Northeast Energy Efficiency Partnerships

EM&V Forum: Primary Research – Ductless Heat Pumps

April 2014

About NEEP & the Regional EM&V Forum

REGIONAL EVALUATION,
MEASUREMENT & VERIFICATION FORUM

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The Regional Evaluation, Measurement and Verification Forum (EM&V Forum or Forum) is a project facilitated by Northeast Energy Efficiency Partnerships, Inc. (NEEP). The Forum's purpose is to provide a framework for the development and use of common and/or consistent protocols to measure, verify, track, and report energy efficiency and other demand resource savings, costs, and emission impacts to support the role and credibility of these resources in current and emerging energy and environmental policies and markets in the Northeast, New York, and the Mid-Atlantic region.

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ERS was founded in 1995 to provide energy efficiency services and over the past 19 years has developed an exclusive consulting practice focused on energy efficiency and emerging technologies. With offices in Massachusetts, Maine, New York, Texas, California, and Oregon, our staff of more than eighty professionals includes professional engineers, certified energy managers, and LEED-accredited professionals. ERS is involved in many diverse activities, including engineering analysis and modeling of energy efficiency measures; all levels of program evaluation including process and program assessments, impact analysis, and program planning and delivery.

Emerging Technology Program Primary Research – Ductless Heat Pumps

prepared for

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energy & resource solutions

120 Water St., Suite 350
North Andover, MA 01845
(978) 521-2550
Fax: (978) 521-4588
www.ers-inc.com

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Brian McCowan of ERS managed the project, and was the primary researcher and author of this report. He was assisted by ERS Senior Engineer Frank Arena, ERS Engineers Cameron Kinney and Michael Wacker, and ERS Editor Lisa Markowski. For the overall Emerging Technology project, additional ERS contributing staff included Todd Winner, Richard Doughty, Sarah Eckstein, and Jill Rogers. Additional research was conducted by Dunsky Energy Consulting, Livingston Energy Innovations, and Opinion Dynamics.

Elizabeth Titus of the EM&V Forum coordinated the project with management and technical assistance from Chris Neme of Energy Futures Group.

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1 Executive Summary

This report presents the results of primary research conducted to better determine the potential for energy savings and efficiency program support for ductless mini-split heat pumps (DHPs) in the residential sector, particularly in heating-dominant climates, and to determine appropriate methodologies for assessing the savings. The primary research conducted is part of a continuing effort to assess several emerging technologies and innovative program approaches by the Regional Evaluation, Monitoring and Verification Forum (EM&V Forum or Forum) managed by the Northeast Energy Efficiency Partnerships (NEEP). It also is informed by and builds on the Northeast/Mid-Atlantic Air Source Heat Pump Market Strategies Report prepared for NEEP (December 2013).

Prior to this primary research, secondary research was conducted for several emerging technologies, one of which was DHPs for residential applications[1]. The goals of the secondary research were to provide performance and savings guidelines allowing the Forum members to develop measures and programs that realize measurable savings, and to identify knowledge gaps that require further study to close.

With the cooperation of the New Hampshire Electric Cooperative (NHEC) and their customers, nine residential DHP installations were monitored. The weather station closest to the monitored installations is in Laconia, NH. According to National Weather Service data, the average annual heating degree days for Laconia for the last 10 years is 7,109, based on a 65°F balance point. Eight of the DHPs monitored are considered to be "cold-climate" systems, capable of delivering 100% of rated capacity at 5°F. [2] One larger DHP was monitored that is capable of delivering 100% of rated capacity at 17°F. Table 1.1 provides summary information regarding the participant homes and DHPs installed. Further details regarding the monitored systems are presented in Section 3, including summary tables 3.1 – 3.4. A tenth site was withdrawn when the homeowner decided against moving forward with the monitoring process. The monitoring

[1] Other technologies investigated include: advanced power strips, heat pump (hybrid) water heaters, set-top boxes for home entertainment, LED lighting, and biomass pellet heating systems. Primary research was also completed on commercial applications for advanced power strips. See http://www.neep.org/emv-forum/forum-products-and-guidelines/index for reports from these efforts.

[2] DOE ENERGY STAR program references NEEP and Vermont efficiency programs in defining cold-climate DHPs as those capable of delivering 100% of rated capacity at 5°F. https://www.energystar.gov/products/specs/sites/products/files/NEEP%20Supplementary%20Comments%202.pdf

period ran from February 2013 to early September 2013, including 4 months of the heating season, the spring shoulder season, and a full cooling season.

The monitoring of the NHEC sites demonstrated that the systems performed very well for both heating and cooling, in most cases, exceeding the expectations of the homeowners. Performance during cold weather periods, including well below 0°F, and energy savings are both impressive, with most homeowners relying on the DHPs as their primary heating and cooling systems.

In addition, ERS is teamed with Navigant Consulting in evaluating DHP installations in the New York City service territory of Consolidated Edison (Con Edison), as part of a multi-program impact evaluation. The twenty-five units monitored as part of this separate evaluation were not cold-climate models, and the conclusion based on the metered data, as well as participant interviews, is that they are used primarily for cooling. The impact evaluation has been completed, but as of the completion of this report, it is under final review by Con Edison. While the results have not been released, some high-level findings noted in this report were made available to compare with the NH metered findings. The final results of the Con Edison research will be included as an appendix to this report when available.

