Some manufacturers also include the COP (or power input) for those rating test conditions in their product datasheets. Now the NEEP Cold Climate Specification includes the COP at 5°F as an additional performance metric for cold climate heat pumps. Manufacturers must provide Capacity and COP for minimum and maximum speed operation at three temperatures (5°F, 17°F, and 47°F). The NEEP database is known as the Cold Climate Heat Pump List: https://neep.org/ashp.

The ASHP bin analysis presented in this paper is based on published data available from the NEEP database for ASHPs that meet their Cold-Climate Specification[1]. These metrics are listed in Table 1 below.

Table 1. Key Performance Metrics Available from AHRI and the NEEP Database

Published Data	Description	Source
HSPF	Heating Seasonal Performance Factor (based on DOE Region IV)	AHRI 210/240
$QH_{47,\ rated}$	Rated Heating Capacity at 47°F	AHRI 210/240
$QH_{17,\ rated}$	Rated Heating Capacity at 17°F	AHRI 210/240
$COP_{47,\ rated}$	Rated Heating COP at 47°F	NEEP
$COP_{17,\ rated}$	Rated Heating COP at 17°F	NEEP
$QH_{47,\ max}$	Maximum Heating Capacity at 47°F	NEEP
$QH_{17,\ max}$	Maximum Heating Capacity at 17°F	NEEP
$QH_{5,\ max}$	Maximum Heating Capacity at 5°F	NEEP
$COP_{47,\ max}$	COP at Maximum Heating Capacity at 47°F	NEEP
$COP_{17,\ max}$	COP at Maximum Heating Capacity at 17°F	NEEP
$COP_{5,\ max}$	COP at Maximum Heating Capacity at 5°F	NEEP
$QH_{47,\ min}$	Minimum Heating Capacity at 47°F	NEEP
$QH_{17,\ min}$	Minimum Heating Capacity at 17°F	NEEP
$QH_{5,\ min}$	Minimum Heating Capacity at 5°F	NEEP
$COP_{47,\ min}$	COP at Maximum Heating Capacity at 47°F	NEEP
$COP_{17,\ min}$	COP at Maximum Heating Capacity at 17°F	NEEP
$COP_{5,\ min}$	COP at Maximum Heating Capacity at 5°F	NEEP

[1] As of June 2019, more than 4000 units were included in the database.

ASHP Bin Analysis for Central, Ducted ASHP

The plots in Figure 1 and Figure 2 graphically show the load, capacity and COP trends used in the bin analysis for an example central, ducted heat pump.

Building Load Line is shown on Figure 1 as a black line. The line is defined by the Design Heating Load (in Btu/h from Manual J), or BHL_{des}, which occurs at the design heating temperature (TD). The point where the heating load goes to zero is the balance point temperature (TBP). TBP is assumed to be 57.5°F in this case. The load as a function of temperature is expressed as:

$$BHL(T) = BHL_{des} \times \frac{1}{1.1} \times \frac{(T - TBP)}{(TD - TBP)} \quad \text{where the result is greater than zero} \qquad \text{(eqn 1)}$$

The factor of 1.1 is from the GSHP White Paper (Henderson 2020) and accounts for the fact that Manual J does not include internal gains resulting in design loads that are approximately 10% larger than the "perfectly sized" case.

Maximum Heat Pump Capacity as a function of outdoor temperature, $QH_{max}(T)$, is shown as the red dotted line on Figure 1. The trend is a linear fit between the points defined by $QH_{5,\,max}$, $QH_{17,\,max}$, and $QH_{47,\,max}$ from the NEEP database. The piece-wise linear trend is extrapolated above 47°F and below 5°F. If the rate of change (or slope) below 5°F is less steep than 2% per °F, then the slope for extrapolation below 5°F is set to 2% per °F (to provide physically realistic results). The heat pump sizing ratio is 0.9 in this case, which means that $QH_{5,\,max}$ is 90% of BHL_{des}.

Minimum Heat Pump Capacity as a function of outdoor temperature, $QH_{min}(T)$, is shown as the green dotted line on Figure 1. The trend is a linear fit between the points defined by $QH_{5,\,min}$, $QH_{17,\,min}$, and $QH_{47,\,min}$ from the NEEP database. The piece-wise linear trend is extrapolated above 47°F and below 5°F. If the rate of change (or slope) below 5°F is less steep than 2% per °F, then the slope for extrapolation is set to 2% per °F.

Heat Pump Capacity (rated) is shown as the purple diamonds and line on Figure 1. The trend is a linear fit between the points defined by $QH_{17,\,rated}$ and $QH_{47,\,rated}$ from the AHRI values. The rated data is not used in the bin analysis but is only shown on the plot for reference. In this case the rated capacity matches the maximum capacity.

ASHP Delivered Capacity, as shown by the blue line on Figure 1, is the resulting portion of the heating load that is met by the heat pump. The delivered capacity is calculated as a function of outdoor temperature and used in the bin analysis. For this central, ducted unit, the heat pump's delivered capacity matches the load until the load line reaches the maximum heating capacity line. At temperatures below the intersection point, the ASHP only provides its maximum capacity (and backup heating would be required to meet the rest of the load).

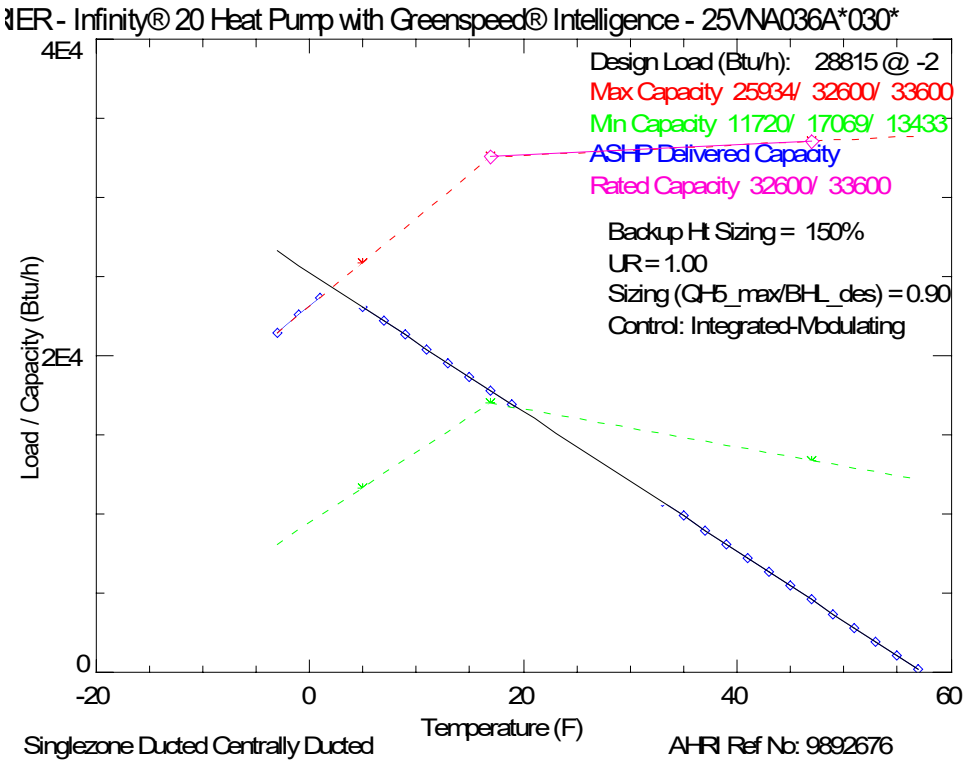

Figure 1. Plot Comparing Building Load Line and Heat Pump Capacity Trends Versus Outdoor Temperature

Heat Pump COP at Maximum Capacity , $COP_{max}(T)$, is shown as the red dotted line on Figure 2. The trend is a linear fit between the points defined by $COP_{5,\,max}$, $COP_{17,\,max}$, and $COP_{47,\,max}$ from the NEEP database. The piece-wise linear trend is extrapolated above 47°F and below 5°F. If the rate of change (or slope) below 5°F is less steep than 2% per °F, then the slope for extrapolation below 5°F is set to 2% per °F. A heat pump generally has lower efficiencies when providing the maximum possible capacity.

Heat Pump COP at Minimum Capacity, $COP_{min}(T)$, is shown as the green dotted line on Figure 2. The trend is a linear fit between the points defined by $COP_{5,\,min}$, $COP_{17,\,min}$, and $COP_{47,\,min}$ from the NEEP database. The piece-wise linear trend is extrapolated above 47°F and below 5°F. If the rate of change (or slope) below 5°F is less steep than 2% per °F, then the slope for extrapolation below 5°F is set to 2% per °F. A heat pump generally has higher efficiencies when providing the minimum possible capacity. At 5°F the minimum and maximum COPs uncharacteristically invert at temperatures below approximately 9°F, for this particular ASHP unit.

Heat Pump Rated COP is shown as the purple diamonds and line on Figure 2. The trend is a linear fit between the points defined by $COP_{17,\,rated}$ and $COP_{47,\,rated}$. The rated data was not used in the bin analysis but is only shown on the plot for reference.

HSPF converted to dimensionless units[2] is also shown on Figure 2 as the horizontal dotted line. It is shown for reference.

[2] That is, HSPF (in units of Btu/Wh) divided by 3.412 provides a dimensionless value that is equivalent to COP.

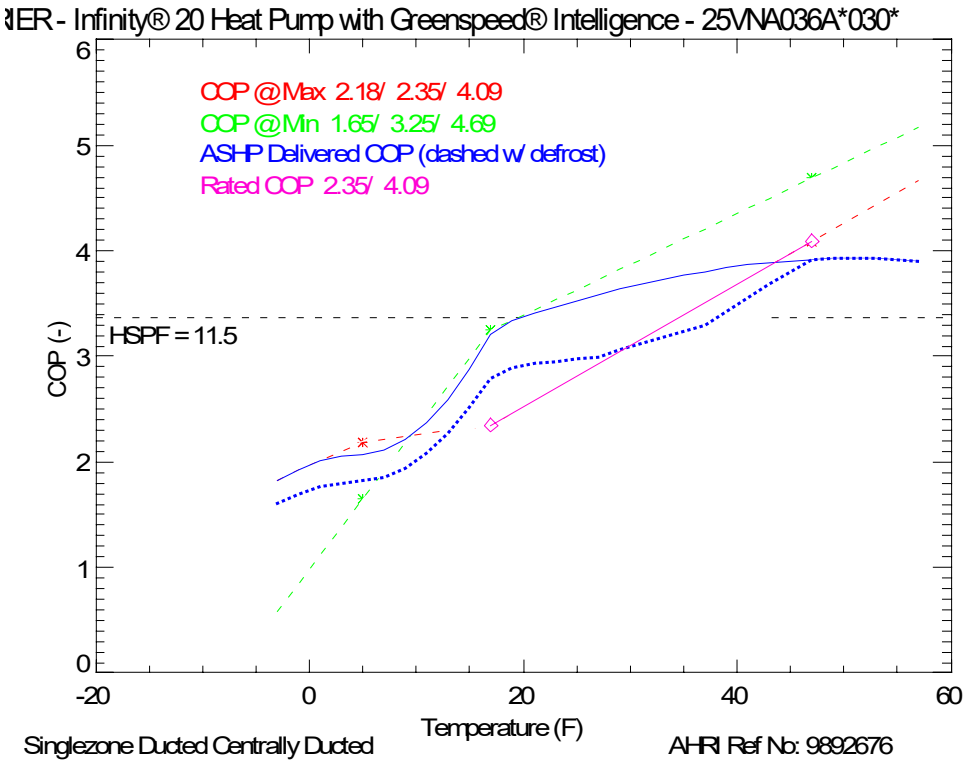

Figure 2. Plot of Heating COP Trends Versus Outdoor Temperature

ASHP Delivered COP, is shown by the blue line on Figure 2. It is determined by relating the maximum and minimum capacity of the heat pump to the building load. As the building load imposed on the heat pump transitions from the minimum capacity (at milder temperatures) to the maximum capacity (at the colder temperatures) – as shown on Figure 1 – the delivered COP also transitions from the COPs corresponding to the minimum and maximum lines.

The actual delivered COP is also affected by the impact of on/off cycling when the minimum capacity of the heat pump exceeds the building load as well as by defrost degradation. Each of these degradation effects is discussed below.

Part Load or Cyclic Losses occur when the load drops below the minimum capacity of the heat pump and it must cycle on and off to meet the load[3]. AHRI Standard 210/240 defines the Cyclic Degradation coefficient (C_D) to capture the efficiency reduction from cycling, known as PLF:

$$PLF(T) = 1 - C_D \times \left(1 - QH_{ASHP}(T) / QH_{min}(T) \right) \quad \text{when } QH_{ASHP}(T) < QH_{min}(T) \qquad \text{(eqn 2)}$$

The factor PLF is multiplied by resulting steady state COP to determine the delivered COP, shown in Figure 2 by the solid blue line. The impact of cycling in this case only occurs when the load is lower than

[3] Cyclic losses can also occur when the delivered capacity in the load sharing mode is lower than the minimum capacity. This is discussed for the displacement application below.

the minimum capacity, so actual or delivered COP is degraded at milder temperatures. The C_D is assumed to be 0.25 (the default value from ANSI/AHRI Standard 210/240).

Defrost Degradation is considered in HSPF but not in the $COP_{max}(T)$ or $COP_{min}(T)$ trend lines (nor in the Rated COP values). Therefore, we apply the defrost degradation factor (FDEF) used in a NIST heat pump simulation study (Domanski, Henderson and Pane, 2014). The factor is from data for a conventional Carrier ASHP circa 1999. It is likely that this dimensionless data still applies to modern heat pump equipment.