Table 1-1. Participant Site Summary

Site #	Location	Year Built	Total Building Area (ft^2)	Total # DHPs Installed	Monitored DHP	DHP Size (Tons)	Space Served	Room Area (ft^2)	Heating Displaced	Cooling Replaced/Displaced*
1	Plymouth	1950	1,500	1	Fujitsu 30RLX	3	Living room	280	Kerosene & propane space heaters, wood stove	Window A/C
2	Gilmanton	1995	2,000	2	Mitsubishi FE09NA	0.75	Living room	320	2 propane fireplaces	Window A/C (5)
3	Meredith	1995	Apt. 1,500 Total 6,500	2	Mitsubishi FE12NA	1	In-law apt. living room	168	Oil-fired boiler	Window A/C
4	Northfield	2004	1,875	3	Mitsubishi FE12NA	1	Living room	270	pellet stove, propane fireplace, 3 kerosene heaters	Window A/C (2)
5	Tuftonboro	1990s	2,000	2	Mitsubishi FE12NA	1	Kitchen	540	Propane-fired boiler, pellet stove	Window A/C (2); plannedrepair central
6	Alton	2005	2,400	1	Mitsubishi FE18NA	1.5	Kitchen/ great room	1250	Oil-fired boiler	No existing A/C; planned central A/C
7	Holderness	1996	1,600	2	Mitsubishi FE18NA	1.5	Great room	600	Oil-fired boiler	No existing A/C; planned central A/C
8	Sanbornville	1986	2,200	2	Mitsubishi FE18NA	1.5	Living room	800	Oil-fired boiler	Window A/C
9	Campton	1995	4,000	1	Mitsubishi FE12NA	1	Sunroom	600	Oil-fired boiler	No existing A/C; no planned A/C

1.1 Project Goals

The focus of this primary research is to assist in closing the knowledge gaps associated with DHPs that were identified during the first phase of this project, and to further refine the recommendations for establishing savings calculation methodologies. The goals are summarized as:

- **Energy performance** – Estimate system energy performance through the monitoring of electrical demand and usage, as well as indoor and outdoor temperatures. An additional goal was to estimate coefficient of performance (COP) levels at various climate conditions.

- **Cold weather performance** – The recent and current DHPs designed for residential heating demonstrate great promise for heating in cold climates. This study proposed to determine if the systems are efficiently delivering adequate heat during periods of low outside air temperatures (OATs) and meeting homeowner expectations for cold-weather heating.
- **Potential cooling season load building** – There is a concern that in heating-dominant climates, DHPs purchased for heating will also be used to cool spaces that were not previously cooled, and for which there were no plans for cooling, thereby building summer peak loads. By determining purchase motivators and cooling performance, we sought to identify the potential for cooling season load building, and/or savings.
- **Load shape** – For the peak hours (1:00–5:00 p.m. during non-holiday weekdays June – August) identified by the New England Independent System Operator (NE-ISO), the study sought to determine the average load shape of the monitored systems.
- **Identify user operational procedures** – Unlike central heating and cooling systems which are typically controlled by automatic thermostats, DHPs offer the user the ability to control the units with a handheld remote control that offers many different operational modes and adjustments. A study goal is to determine typical usage of this feature and how this usage interacts with other space conditioning systems installed.
- **Understand purchasing decisions** – Program administrators have a particular interest in learning the motivations associated with the purchase of energy efficient equipment. In the case of DHPs, whether purchases/installations are driven by a desire to heat, cool, or both is important, as is the decision to replace conventional systems or displace a portion of the heating/cooling they contribute to the home.
- **Replacement and displacement of conventional systems** – Directly related to the above is the actual replacement and/or displacement of conventional systems/fuels after initial operation of the DHP; i.e., is the DHP operated as originally intended, or do operators make adjustments following their initial experience with the systems.
- **Comfort levels** – Fan-forced heating is known to introduce discomfort for occupants if temperate air is directed onto skin. Because the DHPs we monitored deliver air from a single fan unit at a variety of airflow rates and temperatures, we sought to learn if the study participants had experienced such discomforts, and/or had made adjustments to the systems or their operation for comfort reasons.
- **EM&V methodologies** – Phase 1 of this project proposed algorithms and methodologies for calculating energy savings associated with DHPs. The data collected during this phase was intended to enhance those recommendations.

1.2 Conclusions

All of the stated goals of this project were addressed, with some limitations associated with the difficulty of monitoring DHP performance in occupied homes. Estimated savings associated with multiple baselines (i.e., electric baseboard, oil-fired boiler, gas furnace, and minimum standard DHP heating sources) were calculated using normalized weather data. We were able to identify the ability of the systems to provide heat during periods of extremely cold OATs. In addition, the project has identified a great deal of information regarding decision making related to both purchasing and operating DHPs in heating-dominant climates. Of particular interest is the observation that participant operational usage of the systems evolved following their initial experiences with the systems, as most owners who initially considered their DHPs as supplemental heating systems began to rely on their systems as primary heat sources.

The results detailed in this report are summarized in the three sections that follow.

1.2.1 Significant Heating Savings Are Achieved Compared with Electric and Fuel Oil Baselines

The monitoring of heating performance for 4 months of the heating season, and extrapolating weather-normalized performance for an entire heating season demonstrates that the systems are capable of delivering significant energy and cost savings in the New England climate, as shown in the tables below. The estimated savings for the eight cold-climate DHPs, average approximately $832 per heating season compared with an electric resistance heat baseline, and approximately $398 compared with a standard efficiency air-source heat pump (ASHP). Savings associated with an oil heat baseline, which is the actual baseline for a majority of the participant sites, are also significant at an average of $613 per heating season (September 15 – May 31).

Tables 1-2 through 1-4 present the weather-normalized estimated heating season energy usage and savings compared with the three baselines:

- Electric resistance baseboard heat
- An ASHP that meets minimum federal efficiency standards
- An oil-fired boiler with an average system efficiency (includes distribution losses) of 78%

A weighted average savings is also calculated for each baseline at 1 ton of heating (12,000 Btu) to allow for the simple calculation of average savings for different size DHPs.

Potential savings associated with a natural gas baseline were also estimated. Due to the current price of natural gas, the savings are small. The estimated natural gas savings are presented in Section 4.3.5.

For all savings calculations presented, the baseline usage is the amount of fuel (electricity, oil, or natural gas) that would be required to produce the same amount of heat produced by the metered DHP.

Table 1-2. Monitored DHP Normalized Heating Season (Sept 15 – May 31) Usage & Savings Compared with Electric Resistance Baseline

Site #	1*	2	3	4	5	6	7	8	9	Average* (Sites 2-9)
System mfg.	Fujitsu	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	N/A
Model	30RLX	FE09NA	FE12NA	FE12NA	FE12NA	FE18NA	FE18NA	FE18NA	FE12NA	N/A
Heat cap. (Btu/h)	37,500	10,900	13,600	13,600	13,600	21,600	21,600	21,600	13,600	16,263
Rated HSPF	9.5	10.0	10.6	10.6	10.6	10.3	10.3	10.3	10.6	10.4
Adjusted HSPF**	8.55	9.00	9.54	9.54	9.54	9.27	9.27	9.27	9.54	9.4
Avg heating COP	2.51	2.64	2.8	2.8	2.8	2.72	2.72	2.72	2.8	2.8
Baseline electric resistance usage (kWh)	9,226	10,054	7,030	5,531	5,416	10,164	10,630	14,460	11,605	9,361
DHP energy usage (kWh)	3,682	3,812	2,514	1,978	1,937	3,741	3,913	5,323	4,151	3,421
DHP savings (kWh)	5,544	6,242	4,515	3,552	3,478	6,423	6,717	9,137	7,454	5,940
Savings @ $0.14/kWh	$776	$874	$632	$497	$487	$899	$940	$1,279	$1,044	$832
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating (kWh)										**4,502**
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating										**$630**

* Site #1 is not included in the average calculations, as it is not a cold-climate model.

** The HSPF is adjusted by a factor of 0.9 to account for climate conditions for central New Hampshire.

Table 1-3. Monitored DHP Normalized Heating Season (Sept 15 – May 31) Savings Compared with Standard Air-Source Heat Pump Baseline

Site #	1*	2	3	4	5	6	7	8	9	Average*
Baseline ASHP energy usage (kWh)	6,173	6,727	4,703	3,700	3,623	6,801	7,113	9,675	7,765	6,263
DHP energy usage (kWh)	3,682	3,812	2,514	1,978	1,937	3,741	3,913	5,323	4,151	3,421
DHP energy savings (kWh)	2,491	2,915	2,189	1,722	1,686	3,059	3,199	4,352	3,614	2,842
Savings @ $0.14/kWh	$349	$408	$306	$241	$236	$428	$448	$609	$506	$398
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating										**2,154**
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating										**$302**

* Site #1 is not included in the average calculations, as it is not a cold-climate model.

Table 1-4. Monitored DHP Normalized Heating Season (Sept 15 – May 31) Savings Compared with Fuel Oil Baseline

Site #	1*	2	3	4	5	6	7	8	9	Average*
Baseline #2 fuel oil displaced (gallons)	291	317	222	174	171	321	335	456	366	295
Baseline #2 fuel oil cost @ $3.70	$1,077	$1,173	$820	$645	$632	$1,186	$1,241	$1,687	$1,354	$1,092
DHP energy usage (kWh)	3,682	3,812	2,514	1,978	1,937	3,741	3,913	5,323	4,151	3,421
DHP energy usage cost @ $0.14/kWh	$516	$534	$352	$277	$271	$524	$548	$745	$581	$479
Net savings	$561	$640	$468	$368	$361	$662	$693	$942	$773	$613
Weighted average savings per ton (12,000 Btu) of heating										**$465**

* Site #1 is not included in the average calculations as it is not a cold-climate model.

1.2.2 The Systems Perform Well at Extremely Cold Temperatures

All of the systems monitored performed well at cold temperatures, with all but one of the systems producing effective heat well below 0°F. Eight of the nine systems continued producing heat down to their lowest minimum outdoor temperature limit of -18°F, although full output is not maintained when outdoor temperatures are below 5°F. The remaining system is a larger (30,000 Btu/h) DHP (Site #1) that continued to deliver heat down to its minimum operational limit of 0°F.

1.2.3 Additional Conclusions

In addition to potential savings and cold-climate performance, we were able to formulate several other conclusions regarding the performance of DHPs, as follows:

- **Cold weather performance is critical for heating in the study-area climate zone**. Both customer satisfaction and monitored performance were substantially lower for the DHP installed at site #1, which was not among the "cold-climate" systems, but rather a system with a 17°F full output low temperature rating and a 0°F operational limit. The participant recently added electric baseboard to the same area.
- **Published HSPF and COP ratings can be misleading**. Because of the modulating nature of DHPs and variations in climate conditions, the standard rating methods are not necessarily good predictors of field performance. However, applying adjustment factors to the ratings improves the ability to use the ratings for predicting savings. For this study, a factor of 0.9 (10% reduction in the rating) was applied to the published HSPF rating for the monitored DHPs. In contrast, the adjustment factor used by the DOE calculator HeatCalc for standard ASHPs for the same climate region is approximately 0.66 (34% reduction in the rating). Published HSPF ratings are typically based on the climate conditions in "AHRI Zone IV," which extends as far north as coastal southern New England. Details of heat pump rating systems and their adjustment factors are found in Section 4.
- **DHPs typically displaced conventional heating**. No heating systems were uninstalled due to the installation of the DHPs. However, all systems fully or partially displaced heating produced by non-heat pump sources. The fuels and systems displaced included central oil boilers, vented kerosene and propane space heaters, unvented propane space heaters, and biomass pellet stoves. Table 3-3 provides details of displaced heating.
- **DHPs installed for supplemental heating often become a primary heating system with owner experience over time**. The participants progressively tended to rely on the installed DHPs as the primary heating system, as they learned the benefits through experience. Most participants reported an initial intent to utilize the systems to supplement the heat from installed central systems, but after positive experiences they began to rely on the systems as the primary heat source, especially when multiple DHPs had been installed.
- **For cooling, DHPs both replaced and displaced less-efficient air conditioning systems.** Six of the participants replaced standard window-installed air conditioners

(A/Cs). One of those six was also considering the repair of an unused central A/C system prior to installing the DHPs. In addition, two participants had contacted an HVAC dealer requesting a quotation for installing central A/C. The same dealer proposed and installed the DHPs as a heating and cooling alternative to the A/C only central system. One participant installed the DHP for heating and cooling, where there were no existing or proposed cooling systems. Table 3-4 provides details of replaced and displaced cooling systems.