Outdoor Temp (F)	-20 and below	17	27	37	47 and above
FDEF	0.075	0.085	0.110	0.09	0.00

The bin analysis uses linear interpolation between the points listed above to find the factor at each temperature: FDEF(T). Then the resulting COP after considering defrost (dotted blue line on Figure 2) becomes:

$$COP_{def}(T) = COP(T) \times (1 - K \times FDEF(T)) \quad \text{(eqn 3)}$$

Most central, ducted heat pumps also turn on the electric resistance elements during a defrost cycle. The factor FDEF(T) does not consider the impact of the resistance heating on COP. Therefore, the factor 'K' the equation above is 1.5 for central systems that use resistance elements during defrost[4]. Ductless units that defrost without resistance elements should use K=1.

[4] The 50% increase in the defrost penalty was derived by assuming defrost runs for 5 minutes of each hour and the base COP is approximately 2.5: ([1 COP] x 5 min + [2.5 COP] x 55 min)/60 min = 2.375, which is 5% lower. The 5% degradation due to resistance heat adds approximately 50% to the adjustment factor FDEF at peak condition at 27°F.

The performance trends shown on Figure 1 and Figure 2 are combined with the temperature bin data in Figure 3 to find seasonal average performance. The temperature bin data is developed from TMY3 hourly weather data formed into 2°F bins, excluding hours from June through September. The summations for the bin analysis are shown by the equations below. The results of the bin analysis are reported on Figure 3.

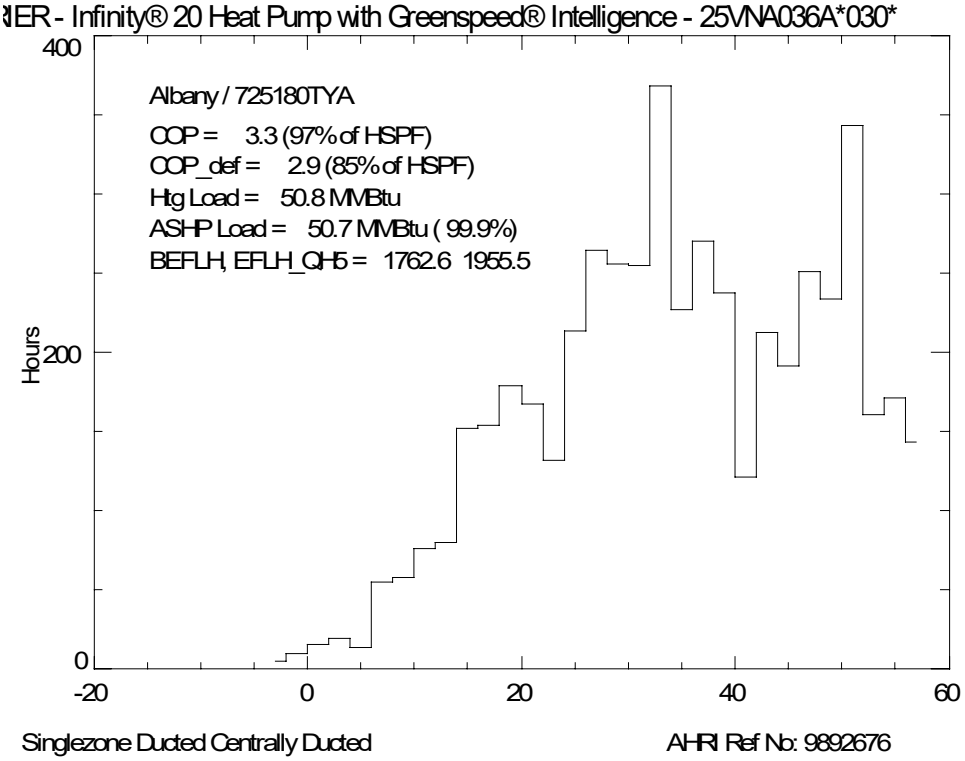

Figure 3. Plot of Outdoor Temperature Data with Seasonal Results Shown

$$Building\ Heat\ Load\ (MMBtu) = \frac{1}{10^6} \sum_j^{Nb} BHL(T_j)\ x\ n_j \qquad \text{(eqn 4)}$$

Where BHL(Tj) is the black line on Figure 1. Nb is the number of temperature bins, n_j is the number of hours in each bin, and T_j is the mid-point temperature for each bin, as shown on Figure 3.

Then using the trends from Figure 1 and Figure 2 described above we can also calculate the seasonal data listed below.

$$ASHP\ Heat\ Load\ (MMBtu) = \frac{1}{10^6} \sum_j^{Nb} QH_{ASHP}(T_j)\ x\ n_j \qquad \text{(eqn 5)}$$

$$ASHP\ Electric\ Use\ (kWh) = \frac{1}{3412} \sum_j^{Nb} \frac{QH_{ASHP}(T_j)}{COP_{def}(T_j)}\ x\ n_j \qquad \text{(eqn 6)}$$

$$Seaonal\ Heating\ COP = \frac{ASHP\ Heat\ Load\ (MMBtu) x\ 1000}{ASHP\ Electric\ Use\ (kWh)\ x\ 3.412} \qquad \text{(eqn 7)}$$

$QH_{ASHP}(T_j)$ is the blue line on Figure 1 and $COP_{def}(T_j)$ is the dotted blue line on Figure 2. These seasonal metrics can also be used to find the equivalent full load hours for the overall building (BEFLH). The EFLH of the ASHP relative to its nominal size ($EFLH_{QH5}$) is discussed further in Appendix D.

$$BEFLH = \frac{Building\ Heat\ Load\ (MMBtu)\ x\ 10^6}{BHL_{des}} \quad \text{(eqn 8)}$$

$$EFLH_{QH5} = \frac{ASHP\ Heat\ Load\ (MMBtu)\ x\ 10^6}{QH_{5,max}} \quad \text{(eqn 9)}$$

The example given by Figure 1, Figure 2, and Figure 3 is for a central, ducted ASHP sized to meet approximately 90% of the design building load ($QH_{5,max}$ is 90% of BHL_{des}). The resulting or predicted seasonal average heating COP is 2.9, which is 85% of the 11.5 HSPF in dimensionless units. The ASHP in this case was able to meet 99.9% of the total annual building load.

Considering Ductless ASHPs in a Displacement Application

The bin analysis also considers the performance of ductless ASHPs that meet only part of the heating load using analysis techniques developed by Bruce Harley Energy Consulting (BHEC) in a memo to NYSERDA dated 10 January 2019. These aspects of ASHP performance are described below.

Utilization Ratio. A key aspect of the displacement modeling approach is the Utilization Ratio[5] (UR), which considers the ability of a ductless indoor section installed in one zone to meet the entire house load. UR determines the portion of the ductless heat pump's heating capacity, $QH_{max}(T)$, that can effectively meet the overall building load BHL(T). When the ratio of BHL(T) to $QH_{max}(T)$ exceeds UR, then the heat pump shares the heating load with the backup heating system (this is not because the heat pump has inadequate capacity, but because the heat is not effectively distributed to all spaces in a typical displacement scenario). When the ratio of BHL(T) to $QH_{max}(T)$ is less than UR, the single zone heat pump is able to serve the entire house load.

Figure 4 shows the case for a ductless ASHP sized to meet only 50% of the total house load ($QH_{5,max}$ /BHL_{des} =0.5) and with UR = 35%. In this case when BHL(T) is less than 35% of $QH_{max}(T)$, then the delivered capacity (blue line) equals the building load. When BHL(T) exceeds 35% of $QH_{max}(T)$, then the ASHP only meets part of the load (blue line) and remainder is met with the backup heating system. When the load is being shared, the load served by the ASHP is defined as:

$$QH_{ASHP}(T) = BHL(T)\ x\ \frac{QH_{max}(T)}{(QB + QH_{max}(T))} \quad \text{(eqn 10)}$$

Where QB is the heating output of the backup heating system. The blue line on Figure 4 below approximately 42°F shows the impact of load sharing for UR = 35% and with separate control. For this example, the backup system capacity (QB) is sized to be 150% of the design heating load (BHL_{des}).

[5] In the BHEC memo from January 2019, this concept is described as the Utilization Limit (UL) which is 1/UR.

Appendix C shows the sensitivity or impact of assuming different values of UR, QB and balance point temperature on the results. The parameter UR has the most significant impact on the results.

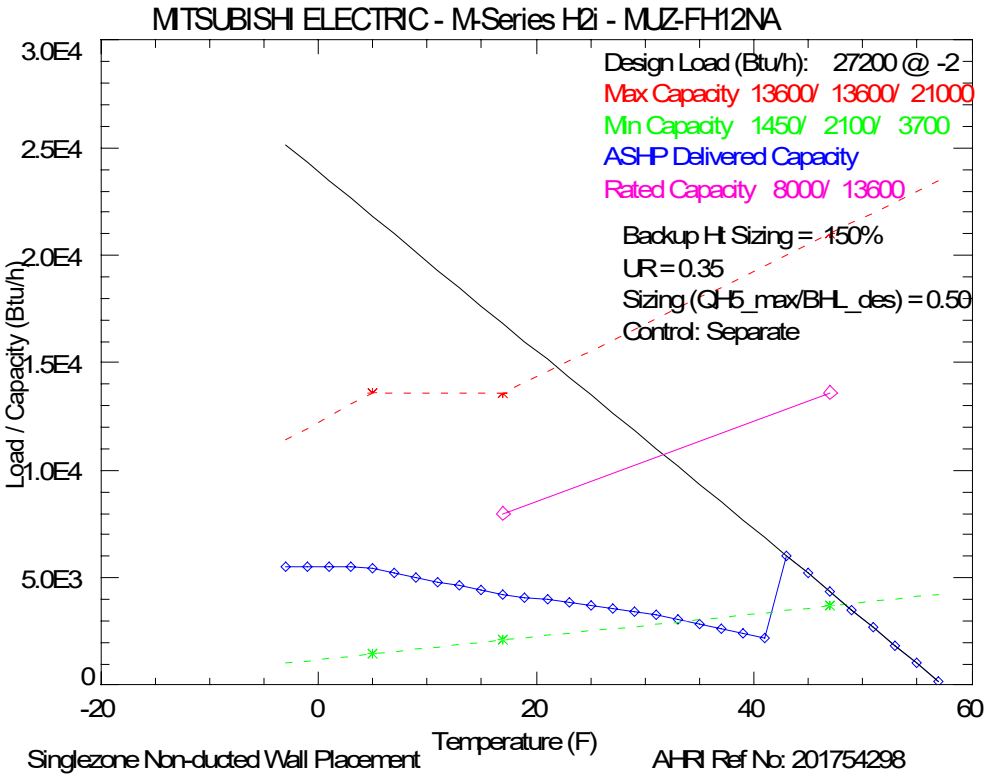

Figure 4. Plot Comparing Building Load Line and Heat Pump Capacity Trends Versus Outdoor Temp – Ductless ASHP

In contrast, the factor UR is 100% for the central, ducted ASHP system shown in Figure 1. In this case, the entire heating capacity of the central, ducted ASHP can serve the building load over the range of outdoor temperatures. Backup heat is only required when the load BHL(T) exceeds the maximum capacity $QH_{max}(T)$.

Separate versus Integrated Controls

The central, ducted system in Figure 1 has integrated thermostatic controls, which physically corresponds to an integrated two-stage thermostat with modulating backup heat. Therefore, the heat pump can meet a larger share of building load when it operates in conjunction with the backup system. In this case the heat pump provides its full capacity as the first stage of heating. Then, the backup heating source is used only when load exceeds the maximum ASHP capacity and space temperatures drop to the lower, second stage setpoint. In this case the share of load met by the ASHP is

$$QH_{ASHP}(T) = MIN(\ BHL(T),\ QH_{max}(T)\ ) \qquad \text{(eqn 11)}$$

In contrast, the ductless ASHP with UR = 35% uses load sharing corresponding to equation 10 (as shown in Figure 4) which corresponds to separate controls, where the ductless ASHP and the existing backup heating system are on separate thermostats. In this analysis we defined four types of possible ASHP control as shown in Table 2.

Table 2. Types of ASHP Control

Type of Control	**Description**	**Method to Determine Load Met by ASHP $QH_{ASHP}(T)$**
Separate	The ASHP and backup heating system are on separate thermostats	Equation 10
Integrated/ Modulating Capacity	The ASHP and backup heating system are on the same thermostat. The backup heater can modulate to meet the load without limiting the ASHP from delivering its maximum capacity. Also called "perfect" control.	Equation 11
Integrated/ Fixed Capacity	The ASHP and backup heating system are on the same thermostat. The backup heater has a fixed capacity to meet the load. The backup heater is larger than the ASHP, so the ASHP is not always able to deliver its maximum capacity (the backup heater supplies a larger share of the load when both are running)	Average of Equations 10 and 11
"Dual Fuel" Either ASHP or Backup Heater	The central ASHP is installed on the same ductwork as the fuel-fired furnace. Once the ASHP can no longer meet the load (or when a changeover temperature is reached). The ASHP shuts off and the furnace meets the load.	Change from ASHP to furnace when the load exceeds the max ASHP capacity (or when changeover temperature is reached, e.g., 15°F).

Figure 5 and Figure 6 show the COP trends, temperature bin data and results corresponding to the ductless ASHP in displacement application from Figure 4. Figure 6 shows the resulting or predicted seasonal average heating COP is 3.1, or 83% of the 12.5 HSPF (in dimensionless units). The ASHP with sizing ratio of 50% in this case can serve 36.2% of the building seasonal heating load.

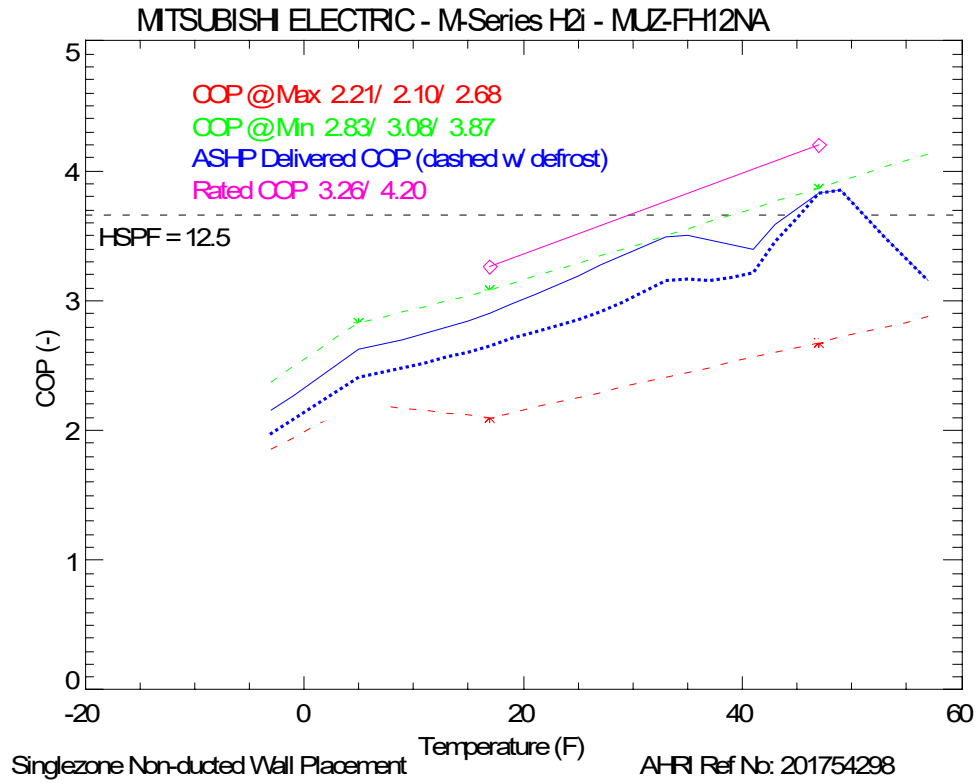

Figure 5. Plot of Heating COP Trends Versus Outdoor Temperature – Ductless ASHP

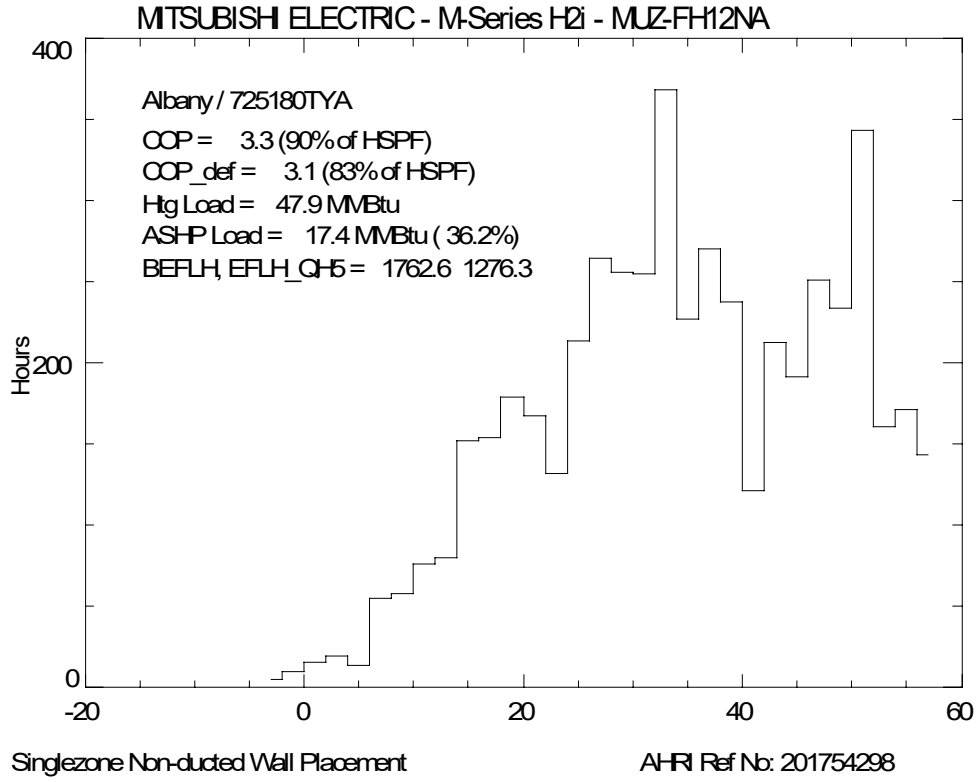

Figure 6. Plot of Outdoor Temperature Data with Seasonal Results Shown – Ductless ASHP

Analysis of Various Heat Pump Scenarios

ASHP units come a variety of equipment configurations:

- Central, ducted ASHPs that are installed like a traditional, central air conditioner or heat pump.
- Single zone ductless mini-split (non-ducted) units with indoor sections that can be wall-mounted, floor-mounted or ceiling-mounted. These units are "single zone" because there is one indoor section corresponding to one outdoor section.
- Multi-zone units with one outdoor section and multiple indoor sections. The indoor sections can be in combinations of ductless, compact ducted or mixed ductless and ducted.
- "Compact" ducted mini-split with limited ducting

All these systems can all be installed as Whole House application where they are sized to meet the entire load. In some Whole House cases, the original heating system might remain in place. Or ASHP units can be installed as a Displacement application, where the original heating system remains in place to serve any heating loads not met by the ASHP unit.

The scenarios listed in Table 3 were considered using the database of 4000+ units available from the NEEP cold climate list. The equipment and sizing and control parameters associated with each scenario correspond to the four most common ASHP configurations we expect to see used in the market. The UR and control parameters are not fully independent, so they are in some cases changed in tandem to represent important aspects of the key scenarios.

Table 3. Scenarios Considered Using ASHP Data from the NEEP Database

Scenario	Description	ASHP Type	Application	Controls	HP Sizing	UR
1a	Central ASHP with different controls	Central, Ducted	Whole	Integrated/ Modulating	0.9	1.0
1b				Integrated/ Fixed capacity		
1c				DF/Either HP or Furnace		
1d				Integrated/ Modulating	1.0	1.0
2a	Ductless Mini-spilt sized at 30%, 50%, 70% and 90/100%	Single-zone Ductless	Displace	Separate	0.3	0.35
2b					0.5	0.35
2c					0.7	0.4
2d				Integrated/ Fixed Capacity	0.3	0.6
2e					0.5	0.6
2f					0.7	0.6
2g			Whole	Integrated/ Modulating	0.9	0.8
2h					1.0	0.8
3a	Ductless MULTI-split sized to 70% and 90/100%	Multi-zone Ductless	Displace	Separate	0.7	0.6
3b			Whole	Integrated/ Modulating	0.9	0.7
3c					1.0	0.7
4a	Compact Ducted Mini-split sized to 50% ,70% and 90/100%	Single-zone Compact Ducted	Displace	Separate	0.5	0.4
4b					0.7	0.4
4c			Whole	Integrated/ Modulating	0.9	0.75
4d					1.0	0.85

Note: Scenario 1a corresponds the system shown in Figures 1 to 3. Scenario 2b corresponds to the system shown in Figures 4 to 6. UR values have been reduced compared to the July 15, 2019 draft of this white paper, to better align savings with results from the Vermont field study of cold climate heat pumps(see Appendix E).

Figure 7 and Figure 8 show the results from conducting the bin analysis described above for Scenario 1a in Albany using appropriate ASHP units from the NEEP database. There were 3583 central, ducted units for Scenario 1a, after applying screening criteria to eliminate units with questionable data[6].

Figure 7 compares the calculated seasonal heating COP to the dimensionless HSPF for each unit in Scenario 1a. The results for the specific ASHP system analyzed in Figures 1 through 3 show up as a single data point on this plot. The HSPF range is also shown on the bottom of the plot for reference. On average, the seasonal COP for all the ASHP units was 87.7% of the dimensionless HSPF. The red line shows the linear best fit to the data.

Figure 8 shows the distribution of the ASHP load fraction for the sample of 3583 central ducted ASHP systems. The average ASHP load fraction was 99.8% with a sizing fraction of 90%.

[6] In each scenario, units were only included in the analysis when the calculated seasonal COP and the dimensionless HSPF was between 1 and 5 and where no key data values from the database were zero. Some of the unit data in the database may be questionable.

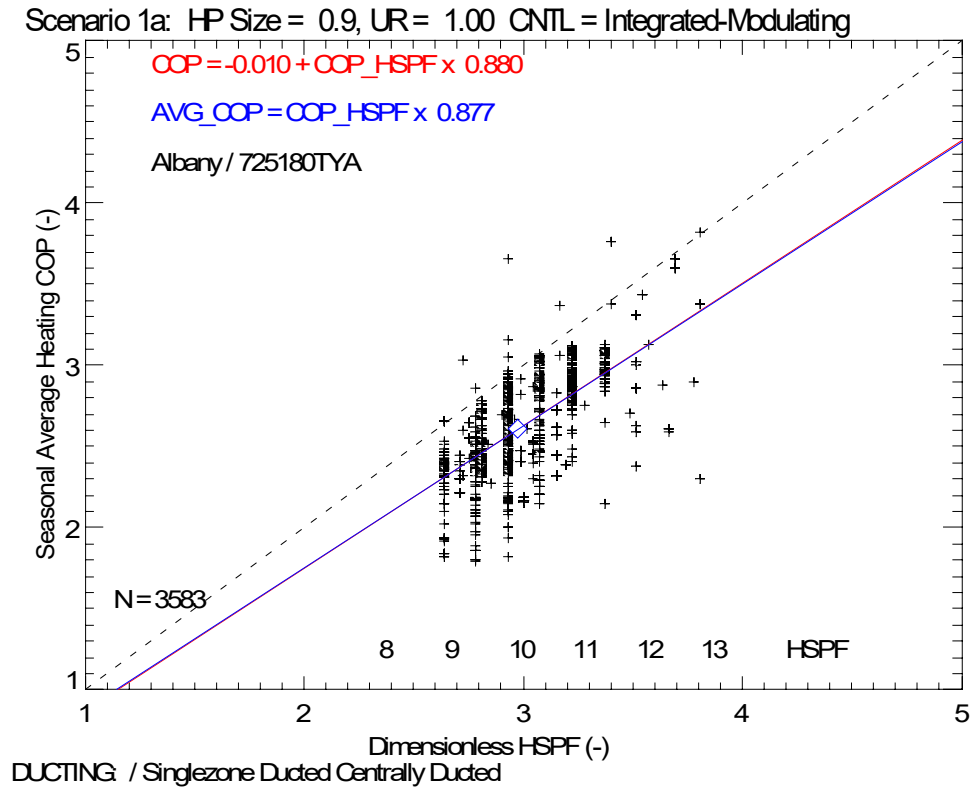

Figure 7. Plot of Seasonal Heating COP compared to Dimensionless HSPF – Scenario 1a: Central ASHP (N=3583)

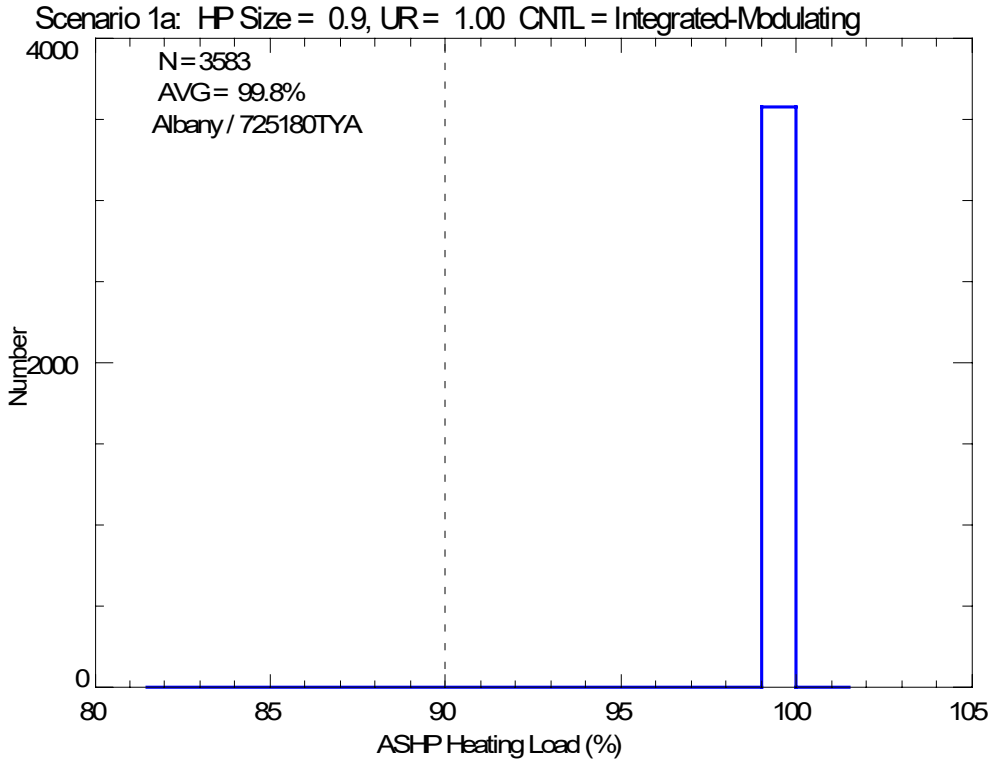

Figure 8. Histogram of ASHP Heating Percentage – Scenario 1a: Central ASHP (N=3583)

The analysis results with Albany weather data for the other scenarios listed in Table 3 are included as similar plots in Appendix A. All these results for Albany are also summarized in Table 4. Results for all the TRM Cities are given in Appendix B.