- **Cooling season load-building was not a significant factor for the monitored installations.** In all but one case, the DHPs installed replaced window A/C units, or DHPs were purchased instead of installing central A/C. With the increased efficiency of DHPs it can be concluded that summer peak load-building was not significant, and that even in the central NH climate, some cooling savings were achieved.
- **Average summer load shape is coincident with New England ISO targeted peak periods**. Although the cooling loads in Central New Hampshire are relatively small, the peak demand and the peak savings associated with cooling are coincident with the 1 p.m. to 5 p.m. weekday time periods identified as peak demand periods by the New England ISO. Section 5.2 provides cooling load shape charts and details.
- **Purchase decisions varied, but they were often associated with A/C**. The project participants decided to install DHPs for a variety of reasons, including: replacing fossil fuel space heaters, replacing window A/Cs, supplementing central heating systems, dehumidification, and even experimenting to assess the savings potential. Eight of the nine participants either replaced existing cooling systems or purchased the DHPs as an alternative to a standard cooling system.
- **Participants preferred simple remote control operation.** All participants control their DHPs with handheld remote controls. None have installed the optional wall-mount thermostats. Most users select a heating or cooling setpoint, depending on the season, and select "auto" for the fan speed. No participants reported utilizing automatic set-back features, and if any set-back/set-forward settings are selected it is done manually for specific individual time periods. None of the participants have utilized any of the special heating or cooling settings, such as "economy," available with the remote control, typically expressing that they had not yet seen a need to do so. Some of the participants reported selecting a fan speed rather than utilizing the "auto" setting at certain times, due mostly to sound levels. Dehumidification modes are sometimes selected during cooling season.
- **Comfort levels are high.** With the exception of site #1, participants were universally enthusiastic about the comfort levels achieved. None reported experiencing any negative comfort effects from conditioned air being blown directly on them. This can be attributed at least partly to proper placement and installation of the fan-coil units.
- **Typical usage varies by climate zone and fuel availability**. The monitoring in NH and the New York City area demonstrates that DHPs may be used very differently depending on the climate zone, as well as other factors. The preliminary findings of the Con Edison

impact evaluation include a conclusion that 80% of the savings in the territory are attributable to cooling. This is in direct contrast with the findings of this study, which determined that the great majority of the savings are associated with heating. In addition to climatic differences, the Con Edison program places a promotional emphasis on cooling, and both natural gas and district steam heat are prevalent.

1.3 Recommendations

The following recommendations for evaluation methodologies, program implementation strategies, and further study to close knowledge gaps are in addition to the recommendations made in Phase 1 of this study.

- **EM&V methodologies** – Phase 1 of this project proposed algorithms and methodologies for calculating energy savings associated with DHPs. The report also cautioned about the difficulty of assigning simple deemed values for systems that have highly variable usage and performance patterns based on climate conditions and occupant intervention. The data collected during this phase further reinforces those concerns and informs the following recommendations:
  - **Utilize standard ratings, recognizing the limitations.** SEER and HSPF, the ratings utilized for cooling and heating seasonal performance, respectively, are based on strict operational parameters, under several steady-state laboratory conditions. The AHRI methodology for calculating HSPF includes coefficients for six heating zones, which are differentiated from the climate zones utilized for energy codes. However, the HSPF ratings are typically published only for zone IV, which includes coastal southern New England, New Jersey, Virginia, Kentucky, Kansas, etc. (a map of the zones is presented in Section 4, Figure 4-1). The outside air temperature (OAT) covered by the rating is 17°F –47°F, which is appropriate for that climate zone. When performance for regions north and south of zone IV is predicted, heating performance will be inaccurate.
  - **An accurate savings tool for DHPs is needed**. HeatCalc, a DOE-supported downloadable spreadsheet tool, includes a calculator to adjust published HSPF ratings for the local climate.[3] However, the adjustment factors were formulated prior to the introduction of cold-weather performing heat pumps and these factors assume that electric resistance coils contribute part or all of the heating at colder temperatures. Updating the tool to be consistent with the cold-weather performance of cold-climate DHPs would provide program administrators, as well as market actors, a simple tool for estimating DHP savings.
  - **Utilize performance monitoring and billing analysis to assist in predicting savings**. Standards organizations, as well as the heat pump industry, recognize the limitations of COP, SEER, and HSPF for calculating the performance of continuously

[3] www.eia.gov/neic/experts/heatcalc.xls.

modulating DHPs. Field studies and impact evaluation efforts that include monitoring and/or billing analysis should be used to further inform and adjust savings calculations for DHPs. Impact evaluation sponsors should allocate enough elapsed time for DHPs to be evaluated over a minimum of three seasons. An added benefit of evaluating performance over three or more seasons, or for multiple years, is that it allows for the capture of changes in usage that typically take place as the users gain experience with the systems.

- **Program implementation strategies**. The following recommendations are related to promoting DHPs as components of efficiency program portfolios.
  - **Stay current with DHP advances**. Program administrators should work to stay current with this advancing technology. As this is being written, at least one manufacturer is in the process of introducing yet another increase in efficiency for DHPs, as well as larger and multi-head units that perform at the low temperatures currently reached only by single-head units.
  - **Promote DHPs appropriate for the climate zone.** For heating-dominant climates, program administrators should consider restricting program participation to the installation of systems that will operate at near full-load conditions at the design temperature for the region. Customer disappointment and savings snap-back are likely if support is given to DHPs that perform marginally in the lower ranges of the regional OATs.
  - **Consider DHPs for fuel switching**. In jurisdictions where incentives are allowed, DHPs are excellent candidates for fuel switching from oil heat. Where it is not allowed, DHP performance on a direct fuel cost basis is attractive compared to oil heat. When climate change is considered, replacing fossil fuel systems with DHPs becomes even more attractive. Most climate change studies suggest that replacing fossil fuel systems with efficient electric systems powered by clean generation and/or renewable energy sources is a necessary component of meeting long-term climate goals. A recent European study concluded that "achieving an 80% greenhouse gas (GHG) reduction across the economy will likely require massive electrification of space heating, water heating, and personal transportation while simultaneously de-carbonizing the power sector" (i.e., 95%–100% reliance on renewable, nuclear, and/or fossil fuels with carbon capture and storage).[4] A study of GHG emission reduction options for the state of California reached similar conclusions.[5] However, even before the grid is decarbonized, an efficient DHP would result in fewer carbon emissions than an efficient gas furnace or boiler under many scenarios. For example, consider a DHP

[4] European Climate Foundation, *Roadmap 2050: Practical Guide to a Prosperous, Low-Carbon Europe*, Volume 1, April 2010, p. 6. See www.roadmap2050.eu .

[5] Price, Snuller, Energy and Environmental Economics, "Meeting California's Long-Term Greenhouse Gas Reduction Goals", prepared for Hydrogen Energy International, November 2009.

with a seasonal average COP of 2.7 that receives its power from a 45%-efficient natural gas power plant and a grid with marginal line losses of 10%. The delivered efficiency of the heat – from power plant to home heat – is 110% (0.45 × 0.9 × 2.7). In contrast, even a very efficient gas home heating system (condensing furnace or boiler, coupled with an efficient distribution system) will typically not be more than about 90% efficient. Since both are ultimately using gas, the DHP will produce approximately 20% less carbon emissions.[6]

- **Closing knowledge gaps** – It is hard to close all the knowledge gaps on rapidly advancing technologies. But if efficiency programs are to meet goals, knowledge of products, applications, and usage patterns is critical. Key points include:
  - **Information sharing** – A lot of work is being performed right now on DHP performance. This study focused attention on user operational experiences and equipment performance, but only for one part of the country. The planned EM&V Forum meta-study of DHP research, combined with other efforts, is intended to further increase access to more comprehensive assessment of DHP under various baseline scenarios.
  - **Control options** – Control options for DHPs include both programmable wall-mounted thermostats and hand-held remote controls. As noted, all of the DHPs monitored for this study were controlled by hand-held remote controls. The hand-held remote controls utilize a thermistor (an electrical resistor that varies with temperature) inside the DHP return air stream to monitor room temperature while the remote wall-mounted unit bypasses the built-in thermistor, sensing temperature at the thermostat location. Although it was not analyzed as part of this study, the fact that these control options operate differently poses the question of how performance might be impacted. Comparative studies of DHP performance with the two control types would inform decision-making for program administrators, market actors, and homeowners.
  - **Controls integration** – This study revealed that from nine participants came nine different methodologies for controlling their DHPs in relation to other heating systems. The methods can best be described as "work-arounds." Especially with the advancement of "smart" controls, and Web-accessible thermostats, program implementers and evaluators should work with the industry to identify advantageous methodologies for controlling multiple systems, in order to encourage optimized control of the systems.
  - **Commercial markets** – To date, most of the focus on promoting DHPs, especially recently introduced high efficiency/low temperature models, has been on the residential market. The time is right to increase the penetration of high efficiency DHPs in the small/medium commercial market. In addition to the ability to perform

[6] Neme, Energy Futures Group, correspondence, March 2014.

at high efficiency levels, DHPs are able to: solve difficult heat/cooling zone issues; avoid simultaneous heating and cooling; isolate ventilation and conditioning systems; condition specialty areas such as server rooms; provide variable control for areas of variable occupancy such as conference rooms, etc.

- **Performance ratings** – Replacements for SEER and HSPF may not be available soon. A reasonable goal would be for manufacturers to supply the SEER and HSPF ratings appropriate for the efficiency program territories. Although it is understandable that the industry desires to publish one set of numbers, efficiency programs need performance metrics for the local climate in order to accurately predict savings, and market actors would benefit from improved performance predictors for sizing systems. Although the DOE-supported tool HeatCalc includes a calculator to adjust published HSPF and SEER ratings for the local climate, it too is inaccurate for predicting the heating performance of cold-climate DHPs.

2 DHP MONITORING PROCEDURE

The Project team monitored electric demand and usage, run times, discharge air temperature, and space temperature. Where possible, conventional incumbent heating systems were also monitored to help us understand how the DHPs are being used with the conventional heating systems.

The site work procedures are outlined as follows:

- Record information related to the construction of the home and occupancy patterns.
- Record equipment make and model number for each DHP installation monitored.
- Record the total HVAC configuration for the home: all fuels, all systems.
- Install data logging equipment:
  - DHP electric demand and usage:
    - Install logging power monitor in compressor panel – current transducer (CT) and Hobo downloadable logger recording power and usage.
  - DHP run time:
    - Recorded from same logging power meter as above
  - Delivered air temperature
    - Hobo temperature logger with remote probe. Logger on top of fan unit, hidden from occupant view. Probe located in the delivery airstream.
  - Indoor ambient temperature
    - Hobo temperature monitors. One located in the room with the DHP. Where appropriate, depending on the particular field situation, a second temperature monitor was installed in a second location in order to help identify differences in how spaces are conditioned.
  - Other heating/cooling systems:
    - Temperature logger installed on kerosene space heaters.
    - Central systems – Record run time. Depending on the installation, this was done with a power logger or a magnetic logger that records motor run-time.
    - Biomass – During the interviews, request usage and purchase data from customer.
  - Outside weather conditions were recorded from the nearest (Laconia, NH) weather station.
  - Logger installation homeowner interviews were conducted to determine usage patterns, perceived savings, comfort levels, etc.