Table 4. Average Results for Each Scenario in Albany

Scenario	No of ASHPs		Average Ratio of Predicted Efficiency to dimensionless HSPF (%)	Offset (a) and Slope (b) for Best-fit Linear Model COP = a + b*HSPF/3.412 a	b	Fraction of load met by ASHP (%)
1 – Central ASHP	3583	1a	87.7	-0.010	0.880	99.8
		1b	87.8	-0.040	0.891	99.4
		1c	92.5	0.056	0.906	85.9
		1d	86.3	-0.117	0.902	99.9
2 – Ductless Mini-Split ASHP	569	2a	96.0	2.552	0.197	25.0
		2b	95.9	2.351	0.256	38.9
		2c	95.8	2.441	0.228	56.2
		2d	89.6	2.615	0.113	54.4
		2e	89.5	2.574	0.125	74.2
		2f	89.5	2.458	0.160	86.2
		2g	87.5	2.351	0.172	97.9
		2h	87.8	2.336	0.179	99.0
3 – Multi-Split ASHP	284	3a	90.7	-3.334	1.993	71.9
		3b	86.2	-2.683	1.735	95.6
		3c	85.6	-2.823	1.775	97.5
4 – Compact Ducted ASHP	78	4a	103.4	-0.333	1.141	43.0
		4b	102.5	-0.402	1.155	59.0
		4c	95.0	-0.166	1.004	97.3
		4d	94.7	-0.240	1.024	99.2

Note: The results for Scenario 1a in this table are taken from Figure 7 and Figure 8.

Figure 7 shows that the simple factor (or multiplier) to adjust HSPF (the blue line) is very close to the linear model with a slope and offset to adjust HSPF (the red line) for the Central ASHP units. However, the simple factor and the slope-offset method diverge for the other scenarios with different system types, as can be seen by the slope-offset values in Table 4 and the plots in Appendix A. The divergence is especially pronounced for the ductless mini-split units, which show that HSPF is a weak predictor of seasonal average efficiency for these systems[7]. **We therefore recommend that the slope-offset method be used in the TRM to capture these performance differences. We recommend that seasonal efficiency be determined by: COP = a + b*HSPF/3.413.**

[7] These ductless units with higher HSPFs are also expected to be affected more than other unit types by the forthcoming changes to HSPF calculation procedures in AHRI Standard 210/240/Appendix M developed based on laboratory testing by ORNL (Rice et al 2015 and 2016). Similarly, the new loads-based testing standard from the Canadian Standards Association (EXP07:19) has revealed similar discrepancies between HSPF and predicted seasonal efficiency.

Developing Factors for Cooling

The analysis above was also completed for cooling to develop factors to relate seasonal cooling efficiency to SEER and to determine the fraction of the annual cooling load met by the ASHP unit. The bin analysis was similar to the heating analysis with the following exceptions:

- Load Line Differences. The design cooling load (**BCL_{des}**) was assumed to occur at the design cooling temperature for each city. The 10% adjustment in equation 1 was not used for cooling (i.e., the factor of 1.1 is replaced with 1.0). The zero-load cooling balance point was set to 72.5°F. The TMY3 data was binned without excluding any seasons.
- Sizing. The factor defining sizing for cooling operation is: **$AC\_SIZ = QC_{95,max} / BCL_{des}$** However, we made assumptions about the relationship between the Manual J heating load (**BHL_{des}**) and cooling load (**BCL_{des}**) based on climate. The ratio of cooling-to-heating design loads was assumed to be 0.5 for most upstate cities, and then 0.35 in Massena, 0.6 in Poughkeepsie, and 0.75 in NYC (based on the review of several Manual J reports). Appendix B provides more details.
- UR and Control Interactions. The factor of UR was not used for cooling (or UR is assumed to be 1). The integrated control options were not applied for cooling (ie., integrated modulating, or perfect control, was assumed).

The resulting factors for cooling efficiency are of the form: **$EER_{season,ee} = c + d \times SEER$**. Appendix B also lists the proposed factors for cooling corresponding to the scenarios shown above in Table 3. In some cases the scenarios are identical for cooling, so the same coefficients and factors are repeated in the tables (e.g., Scenarios 2a and 2d are the same for cooling). Several Central ASHPs (Scenario 1) in the NEEP database had unrealistically high cooling efficiencies at minimum speed. Therefore, these units were excluded from the analysis used to develop the correlations and coefficients. The cooling results are also given in Appendix A and Appendix B.

Recommendations for the ASHP TRM

We propose to use the results in Table 4 above and from Appendix B to develop factors for the TRM. The TRM calculation process is summarized as:

1. The heating capacity at 5°F ($QH_{5,max}$) for the proposed ASHP will be taken from the NEEP database and divided by the Manual J design heating capacity (BHL_{des}) to determine the HP sizing factor. This sizing ratio as well as other information about the ASHP system type, controls and arrangement will determine which scenario applies.
2. The seasonal heating load will use the design heating load and the appropriate BEFLH values (as described in the GSHP White Paper) to determine the seasonal heating load for the building. Then the fraction of heating load met by the ASHP will be determined by the fractions in Table 4 (or Appendix B) for the various scenarios.
3. The HSPF for the proposed ASHP will be multiplied by the slope and offset in Table 4 (or from Appendix B) to determine the seasonal heating COP.
4. Similar factors for cooling are also provided in Appendix B for use in the TRM.
5. For ASHP displacement applications, it may be possible to use simplifying assumptions so that the building design load (BHL_{des}) and HP sizing do NOT have to be known to determine savings.

Appendix D provides an example for Scenarios 2a, 2b, and 2c where the savings can be determined from the nominal heating capacity of the ASHP using $EFLH_{QH5}$.

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Appendix A

Plots of Results for All ASHP Scenarios with Albany Weather (from Table 3)

(see PDF version of the appendix for plots)

Appendix B

Tables of Factors Required for TRM Calculation Process for ASHPs

Revised December 2019

This appendix summarizes the factors that will be used in the TRM. Factors are given for each scenario and each TRM weather city. The Scenarios are defined by the table below.

Scenario	Description	ASHP Type	Application	Controls	HP Sizing	UR
1a	Central ASHP with different controls	Central, Ducted	Whole	Integrated/ Modulating	0.9	1.0
1b				Integrated/ Fixed capacity		
1c				DF/Either HP or Furnace		
1d				Integrated/ Modulating	1.0	1.0
2a	Ductless Mini-spilt sized at 30%, 50%, 70% and 90/100%	Single-zone Ductless	Displace	Separate	0.3	0.35
2b					0.5	0.35
2c					0.7	0.4
2d				Integrated/ Fixed Capacity	0.3	0.6
2e					0.5	0.6
2f					0.7	0.6
2g			Whole	Integrated/ Modulating	0.9	0.8
2h					1.0	0.8
3a	Ductless MULTI-split sized to 70% and 90/100%	Multi-zone Ductless	Displace	Separate	0.7	0.6
3b			Whole	Integrated/ Modulating	0.9	0.7
3c					1.0	0.7
4a	Compact Ducted Mini-split sized to 50% ,70% and 90/100%	Single-zone Compact Ducted	Displace	Separate	0.5	0.4
4b					0.7	0.4
4c			Whole	Integrated/ Modulating	0.9	0.75
4d					1.0	0.85

Note: Scenario 1a corresponds the system shown in Figures 1 to 3. Scenario 2b corresponds to the system shown in Figures 4 to 6. UR values have been reduced compared to the July 15, 2019 draft of this white paper, to better align savings with results from the Vermont field study of cold climate heat pumps (see Appendix E).

Descriptions in the NEEP Database for each Scenario

Scenario	Name(s) from "Ducting Configuration" field in NEEP Database
Central, Ducted ASHP	Single Zone Centrally Ducted
Ductless Mini-Split	Singlezone Non-ducted Ceiling Placement Singlezone Non-ducted Wall Placement Singlezone Non-ducted Floor Placement
Multi-Split	Multizone All Ducted Multizone All Non-ducted Multizone Mix of Non-ducted and Ducted
Compact ducted	Singlezone Ducted "Compact Ducted"

The HP sizing is determined using the maximum heating capacity at 5°F from the NEEP Cold Climate Heat Pump List and the Manual J design heating load for the home:

HP Sizing = $QH_{5,max}$ / BHL_{des}

Then the AC Sizing is determined by the following process:

BCL_{des} = BHL_{des} x MJ_ratio

Where MJ_ratio = 0.5 for Albany, Binghamton, Buffalo, Syracuse
= 0.35 for Massena
= 0.60 for Poughkeepsie
= 0.75 for New York

MJ_ratio represents the ratio between the Manual J cooling and heating loads for a typical house in each weather city. Then

AC Sizing = $QC_{95,max}$ / BCL_{des}

AC Sizing Fraction

The analysis calculates the AC sizing fraction as a function of the HP Sizing fraction and the assumed ratio of the Manual J cooling and heating loads in each climate (MJ_ratio). Since the relationship between max heating capacity at 5°F ($QH_{5,max}$) and max cooling capacity at 95°F ($QC_{95,max}$) is different for the various types of heat pumps, the AC sizing ratio changes by both scenario and climate. The resulting (average) AC sizing fractions are summarized in the table below. In all but a few cases, the AC system is amply sized relative to the Manual J cooling load (table below) and is therefore able to meet nearly all of the annual cooling load (see the last table on page B-8).

MJ C-to-H Ratio:		0.5	0.5	0.5	0.35	0.75	0.6	0.5
		AC Sizing Fraction (-)						
Scenario	HP Sizing Fraction(-)	Albany	Binghamton	Buffalo	Massena	New York	Poughkeepsie	Syracuse
1a	0.9	3.1	3.1	3.1	4.5	2.1	2.6	3.1
1b	0.9	3.1	3.1	3.1	4.5	2.1	2.6	3.1
1c	0.9	3.1	3.1	3.1	4.5	2.1	2.6	3.1
1d	1	3.5	3.5	3.5	5.0	2.3	2.9	3.5
2a	0.3	0.8	0.8	0.8	1.2	0.5	0.7	0.8
2b	0.5	1.3	1.3	1.3	1.9	0.9	1.1	1.3
2c	0.7	1.8	1.8	1.8	2.6	1.2	1.5	1.8
2d	0.3	0.8	0.8	0.8	1.1	0.5	0.7	0.8
2e	0.5	1.3	1.3	1.3	1.9	0.9	1.1	1.3
2f	0.7	1.8	1.8	1.8	2.6	1.2	1.5	1.8
2g	0.9	2.4	2.4	2.4	3.4	1.6	2.0	2.4
2h	1	2.6	2.6	2.6	3.8	1.7	2.2	2.6
3a	0.7	2.0	2.0	2.0	2.8	1.3	1.7	2.0
3b	0.9	2.6	2.6	2.6	3.7	1.7	2.1	2.6
3c	1	2.8	2.8	2.8	4.1	1.9	2.4	2.8
4a	0.5	1.4	1.4	1.4	2.0	0.9	1.2	1.4
4b	0.7	1.9	1.9	1.9	2.8	1.3	1.6	1.9
4c	0.9	2.5	2.5	2.5	3.5	1.7	2.1	2.5
4d	1	2.8	2.8	2.8	3.9	1.8	2.3	2.8

Heating Efficiency

Method A: the coefficients in the table below are used to find average seasonal heating COP from HSPF:

$$COP_{season,ee} = a + b \times HSPF/3.412$$

	Albany		Binghamton		Buffalo		Massena		New York		Poughkeepsie		Syracuse	
Scenario	a	b	a	b	a	b	a	b	a	b	a	b	a	b
1a	-0.010	0.880	0.113	0.854	0.219	0.842	-0.185	0.838	0.919	0.738	0.333	0.796	0.124	0.860
1b	-0.040	0.891	0.092	0.863	0.197	0.851	-0.168	0.830	0.888	0.750	0.301	0.813	0.093	0.871
1c	0.056	0.906	0.220	0.870	0.310	0.849	-0.057	0.922	0.945	0.741	0.579	0.802	0.183	0.875
1d	-0.117	0.902	-0.003	0.879	0.081	0.876	-0.259	0.847	0.844	0.752	0.237	0.815	-0.007	0.890
2a	2.552	0.197	2.532	0.202	2.645	0.181	2.304	0.204	2.457	0.303	2.620	0.177	2.565	0.195
2b	2.351	0.256	2.419	0.243	2.644	0.188	2.011	0.290	2.453	0.315	2.470	0.228	2.500	0.217
2c	2.441	0.228	2.480	0.224	2.619	0.198	2.358	0.203	2.617	0.271	2.459	0.239	2.550	0.206
2d	2.615	0.113	2.637	0.113	2.702	0.108	2.312	0.150	2.998	0.105	2.725	0.104	2.655	0.107
2e	2.574	0.125	2.601	0.124	2.647	0.122	2.335	0.143	2.896	0.137	2.654	0.119	2.631	0.116
2f	2.458	0.160	2.493	0.155	2.546	0.153	2.322	0.145	2.892	0.139	2.588	0.135	2.513	0.154
2g	2.351	0.172	2.382	0.169	2.440	0.161	2.125	0.178	2.826	0.143	2.424	0.162	2.415	0.163
2h	2.336	0.179	2.386	0.169	2.422	0.168	2.030	0.205	2.827	0.147	2.372	0.177	2.414	0.168
3a	-3.334	1.993	-3.307	1.991	-3.485	2.065	-2.938	1.810	-3.710	2.205	-3.093	1.929	-3.339	2.006
3b	-2.683	1.735	-2.637	1.728	-2.720	1.770	-2.004	1.446	-3.241	2.010	-2.306	1.624	-2.710	1.757
3c	-2.823	1.775	-2.783	1.769	-2.832	1.800	-1.991	1.436	-3.311	2.025	-2.395	1.645	-2.861	1.800
4a	-0.333	1.141	-0.305	1.138	-0.251	1.137	-0.852	1.263	0.373	1.026	-0.407	1.194	-0.308	1.140
4b	-0.402	1.155	-0.393	1.160	-0.300	1.147	-0.865	1.261	0.225	1.057	-0.390	1.175	-0.385	1.160
4c	-0.166	1.004	-0.019	0.963	0.018	0.961	-0.234	0.972	0.282	0.972	0.121	0.926	-0.006	0.962
4d	-0.240	1.024	-0.102	0.983	0.027	0.950	-0.321	0.993	0.224	0.984	0.110	0.920	-0.127	1.001