- Post monitoring interviews – After reviewing the recorded data, the participants were interviewed a second time to gain a full understanding of how they had been operating the systems during the monitoring period, and to record satisfaction levels associated with both comfort and energy savings.

3 PARTICIPANT SITE INFORMATION

This section outlines the collected information specific to each site, including:

- Building descriptions
- Systems installed
- General heating and cooling usage patterns
- Existing systems fully replaced
- Existing and/or planned systems and/or fuels displaced

The weather station closest to the monitored installations is in Laconia, NH. The average annual heating degree days for Laconia for the last 10 years, according to National Weather Service data, is 7,109 based on a 65°F balance point.

Table 3-1 provides summary information regarding the participant sites, and the spaces that the monitored DHPs serve, and heating and cooling systems that were replaced or displaced. Table 3-2 presents the basic specifications for the monitored units. Product specification sheets are included in Appendix A. Participant survey results are included in Appendix B.

Table 3-1. Participant Site Summary

Site #	Location	Year Built	Total Building Area (ft^2)	Total # DHPs Installed	Monitored DHP	DHP Size (Tons)	Space Served	Room Area (ft^2)	Heating Displaced	Cooling Replaced/Displaced*
1	Plymouth	1950	1,500	1	Fujitsu 30RLX	3	Living room	280	Kerosene & propane space heaters, wood stove	Window A/C
2	Gilmanton	1995	2,000	2	Mitsubishi FE09NA	0.75	Living room	320	2 propane fireplaces	Window A/C (5)
3	Meredith	1995	Apt. 1,500 Total 6,500	2	Mitsubishi FE12NA	1	In-law apt. living room	168	Oil-fired boiler	Window A/C
4	Northfield	2004	1,875	3	Mitsubishi FE12NA	1	Living room	270	pellet stove, propane fireplace, 3 kerosene heaters	Window A/C (2)
5	Tuftonboro	1990s	2,000	2	Mitsubishi FE12NA	1	Kitchen	540	Propane-fired boiler, pellet stove	Window A/C (2); plannedrepair central
6	Alton	2005	2,400	1	Mitsubishi FE18NA	1.5	Kitchen/ great room	1250	Oil-fired boiler	No existing A/C; planned central A/C
7	Holderness	1996	1,600	2	Mitsubishi FE18NA	1.5	Great room	600	Oil-fired boiler	No existing A/C; planned central A/C
8	Sanbornville	1986	2,200	2	Mitsubishi FE18NA	1.5	Living room	800	Oil-fired boiler	Window A/C
9	Campton	1995	4,000	1	Mitsubishi FE12NA	1	Sunroom	600	Oil-fired boiler	No existing A/C; no planned A/C

*Window A/C units replaced is the total replaced by the total number of DHPs installed in the home. Planned A/C installations were recorded during participant surveys.

Table 3-2. Basic Specifications of Monitored DHPs

Site #	MFG	Model	Cooling Capacity (Btu/h)	Heating Capacity (Btu/h)	Published Maximum Amperage	Minimum Operating Temp	Minimum Temperature @ Rated Output	Rated Heating HSPF	Rated Cooling SEER
1	Fujitsu	30RLX*	32400	37500	18.5	0°F	17°F (100%)	9.5	17.5
2	Mitsubishi	MUZ-FE09NA	9000	10900	12.5	-18°F	5°F (100%)	10	26
3	Mitsubishi	MUZ-FE12NA	12000	13600	14.32	-18°F	5°F (92%)	10.6	23
4	Mitsubishi	MUZ-FE12NA	12000	13600	14.32	-18°F	5°F (92%)	10.6	23
5	Mitsubishi	MUZ-FE12NA	12000	13600	14.32	-18°F	5°F (92%)	10.6	23
6	Mitsubishi	MUZ-FE18NA	17200	21600	16.7	-18°F	5°F (100%)	10.3	20.2
7	Mitsubishi	MUZ-FE18NA	17200	21601	16.7	-18°F	5°F (100%)	10.3	20.2
8	Mitsubishi	MUZ-FE18NA	17200	21602	16.7	-18°F	5°F (100%)	10.3	20.2
9	Mitsubishi	MUZ-FE12NA	12000	13600	14.32	-18°F	5°F (92%)	10.6	23

* Monitored Fujitsu DHP is not a cold-climate unit. At the time of the completion of this study, no cold-climate DHPs are available in this size (3-ton) range.

Eight of the sites were served by Mitsubishi FE series DHPs ranging in nominal capacity from 9,000 to 18,000 Btu/h. Site #1 is served by a Fujitsu 30RLX DHP rated at a nominal 37,000 Btu/h heating capacity. The Mitsubishi units are rated to operate at a minimum OAT of -18°F and are capable of providing full heating output at 5°F and above. The Fujitsu unit is rated to operate at a minimum OAT of 0°F and is capable of providing full heating output at 17°F and above. Fujitsu also manufactures DHPs in the size and temperature range of the Mitsubishi DHPs, but none were encountered in the sites we monitored.

3.1 Baseline Heating and Cooling

For all participant sites, DHPs displaced heating that was supplied by fossil fuel and/or biomass systems. For six of the nine participant sites, window A/Cs were replaced or unplugged at the time of DHP installation. Two participants were planning installation of central A/C systems prior to learning of DHPs’ capabilities. One participant was considering the repair of an older inoperable central A/C system. Finally, one participant installed the DHP to “experiment” with heating their sunroom and now heats and cools the space. Tables 3-3 and 3-4 present baseline heating and cooling information. More details are presented in Appendix B.

Table 3-3. Baseline Heating Systems/Fuels Displaced

Site #	Heating Displaced	Notes
1	2 kerosene vented space heaters; 2 propane unvented space heaters; wood stove	DHP does not fully meet heating loads; owner recently added electric baseboard
2	2 propane fired wood stove fireplace inserts	No central heating system; DHPs are now primary heat source
3	Oil-fired boiler	Partially displaced; DHPs are now primary heat source
4	2 kerosene vented space heaters	DHP used for heating and cooling during monitoring period; now used primarily for cooling and dehumidification
5	Propane fired boiler & wood pellet stove	3 DHPs displace 100% of propane and 50% of wood pellet usage
6	Oil-fired boiler	Oil heat now used only during very cold weather; DHPs are now primary heat source
7	Oil-fired boiler	2 DHPs now provide the majority of the space heat
8	Oil-fired boiler & wood stove	Oil heat now used only during very cold weather; DHPs are now primary heat source
9	Oil-fired boiler – radiant floor	DHP replaces radiant heat for sunroom

Table 3-4. Baseline Cooling Systems Replaced/Displaced

Site #	Cooling Replaced	Proposed Cooling Displaced
1	1 window A/C unit 5,000 Btu/h (remains in-place, but not used)	N/A
2	5 window A/C units; sizes unknown (2 DHPs installed)	N/A
3	1 window A/C unit; size unknown	N/A
4	2 window A/C units; 6,000 Btu/h & 10,000 Btu/h (3 DHPs installed)	N/A
5	2 window A/C units; sizes unknown (3 DHPs installed)	Planned repair of older central A/C, or additional window A/C
6	No existing A/C	Planned central A/C & dehumidification
7	No existing A/C	Standard central ASHP, or standard DHP
8	1 window A/C unit; 15,000 Btu/h (remains in place, but is not used)	N/A
9	No existing A/C	DHP installed as an "experiment" – no previous plan for cooling

4 Heating Season Monitored Energy Usage and Calculated Savings

Electric demand (kW) and usage (kWh) were monitored for all nine sites. The logged data was uploaded to a spreadsheet tool along with corresponding weather data in order to calculate seasonal performance. In order to normalize the data for average weather conditions, we utilized typical meteorological year version 3 (TMY3) weather data for Laconia, NH. TMY3 data represents average conditions for the most recent 15-year period recorded.

4.1 Procedure

Our site monitoring began in January and continued through the following September, providing for three-season data. In order to calculate heating season savings, the procedure followed included:

- For each site, identification of a contiguous time period(s) when the monitored system was being routinely used for heating. This was typically January through May.
- Correlating the time periods with recorded NOAA weather data
- Utilizing the TMY3 weather data to normalize the logged data
- Extrapolating the normalized data for an entire heating season (September 15 – May 31)

4.2 Heating Season Normalized Electric Usage

Table 4-1 presents the normalized heating season usage for the nine sites. The monitored DHPs installed at sites 3, 4, and 5 are installed in smaller spaces and the homes include additional DHP units, resulting in noticeably lower usage per ton than the DHPs monitored at the other sites.

Table 4-1. Weather Normalized Heating Season (Sept 15 – May 31) Electric Usage

Site #	1*	2	3	4	5	6	7	8	9	Average* (Sites 2-9)
System mfg.	Fujitsu	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	N/A
Model	30RLX	FE09NA	FE12NA	FE12NA	FE12NA	FE18NA	FE18NA	FE18NA	FE12NA	N/A
Heat cap. (Btu/h)	37,500	10,900	13,600	13,600	13,600	21,600	21,600	21,600	13,600	16,263
Rated HSPF	9.5	10.0	10.6	10.6	10.6	10.3	10.3	10.3	10.6	10.4
Adjusted HSPF**	8.55	9.00	9.54	9.54	9.54	9.27	9.27	9.27	9.54	9.4
Avg heating COP	2.51	2.64	2.8	2.8	2.8	2.72	2.72	2.72	2.8	2.8
DHP energy usage (kWh)	3,682	3,812	2,514	1,978	1,937	3,741	3,913	5,323	4,151	3,421

*Site #1 is not included in the average calculations, as it is not a cold-climate model.

**The HSPF is adjusted by a factor of 0.9 to account for climate conditions for central New Hampshire.

4.3 Heating Season Calculated Savings

The normalized usage data was then used to calculate estimated savings using three baselines. The algorithm used for the calculations is the algorithm presented in Section 11 of this report with the exception that the HSPF rating is adjusted and the actual energy metered usage of the DHPs is utilized to determine the equivalent full-load hours (EFLH).

4.3.1 Establishing Baselines

In order for the calculated savings to be useful for program planners and evaluators, one or more baselines need to be established. For the study participants, the heating fuels displaced were a combination of fossil fuels (#2 heating oil, kerosene, propane) and biomass (cordwood, wood pellets). In most cases these are not the baselines that will be used by program administrators. As such, four baselines were selected:

- ❑ Electric resistance baseboard heat at a COP of 1.0
- ❑ Standard air-source heat pump (ASHP) at the current federal minimum efficiency level adjusted for the climate conditions
- ❑ Oil-fired boiler with a system efficiency (includes distribution losses) of 78%, as a baseline for calculating the impacts of fuel-switching from oil
- ❑ Gas-fired boiler or furnace with a system efficiency (includes distribution losses) of 80%, as a baseline for calculating the impacts of fuel-switching from natural gas

The baseline energy usage for the savings calculations represents the amount of fuel needed to provide the same amount of heat as the metered DHP.