Method B: an alternative method to find the seasonal average heating efficiency would be to use the simple factors below. This approach is not recommended for use in the TRM.

$$\mathbf{COP_{season,ee} = F_{HSPF,ee} \times HSPF/3.413}$$

	Average Ratio of Actual Efficiency to dimensionless HSPF (-)						
Scenario	Albany	Binghamton	Buffalo	Massena	New York	Poughkeepsie	Syracuse
1a	0.877	0.892	0.915	0.775	1.046	0.908	0.901
1b	0.878	0.894	0.917	0.774	1.048	0.914	0.903
1c	0.925	0.944	0.953	0.903	1.058	0.997	0.937
1d	0.863	0.878	0.903	0.760	1.035	0.895	0.887
2a	0.960	0.960	0.973	0.891	1.036	0.961	0.962
2b	0.959	0.966	0.979	0.891	1.046	0.967	0.965
2c	0.958	0.966	0.981	0.907	1.052	0.973	0.969
2d	0.896	0.902	0.916	0.842	1.002	0.919	0.902
2e	0.895	0.902	0.914	0.842	1.003	0.913	0.903
2f	0.895	0.901	0.914	0.840	1.004	0.910	0.906
2g	0.875	0.882	0.890	0.813	0.988	0.887	0.885
2h	0.878	0.883	0.892	0.812	0.993	0.886	0.890
3a	0.907	0.915	0.931	0.854	0.997	0.922	0.919
3b	0.862	0.870	0.885	0.794	0.955	0.873	0.875
3c	0.856	0.863	0.878	0.788	0.947	0.865	0.868
4a	1.034	1.040	1.056	0.988	1.147	1.063	1.040
4b	1.025	1.033	1.050	0.982	1.130	1.049	1.036
4c	0.950	0.957	0.967	0.896	1.063	0.965	0.961
4d	0.947	0.951	0.959	0.889	1.057	0.955	0.960

Heating Load

Use the factors in the table below ($F_{load,h}$) to determine the fraction of the annual load that is met by the ASHP unit:

	Fraction of Load Met by ASHP (-)						
Scenario	**Albany**	**Binghamton**	**Buffalo**	**Massena**	**New York**	**Poughkeepsie**	**Syracuse**
1a	1.00	1.00	1.00	0.98	1.00	0.97	1.00
1b	0.99	0.99	0.99	0.96	1.00	0.95	0.99
1c	0.86	0.86	0.89	0.69	0.98	0.82	0.90
1d	1.00	1.00	1.00	0.99	1.00	0.98	1.00
2a	0.25	0.25	0.25	0.23	0.28	0.25	0.24
2b	0.39	0.38	0.39	0.37	0.44	0.39	0.38
2c	0.56	0.55	0.56	0.55	0.61	0.54	0.55
2d	0.54	0.53	0.53	0.51	0.59	0.51	0.54
2e	0.74	0.73	0.73	0.71	0.77	0.69	0.74
2f	0.86	0.85	0.85	0.82	0.89	0.81	0.86
2g	0.98	0.97	0.97	0.94	0.98	0.92	0.98
2h	0.99	0.98	0.99	0.95	0.99	0.94	0.99
3a	0.72	0.71	0.70	0.67	0.77	0.67	0.72
3b	0.96	0.94	0.94	0.92	0.96	0.90	0.95
3c	0.97	0.96	0.96	0.94	0.98	0.92	0.98
4a	0.43	0.42	0.43	0.42	0.49	0.43	0.42
4b	0.59	0.58	0.58	0.56	0.64	0.56	0.58
4c	0.97	0.96	0.97	0.94	0.98	0.92	0.98
4d	0.99	0.99	0.99	0.96	0.99	0.95	0.99

Cooling Efficiency

Method A: the coefficients in the table below are used to find average seasonal cooling EER from SEER:

$$EER_{season,ee} = c + d \times SEER$$

	Albany		Binghamton		Buffalo		Massena		New York		Poughkeepsie		Syracuse	
Scenario	c	d	c	d	c	d	c	d	c	d	c	d	c	d
1a	-0.54	0.961	0.92	0.914	-1.37	1.035	-0.39	0.936	0.97	0.889	-0.51	0.965	-0.81	0.988
1b	-0.54	0.961	0.92	0.914	-1.37	1.035	-0.39	0.936	0.97	0.889	-0.51	0.965	-0.81	0.988
1c	-0.54	0.961	0.92	0.914	-1.37	1.035	-0.39	0.936	0.97	0.889	-0.51	0.965	-0.81	0.988
1d	-0.46	0.946	-0.14	0.966	-1.67	1.041	-0.21	0.916	0.46	0.911	-0.59	0.961	-0.89	0.982
2a	12.33	0.118	11.95	0.116	12.24	0.130	12.36	0.135	11.97	0.109	12.10	0.111	12.18	0.123
2b	11.44	0.208	12.21	0.146	11.01	0.229	10.33	0.273	12.27	0.140	12.13	0.167	11.33	0.210
2c	9.81	0.306	10.94	0.232	9.59	0.302	9.16	0.321	11.25	0.220	10.31	0.282	9.74	0.308
2d	12.33	0.118	11.95	0.116	12.24	0.130	12.36	0.135	11.97	0.109	12.10	0.111	12.18	0.123
2e	11.44	0.208	12.21	0.146	11.01	0.229	10.33	0.273	12.27	0.140	12.13	0.167	11.33	0.210
2f	9.81	0.306	10.94	0.232	9.59	0.302	9.16	0.321	11.25	0.220	10.31	0.282	9.74	0.308
2g	9.40	0.314	9.95	0.300	8.13	0.366	8.50	0.351	10.42	0.280	9.08	0.328	9.08	0.327
2h	8.89	0.336	9.44	0.313	7.96	0.375	8.16	0.364	9.87	0.295	9.13	0.327	9.02	0.332
3a	17.76	-0.189	6.68	0.454	10.96	0.199	18.58	-0.218	10.40	0.221	16.24	-0.105	14.46	-0.008
3b	14.53	0.035	14.10	0.009	15.66	-0.060	11.42	0.204	18.38	-0.232	18.50	-0.205	17.26	-0.147
3c	10.85	0.255	17.07	-0.164	14.63	0.017	13.74	0.066	17.53	-0.160	15.27	-0.009	14.14	0.052
4a	-12.94	1.478	-12.22	1.433	-11.55	1.396	-13.56	1.520	-13.18	1.476	-12.69	1.456	-12.77	1.473
4b	-14.16	1.556	-13.06	1.493	-3.40	0.957	-14.37	1.569	-13.43	1.502	-14.01	1.547	-4.69	1.038
4c	-14.37	1.566	-4.31	1.019	-9.60	1.300	-14.65	1.576	-14.25	1.555	-14.68	1.586	-8.45	1.246
4d	-14.44	1.566	-4.69	1.041	-10.17	1.328	-14.70	1.572	-14.37	1.563	-14.79	1.590	-16.48	1.676

Method B: an alternative method to find the seasonal average cooling efficiency would be to use the simple factors below. This approach is not recommended for use in the TRM.

$EER_{season,ee} = F_{SEER,ee} \times SEER$

	Average Ratio of Actual Efficiency to SEER (-)						
Scenario	**Albany**	**Binghamton**	**Buffalo**	**Massena**	**New York**	**Poughkeepsie**	**Syracuse**
1a	0.932	0.962	0.963	0.916	0.939	0.939	0.945
1b	0.932	0.962	0.963	0.916	0.939	0.939	0.945
1c	0.932	0.962	0.963	0.916	0.939	0.939	0.945
1d	0.922	0.958	0.953	0.905	0.936	0.930	0.936
2a	0.669	0.649	0.676	0.687	0.644	0.652	0.666
2b	0.717	0.690	0.718	0.731	0.688	0.708	0.714
2c	0.741	0.718	0.727	0.728	0.721	0.740	0.740
2d	0.669	0.649	0.676	0.687	0.644	0.652	0.666
2e	0.717	0.690	0.718	0.731	0.688	0.708	0.714
2f	0.741	0.718	0.727	0.728	0.721	0.740	0.740
2g	0.733	0.742	0.726	0.730	0.744	0.731	0.730
2h	0.732	0.732	0.728	0.727	0.735	0.734	0.734
3a	0.748	0.811	0.778	0.767	0.774	0.752	0.753
3b	0.807	0.751	0.764	0.812	0.737	0.774	0.763
3c	0.833	0.733	0.792	0.800	0.768	0.801	0.800
4a	0.794	0.788	0.783	0.803	0.781	0.785	0.798
4b	0.807	0.802	0.775	0.809	0.792	0.805	0.787
4c	0.806	0.788	0.789	0.801	0.801	0.809	0.795
4d	0.802	0.790	0.786	0.794	0.802	0.807	0.805

Cooling Load

Use the factors in the table below ($F_{load,c}$) to determine the fraction of the annual cooling load that is met by the ASHP unit:

	Fraction of Cooling Load Met by ASHP (-)						
Scenario	**Albany**	**Binghamton**	**Buffalo**	**Massena**	**New York**	**Poughkeepsie**	**Syracuse**
1a	1.00	1.00	1.00	1.00	1.00	1.00	1.00
1b	1.00	1.00	1.00	1.00	1.00	1.00	1.00
1c	1.00	1.00	1.00	1.00	1.00	1.00	1.00
1d	1.00	1.00	1.00	1.00	1.00	1.00	1.00
2a	0.90	0.79	0.90	0.95	0.81	0.88	0.89
2b	0.99	0.95	0.99	1.00	0.96	0.99	0.99
2c	1.00	0.99	1.00	1.00	0.99	1.00	1.00
2d	0.90	0.79	0.90	0.95	0.81	0.88	0.89
2e	0.99	0.95	0.99	1.00	0.96	0.99	0.99
2f	1.00	0.99	1.00	1.00	0.99	1.00	1.00
2g	1.00	1.00	1.00	1.00	1.00	1.00	1.00
2h	1.00	1.00	1.00	1.00	1.00	1.00	1.00
3a	1.00	0.99	1.00	1.00	0.99	1.00	1.00
3b	1.00	1.00	1.00	1.00	1.00	1.00	1.00
3c	1.00	1.00	1.00	1.00	1.00	1.00	1.00
4a	0.99	0.96	0.99	1.00	0.97	0.99	0.99
4b	1.00	0.99	1.00	1.00	0.99	1.00	1.00
4c	1.00	1.00	1.00	1.00	1.00	1.00	1.00
4d	1.00	1.00	1.00	1.00	1.00	1.00	1.00

APPENDIX C

Sensitivity of Bin Analysis Results to Key Parameters

The example analysis given in Figures 4, 5 and 6 in the White Paper (WP) are for a ductless heat pump sized at 50% of the design load in Albany. This case also assumes the utilization ratio UR of 0.35. Further we assume that the backup or baseline heating system was sized to be 150% of the design heating load. The tables below show the impact of using differing values for these two parameters.

Impact of UR

The UR determines when the ductless ASHP starts to share the load with the backup heating system. The value 0.35 means that when building load reaches 35% of the maximum ASHP capacity, load sharing begins. Figure 4 in the WP shows that this occurs (for the equipment model used in the example) when the outdoor temperature is about 42°F. Table C-1 shows the impact of assuming different values of UR. If UR is zero (i.e., load sharing occurs at all conditions) then the fraction of the heating load met by the ASHP drops from 36.2% to 28.4%, or a decrease of 22% in the amount of load met by the ASHP[8]. Conversely, an increase of UR from 0.35 to 0.7 changes the percent of load met by the ASHP from 36.2 to 56.0%, resulting in 55% more heating load on the ASHP. Note that the seasonal efficiency relative to the HSPF also changes modestly with these changes in UR.

Table C-5. Impact of Changing UR for the Example Given in Figure 4, 5, and 6 in the White Paper

	← Lower Values		**Default[9]**	Higher Values →	
UR	0.0	0.2	**0.35**	0.5	0.7
% of HSPF	79%	81%	**83%**	83%	79%
% of Load Met	28.4%	31.5%	**36.2%**	43.3%	56.0%

Impact of Backup Heater Sizing

The assumed size of the backup heater also has an impact. The default assumption was 150%. Assuming the backup heater was sized to be 100% of the design load, allows the heat pump to meet more of the load when sharing load with the backup heater and increases the percentage of load met by the ASHP from 36.2 to 43.9%, meeting 21% more load. Conversely, assuming the backup heater is twice the design load (200%) reduces the load met by the heat pump from 36.2% to 31.5%, meeting 13% less load. As above, changing this parameter had only a modest impact on the seasonal efficiency.

Table C-6. Impact of Changing the Backup Heater Size for the Example Given in Figure 4, 5, and 6 in the White Paper

	← Lower Values		**default**	Higher Values →	
Backup Heat Sizing	100%	125%	**150%**	175%	200%
% of HSPF	81%	82%	**83%**	84%	84%
% of Load Met	43.9%	39.5%	**36.2%**	33.6%	31.5%

[8] An analysis from Navigant (Consolidated Edison's consultant) presented in September 2019 advocated for using a calculation approach equivalent to setting UR=0.

[9] Default used for Scenarios 2a and 2b and used for the example in Figure 4 of the WP.

Impact of Heating Balance Point

The balance point temperature where the heating load goes to zero was set at 57.5°F in the base analysis, which corresponds to average construction vintage. The impact of assuming different balance points corresponding to new and older construction vintage where investigated. When the balance point is changed to 55°F (newer vintage), the heating load gets smaller and the portion of the load met by the ASHP decreases from 36.2 to 35.5%. When the balance point is changed to 60°F (older vintage), the heating load gets larger and the portion of the load met by the ASHP increases from 36.2 to 37.9%. Again, changing this parameter had only a modest impact on the seasonal efficiency.

Table C-7. Impact of Changing the Balance Point Temperature Size for the Example Given in Figure 4, 5, and 6 in the White Paper

	← Lower Values	**default**	Higher Values →
Balance Point Temperature (°F)	55	**57.5**	60
% of HSPF	82%	**83%**	84%
% of Load Met	35.5%	**36.2%**	37.9%
BEFLH	1643.3	**1762.6**	1877.9

APPENDIX D

Applying White Paper Results to Displacement Applications for Mini-Split ASHPs when the Whole-House Heating Load is Not Known

The ASHP white paper calculation procedure determines savings based on the Manual J heating load (BHL_{des}) for the entire house. However, in some displacement applications, ASHP units may be installed in situations where the total house heating load is not known. Therefore, the exact HP sizing ($F_{HP\text{-}siz}$) will also be unknown. The analysis below shows how the results from the scenarios in the White Paper for ASHP mini-split units with separate controls (scenarios 2a, 2b, and 2c) could be used to develop simpler estimates of savings that would require less information and ultimately be used to estimate savings by only knowing the nominal capacity of the heat pump (QH_{5_max}).

Finding the EFLH based on Nominal Heating Capacity

The calculations below derive the equivalent full load hours based on the nominal heating capacity of the heat pump ($EFLH_{QH5}$). This simpler factor could be useful in instances where load details about the house may not be known.

By definition: $EFLH_{QH5} = BHL_{des} \times BEFLH \times F_{load} / QH_{5\_max}$

In the ASHP white paper we defined: $F_{HP\text{-}siz} = QH_{5\_max} / BHL_{des}$

Therefore: $EFLH_{QH5} = BHL_{des} \times BEFLH \times F_{load} / (BHL_{des} \times F_{HP\text{-}siz})$

Which simplifies to: $\mathbf{EFLH_{QH5} = BEFLH \times F_{load} / F_{HP\text{-}siz}}$

Below in Table D-1 we apply this equation to the BEFLH values from the White Paper for the average vintage. The result is $EFLH_{QH5}$ values range from 1370 to 1469 hours in Albany as the heat pump sizing ratio changes from 0.3 to 0.7 (30 to 70%). This demonstrates that $EFLH_{QH5}$ does not strongly depend on knowing the actual sizing ratio in this case, even though the sizing ratio varies quite widely. The average value of 1418 across the range of sizing in the scenarios is within 4% of the actual values. The same amount of variation is apparent in the other cities as well (no more than 3 to 4% error), suggesting that the average would be acceptable to use for a wide range of actual sizing ratios.

This implies that it not critical to precisely know the home heating load and sizing details for an ASHP when installed as a displacement application and using separate controls (Scenarios 2a, 2b, and 2c). Simply knowing the nominal capacity of the heat pump (QH_{5_max}) allows a reasonable estimate of the total heating load and therefore the heating savings to be determined, provided that ASHP equipment is intended to be used for heating.

Table D-8. Determining Values of $EFLH_{QH5}$ for Displacement ASHP in Scenarios 2a, 2b and 2c

	Albany	Binghamton	Buffalo	Massena	New York	Poughkeepsie	Syracuse
BEFLH	1,763	1,935	1,952	1,880	1,485	1,912	1,842

		Fraction of Load Met by ASHP (F_{load})						
Scenario	**HP Sizing ($F_{HP\text{-}siz}$)**	**Albany**	**Binghamton**	**Buffalo**	**Massena**	**New York**	**Poughkeepsie**	**Syracuse**
2a	0.3	0.2500	0.2466	0.2464	0.2306	0.2806	0.2473	0.2429
2b	0.5	0.3886	0.3844	0.3890	0.3681	0.4420	0.3876	0.3792
2c	0.7	0.5617	0.5500	0.5589	0.5511	0.6082	0.5407	0.5517

		$EFLH_{QH5}$						
Scenario	**HP Sizing ($F_{HP\text{-}siz}$)**	**Albany**	**Binghamton**	**Buffalo**	**Massena**	**New York**	**Poughkeepsie**	**Syracuse**
2a	0.3	1,469	1,591	1,603	1,445	1,389	1,576	1,491
2b	0.5	1,370	1,488	1,519	1,384	1,313	1,482	1,397
2c	0.7	1,415	1,520	1,559	1,480	1,290	1,477	1,452
	AVERAGE	**1,418**	**1,533**	**1,560**	**1,437**	**1,331**	**1,512**	**1,447**

Appendix E

Comparing Savings from "Evaluation of Cold Climate Heat Pumps in Vermont" to TRM-calculated Savings

In order to compare the results from the VT Study[10] to the TRM-calculated savings, the results from both were converted to be savings per nominal heating ton (i.e., the maximum capacity at 5°F). The sections below do this conversion to both the VT Study and the TRM calculations so they can be compared on an equal footing.

Using VT Study Results

The key results from the VT Study are summarized in the table below. All of the VT Study results were aggregated to an average across all of the study participants. Differentiating heating savings by existing heating fuel, equipment type, or other parameters is thus not possible. Although savings is allocated to electricity, oil, propane and wood (the four existing heating fuels before the heat pump installations), it is done on a "fleet average" basis.

Table E-9. Summary of Results from the VT Study

Parameter	Value	Source
Average Heating Load on ASHP (fleet avg)	21.4 MMBtu	Table 8, page 21
Average ASHP "Size" (Max capacity at 5F)	16,116 Btu/h or 1.34 tons	Table 6, page 20
Average HSPF for Installed Equipment	11.9 Btu/Wh	page 21
Average ASHP Consumption for Heating (including standby and defrost modes)	2,085 kWh	Table 8, page 21
Average Seasonal Heating Efficiency	10.7 Btu/Wh or 3.14	Page 21

To calculate savings from the VT Study using an electric resistance heat baseline:

Electric resistance heat consumption:	21.4 MMBtu x 1000 / 3.412 =	6,272 kWh
Heat Pump consumption:	=	2,085 kWh
Savings:	6,272 – 2,085 =	4,187 kWh
Savings per heating ton:	4,187 / 1.34 =	**3,124 kWh per ton**

Similarly, the fuel savings can be determined by assuming a fuel efficiency of 80.5%[11]:

Fuel Savings:	21.4 MMBtu x 10 / 0.805 = 265.8 therms
Fuel Savings per heating ton:	265.8 therms / 1.34 tons = **198.4 therms per ton**

[10] *Evaluation of Cold Climate Heat Pumps in Vermont*, 2017, Walczyk, J. This Cadmus Group report can be found at http://publicservice.vermont.gov/sites/dps/files/documents/Energy_Efficiency/Reports/Evaluation%20of%20Cold%20Climate%20Heat%20Pumps%20in%20Vermont.pdf

[11] Per NYSERDA baseline for oil boilers.

Using Proposed TRM Calculations

To show the range of results from the TRM, we used weather data for Massena. While Massena is the NY climate closest to Burlington, the number of heating degree days in Massena (base 60) are actually 9% greater. Furthermore, most of the HPs in the VT study were in parts of the state with more moderate weather conditions than Burlington. The TRM equations for savings are shown below.

$$\Delta kWh = \Delta kWh_{cooling} + \Delta kWh_{heating} - kWh_{heating,supplemental,ee}$$

$$\Delta kWh = units \times \left(\left(BCL \times \frac{1}{1{,}000} \times \left(\frac{F_{CEC}}{SEER_{baseline}} - \frac{1}{EER_{season,ee}} \right) \times BEFLH_{cooling} \times F_{load,cooling} \right) + \left(BHL \times \frac{1}{1{,}000} \times \left(\frac{F_{EH}}{COP_{season,baseline}} - \frac{1}{COP_{season,ee}} \right) \times \frac{1}{3.412} \times BEFLH_{heating} \times F_{load,heating} \right) - \left(BHL \times \frac{1}{1{,}000} \times F_{EH,new} \times \frac{1}{3.412} \times BEFLH_{heating} \times \left(1 - F_{load,heating}\right) \right) \right)$$

The second term in the equation is the only one of concern, so we ignore the cooling part of the equation (first term) and "new" electric resistance auxiliary heat potion (third term) as part of the calculations. F_{EH} (flag for electric heat) is 1, and $COP_{season,baseline}$ is also 1 for resistance electric heat (100% efficiency). $BEFLH_{heating}$ for average vintage is 1880 for Massena. In order to make the calculations on per nominal heating ton basis, we set the design load (BHL) to be 12,000 divided by the sizing factor, which is defined as 0.3 for scenario 2a and 0.5 for scenario 2b. These selected scenarios correspond to "separate" control since the units in VT were installed around 2015, several years before "integrated" control concepts were conceived, developed or applied for mini-split HPs. The other inputs for Massena are given in the table below.

Similarly, the fossil fuel savings in the TRM are calculated using the equation below. In this case F_{FFH} (flag for fossil fuel) is 1.

$$\Delta therms = BHL \times \frac{1}{100{,}000} \times \frac{F_{FFH}}{AFUE_{baseline}} \times BEFLH_{heating} \times F_{load,heating}$$

Table E-10. Inputs and Results for Proposed TRM Calculations for Electric Heat and Fossil Fuel Baseline

City	Scenario	HP Sizing Factor	BHL	a	B	$COP_{season,ee}$	F_{loadh}	ΔkWh Savings per heating ton	Δtherms Fuel Savings per heating ton
Massena	2a	0.3	40,000	2.304	0.204	3.02	0.231	3,403	215.7
Massena	2b	0.5	24,000	2.011	0.29	3.02	0.368	3,256	206.2

Comparing Results

Table E-3 compares the TRM-calculated savings for Massena with Scenarios 2a and 2b to the savings from the VT Study. The savings predicted by the TRM for Massena are typically 4 to 9% greater than was observed in the VT Study. Some of the savings differences between the TRM calcs and the VT study could be due to slight weather differences.

Table E-11. Inputs and Results for Proposed TRM Calculations for Electric Heat and Fossil Fuel Baseline per Nominal Heating ton

City	Scenario	HP Sizing Factor	TRM ΔkWh Savings per ton	VT Study ΔkWh Savings per ton	ΔkWh Difference	TRM Δtherms Savings per ton	VT Study Δtherms Savings per ton	Δtherms Difference
Massena	2a	0.3	3,403	3,124	+9%	215.7	198.4	+9%
Massena	2b	0.5	3,256		+4%	206.2		+4%

The impact of Climate on TRM Results

One key question is how well the results from the VT study will apply to other milder climates in NY. The TRM calculations were repeated for NYC, where the *BEFLH* for average vintage is 1485. Table E-4 compares the TRM calculation results for Massena and NYC. The ASHP bin-analysis calculations using different weather data already take into account that the heating loads in NYC occur at warmer temperatures, so the resulting seasonal average COP is higher (3.5 verses 3.0). The annual heating load per installed ton is 26% smaller in NYC[12], but the HP is able to serve a larger fraction of the smaller load, (as shown by $\mathbf{F_{loadh}}$). Therefore, the annual heating load served by the HP in NYC per installed ton is just 4-5% smaller than in Massena. Fossil fuel savings show a comparable reduction. Electric savings compared to resistance heat are actually 2-3% higher in NYC than in Massena, due to the higher seasonal COP values for the HP in NYC.

Table E-4. Comparing TRM Calculations for Electric Heat and Fossil Fuel Baseline per Nominal Heating ton in Massena and NYC

City	Scenario	HP Sizing Factor	BHL	$COP_{season,ee}$	BEFLH	F_{loadh}	MMBtu Load on ASHP per heating ton	ΔkWh Savings per heating ton	Δtherms Fuel Savings per heating ton
Massena	2a	0.3	40,000	3.02	1880	0.231	17.4	3,403	215.7
Massena	2b	0.5	24,000	3.02		0.368	16.6	3,256	206.2
NYC	2a	0.3	40,000	3.51	1485	0.281	16.7	3,500	207.3
NYC	2b	0.5	24,000	3.55		0.442	15.8	3,317	195.6

[12] The annual heating load for the same house would be much larger than 26% in Massena compared to NYC. A larger HP would be needed to meet the larger Manual J load at the lower design condition.

Appendix F

Why is the SEER Adjustment so Large for Some ccASHP Units?

The ductless and multi-split ASHPs (Scenarios 2 and 3 in the while paper) tend to have very high SEERs (e.g., 23 Btu/Wh) that get adjusted down by the bin analysis using the minimum and maximum cooling performance data from the NEEP database.

For instance, using the white paper factors for the multi-split ductless ASHP unit (SEER=23) sized to meet 100% of the load in Binghamton (scenario 3c: c= 17.07 and d= -0.164) has the following seasonal cooling efficiency is calculated

Scenario 3c: $EER_{season,ee}$ = 17.07 - 0.164 * 23 = 13.3 (58% of SEER)

In contrast, the white paper factors for the ducted ASHP with a SEER of 18 sized to meet 100% of the load in Binghamton (scenario 1d: c= -0.14, d=0.996) become

Scenario 1d: $EER_{season,ee}$ = -0.14 + 0.966 * 18 = 17.25 (96% of SEER)

The divergence between rated seasonal cooling efficiency (SEER) and $EER_{season,ee}$ is especially pronounced for the ductless mini-split and multi-split units (Scenarios 2 and 3). The calculations above show that the SEER is a poor or weak predictor of the seasonal average efficiency ($EER_{season,ee}$) for these systems. In contrast, the variable-speed, central ducted units have much closer agreement between the seasonal average efficiency and the rated SEER. In fact, the bin analysis predicts that the 18 SEER central ducted unit has a higher seasonal cooling efficiency than the 23 SEER multi-split.

For conventional, single speed ASHPs the average seasonal cooling efficiency is essentially the same as the SEER.

This issue of especially high SEER for some units has been investigated by laboratory testing by ORNL (Rice et al 2015 and 2016[13]). As a result of the ORNL work, the US Department of Energy has implemented forthcoming changes to SEER and HSPF calculation procedures in AHRI Standard 210/240/Appendix M to revise key assumptions in the standard that result in these unexpectedly high rated efficiencies. Anecdotally, the new loads-based testing standard from the Canadian Standards Association (EXP07:19) has demonstrated similar discrepancies between SEER/HSPF and the predicted seasonal efficiency determined by their method.

[13] Rice, C. K., B. Shen, and S. S. Shrestha, 2015. An Analysis of Representative Heating Load Lines for Residential HSPF Ratings. ORNL TM-2015/281. UT-Battelle LLC, Oak Ridge National Laboratory.

Rice, C. K., B. Shen, and S. Shrestha. 2016. Revised Heating Load Line Analysis: Addendum to ORNL/TM-2015/281. ORNL/TM-2016/293. UT-Battelle LLC, Oak Ridge National Laboratory, Oak Ridge, Tennessee. July.

The figures on following pages show the cooling analysis for the same example ASHP units considered in the white paper (using data available from the NEEP database). Figure F-1, Figure F-2 and Figure F-3 show bin analysis for a central ducted unit in the cooling mode. Figure F-1 compares the cooling load lines to the minimum and maximum capacities at each temperature. Figure F-2 shows how the min, max, and delivered efficiencies vary with temperature and Figure F-3 shows the results of the bin analysis used to calculate overall seasonal efficiency. Figure F-4, Figure F-5 and Figure F-6 show the same plots for a ductless mini-split unit.

Similar to results from white paper factors above, the detailed analysis for these specific units show that the seasonal efficiencies are 92% of rated SEER for the Central ducted unit (Scenario 1) and 60% of the rated SEER for the mini-split. Figure F-5 illustrates the issue for the mini-split unit: The dotted line representing the SEER of 26.1 is much higher than the delivered EER or the minimum and maximum EER lines determined from the NEEP data. In contrast, the SEER line on Figure F-2 is in much better alignment with the EER trends based on the NEEP data.

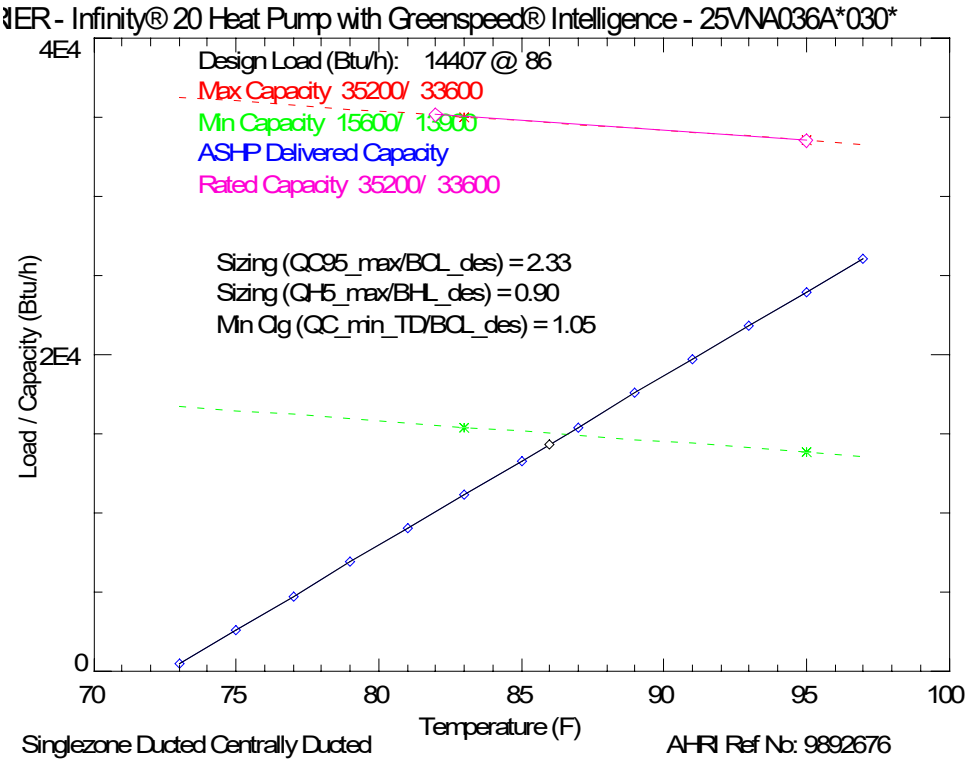

Figure F-9. Plot Comparing Building Load Line and Heat Pump Capacity Trends Versus Outdoor Temp – Central ASHP Cooling

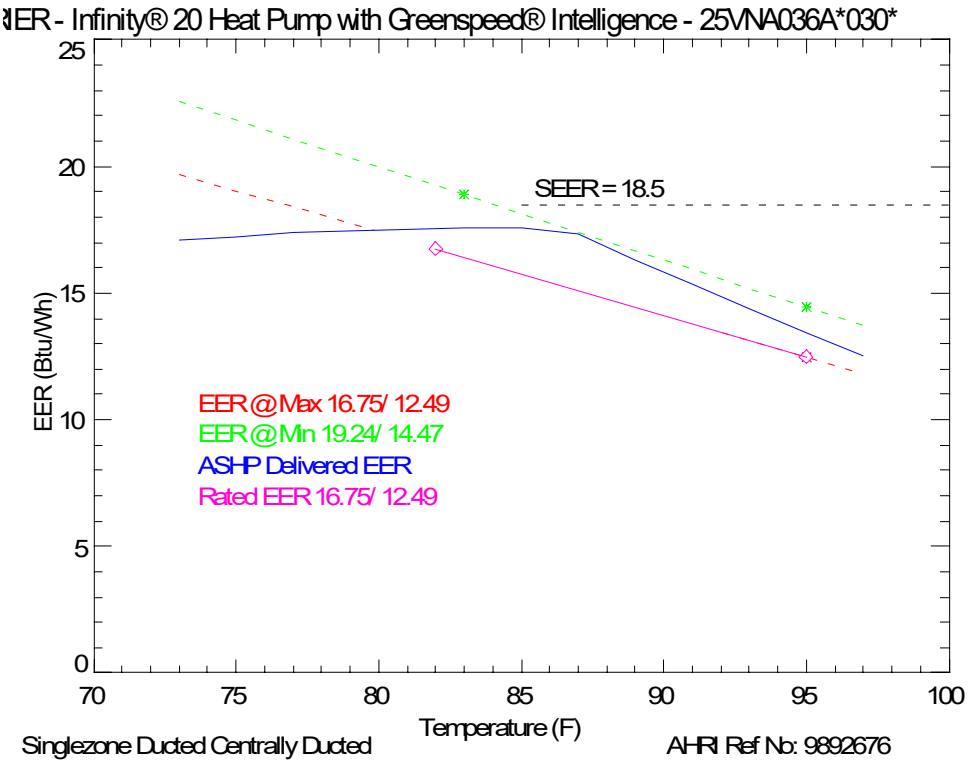

Figure F-10. Plot of Cooling EER Trends Versus Outdoor Temperature – Central ASHP Cooling

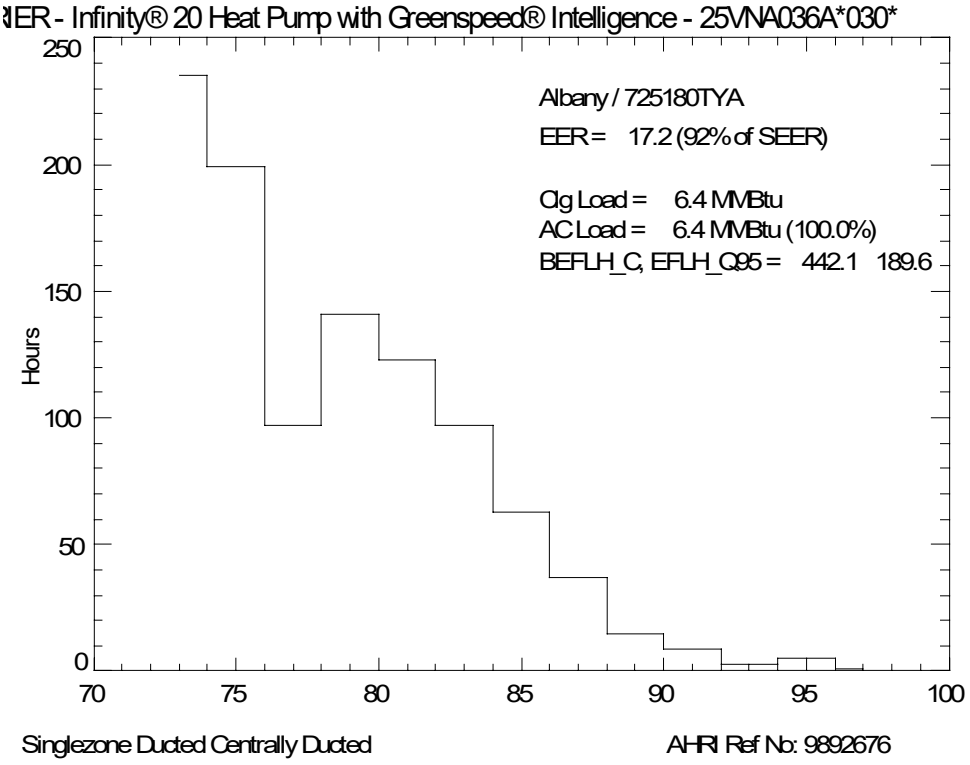

Figure F-11. Plot of Outdoor Temperature Data with Seasonal Results Shown – Central ASHP Cooling

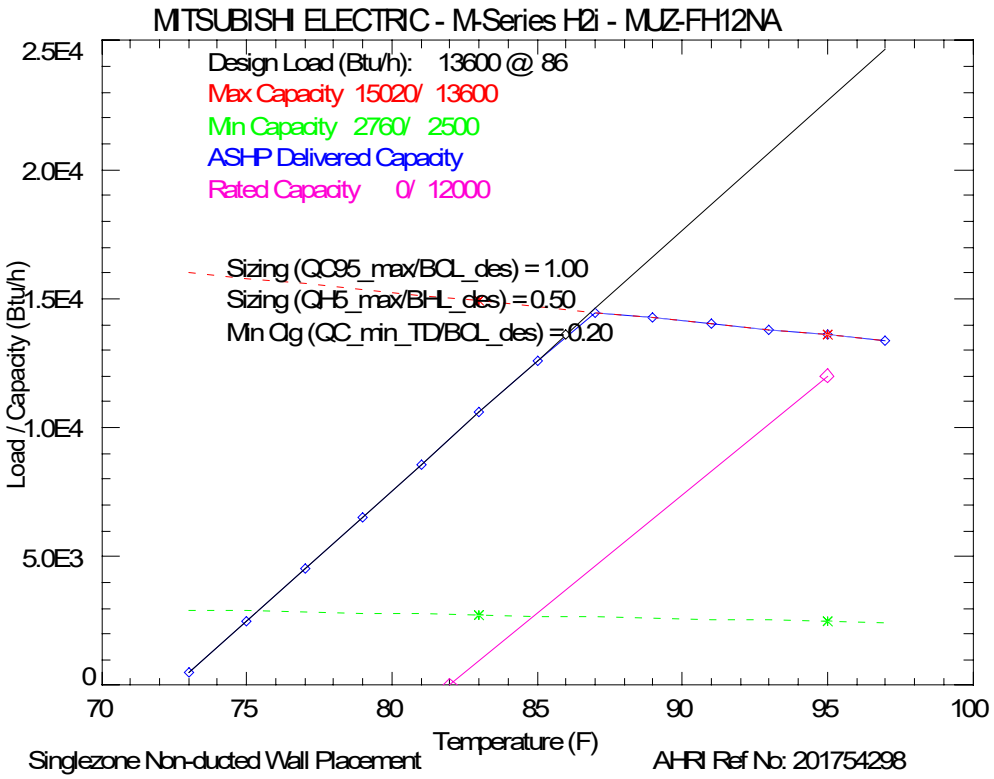

Figure F-12. Plot Comparing Building Load Line and Heat Pump Capacity Trends Versus Outdoor Temp – Ductless ASHP Cooling

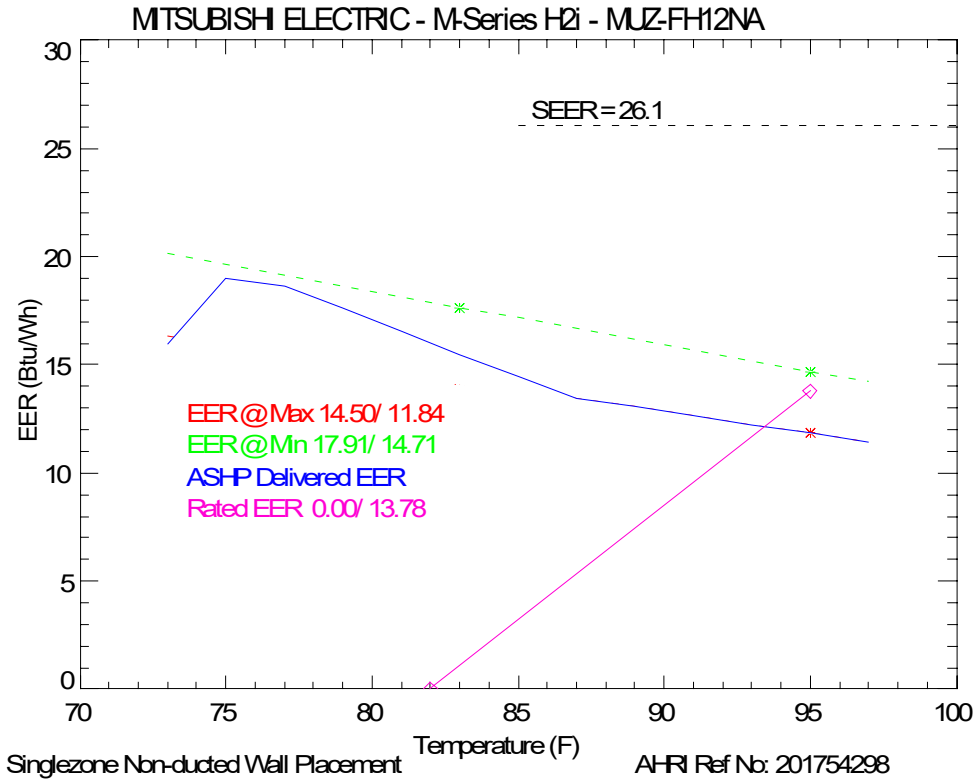

Figure F-13. Plot of Cooling EER Trends Versus Outdoor Temperature – Ductless ASHP Cooling

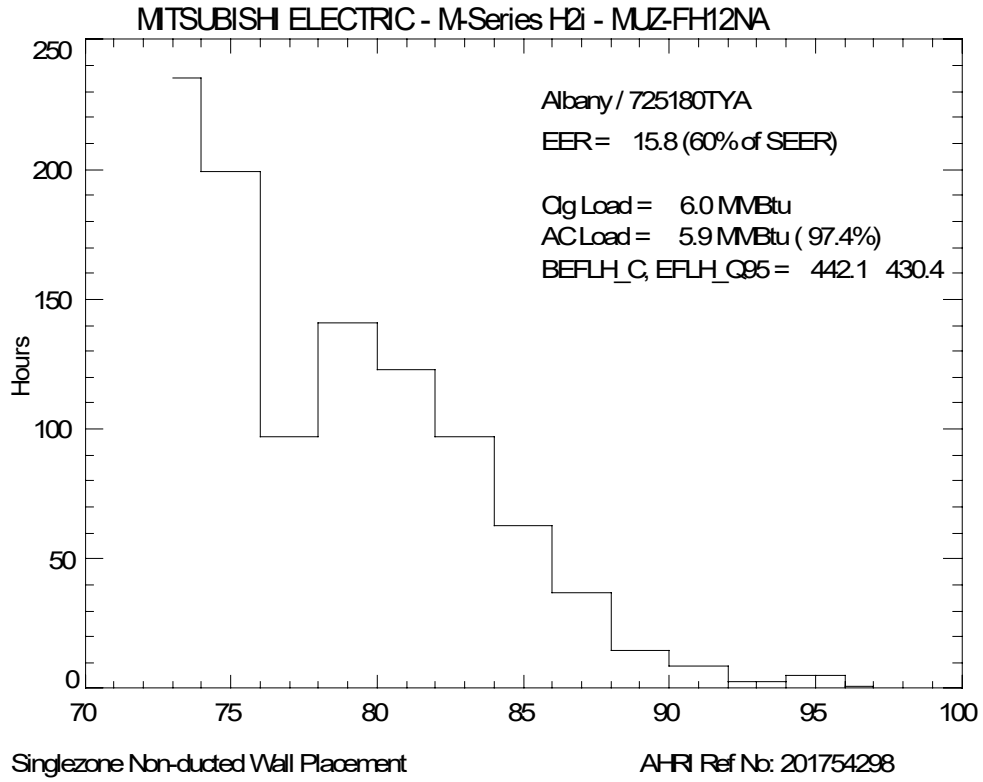

Figure F-14. Plot of Outdoor Temperature Data with Seasonal Results Shown – Ductless ASHP Cooling

Figure F-7 and F-8 show the correlations developed using results for all the units in the NEEP database for each type of unit for the Albany weather data. Again the predicted seasonal efficiency for central ASHPs is closely correlated with SEER. The trend for mini-split ASHPs are very scattered.

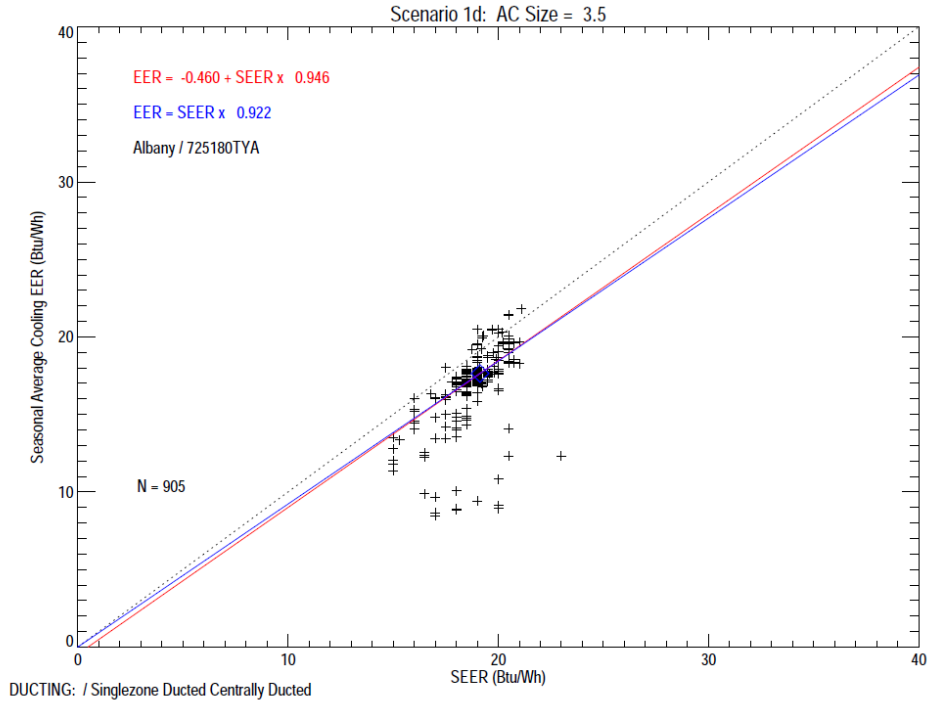

Figure F-15. Correlation Between Rated SEER and Calculated Seasonal Cooling Efficiency – Central Ducted Units

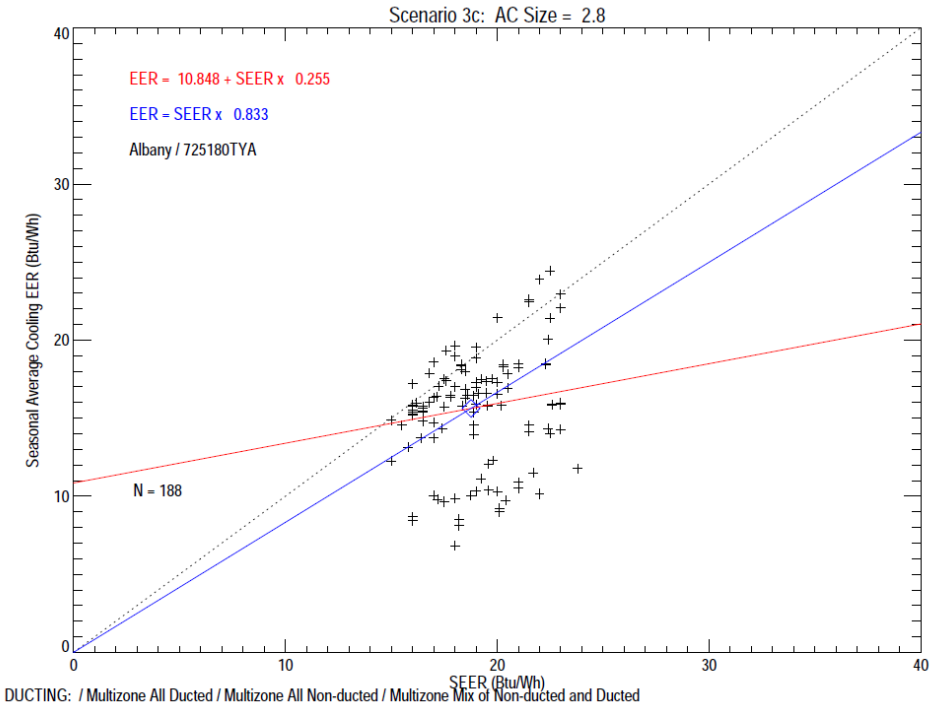

Figure F-16. Correlation Between Rated SEER and Calculated Seasonal Cooling Efficiency – Ductless Mini-split Units

From reviewing data from the NEEP, we identified several characteristics of mini-split and multi-split ASHPs that seem to be related to their high SEERs (and to lessor extend high HSPFs). Central ducted ASHPs generally follow traditional market rules that the rated cooling capacity is the same as the maximum available cooling capacity. This is rationale since a unit's value is based on "size" or the rated cooling capacity. Figure F-9 shows the average ratio of max-to-rated cooling capacity is about 1.0 for central ducted units. However, this ratio is 1.1 to 1.25 for ductless mini-splits and multi-splits.

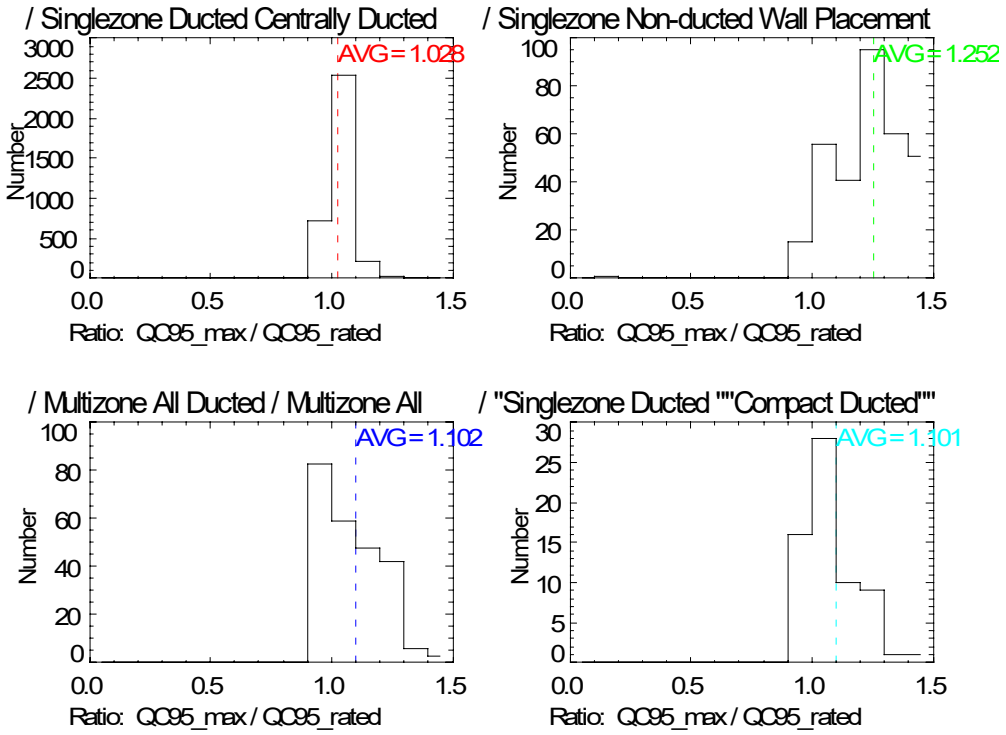

Figure F-17. Distribution of Max-to-Rated Cooling Capacity Ratios Using Data from NEEP Database (Circa July 2019)

Figure F-10 shows that these units are probably de-rated in order to make the maximum heating capacity at 5°F (QH5_max) larger relative to the rated cooling capacity (QC95_rated). For central ducted units the ratio of QH5_max and QC95_rated is about 0.6. However, this ratio is closer to 1.0 for ductless mini-split units.

The de-rating of unit cooling capacity seems to result in the (unrealistically) higher SEER ratings for ductless mini-split and multi-split units.

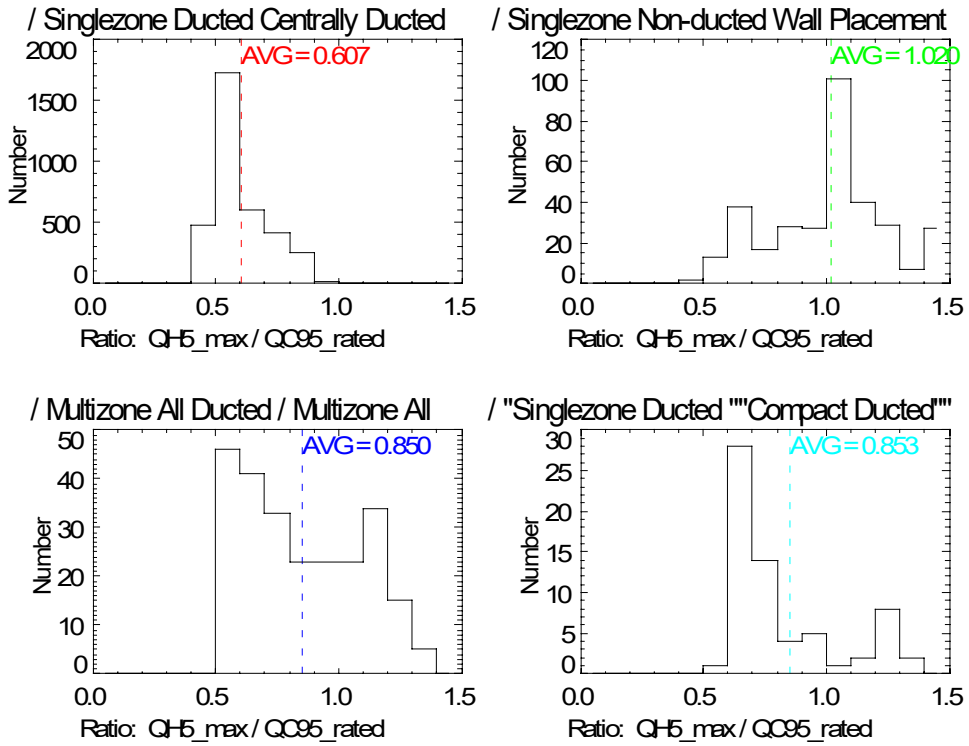

Figure F-18. Distribution of Max Heating-to-Rated-Cooling Capacity Ratios Using Data from NEEP Database (Circa July 2019)