4.3.2 Adjusted Performance Ratings

As noted in the Executive Summary, standard performance ratings represent performance at particular operating and weather conditions that may not be appropriate for the actual installed conditions. Heating seasonal performance factor (HSPF) is the rating that is currently mandated for published performance data for DHPs sold in the US. Although HSPF can be calculated for six differing heating climate zones, manufacturers are required to publish HSPF for only one; zone IV which includes parts of coastal Southern New England, and extends south and west to include parts of Virginia, Kansas, and New Mexico. Figure 4-1 presents the “generalized climate zone map” published in the Federal Register that defines the regions and estimated full-load heating hours that are used for AHRI testing procedures. As the map illustrates, New England straddles the boundary dividing zones IV and V. The study region of central NH is located in zone V.

Figure 4-1. Generalized Climate Zone Map Utilized for HSPF Ratings[7]

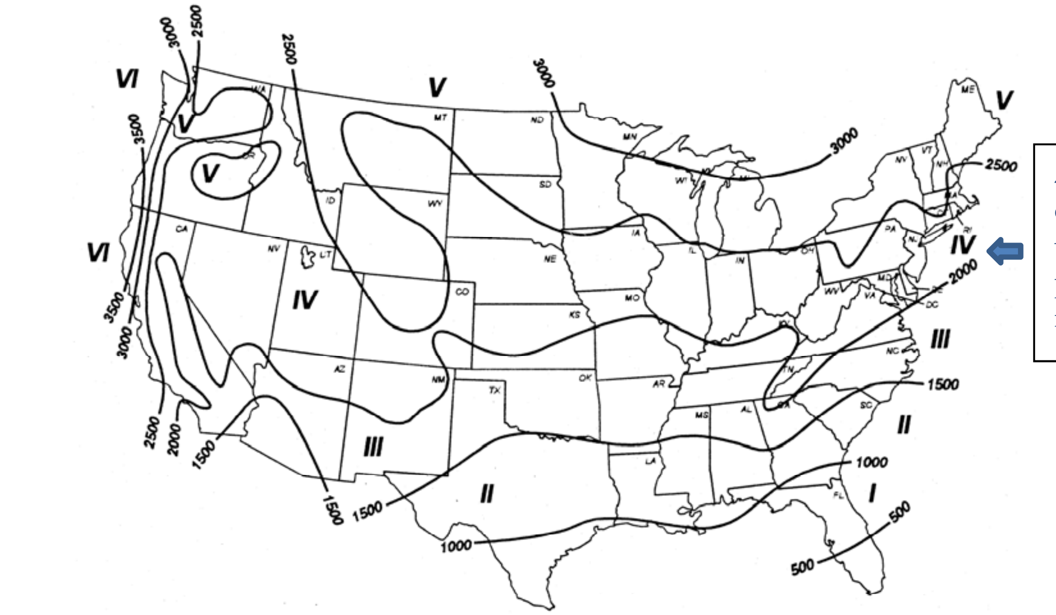


Figure 2 Heating Load Hours (HLH_A) for the United States

Zone IV climate conditions are utilized for published HSPF ratings

HSPF ratings for zone V, or any zone other than zone IV were not available from the manufacturers. In order to better estimate savings for Northern New England, we have made the following adjustments to HSPF ratings

- **Installed DHPs** –Reviewing the climate zone map in Figure 4-1, calculations utilized in other published independent studies, and correlating to our own calculated COP data, we have used a factor of 0.9[8] in order to adjust the performance rating for the New England climate. The factor is appropriately conservative given that the range of heating hours covered by zone IV is 2,000 to 2,500 hours and 2,500 to 3,000 hours for zone V.
- **Standard ASHP** – The DOE supports a tool (HeatCalc) for calculating heating savings for a variety of systems, including ASHPs. The tool includes an adjusted HSPF of 5.1 for Concord, NH, which is adjusted downward from 7.7. This rating adjusts for both the heat pump efficiency at colder temperatures and the use of electric resistance coils below the standard minimum operating temperature of 17°F. We have utilized an adjusted HSPF of 5.1 for the savings calculations. HeatCalc was not used for the installed DHPs because it does not include a rating procedure for cold-climate DHPs. Some efficiency programs

[7] 2010 Standard for Performance Rating of Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment, Air-Conditioning, Heating and Refrigeration Institute.

[8] The 0.9 factor was determined to be reasonable for cold-climate DHPs installed in northern New England, following the review of the estimated COP from this study, and the HSPF adjustment factors developed in other studies including those conducted by Ecova for NEEA. A smaller adjustment factor for southern New England may be appropriate, as the climate conditions are closer to those used to determine published HSPF ratings.

reference the HeatCalc adjusted rating for ASHPs and DHPs. However, it lacks accuracy for cold-climate DHPs because it assigns cold-weather operation below 17°F to electric resistance at a COP of 1.0 when the DHP is actually operating as a heat pump with a significantly higher COP.

4.3.3 Calculated Electric Savings

Table 4-2 presents the estimated normalized savings associated with a baseline of electric resistance baseboard heat at a COP of 1. As discussed in the previous section, the HSPF has been adjusted with a 0.9 factor for the NH climate. The average heating COP is calculated from the adjusted HSPF and is consistent with our findings regarding field-calculated COP, presented in Section 6. Because the installed systems vary in size, a weighted average savings for 1 ton (12,000 Btu) of heating is also presented.

Table 4-2. Monitored DHP Normalized Heating Season (Sept 15 – May 31) Usage & Savings Compared with Electric Resistance Baseline

Site #	1*	2	3	4	5	6	7	8	9	Average* (Sites 2-9)
System mfg.	Fujitsu	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	N/A
Model	30RLX	FE09NA	FE12NA	FE12NA	FE12NA	FE18NA	FE18NA	FE18NA	FE12NA	N/A
Heat cap. (Btu/h)	37,500	10,900	13,600	13,600	13,600	21,600	21,600	21,600	13,600	16,263
Rated HSPF	9.5	10.0	10.6	10.6	10.6	10.3	10.3	10.3	10.6	10.4
Adjusted HSPF**	8.55	9.00	9.54	9.54	9.54	9.27	9.27	9.27	9.54	9.4
Avg heating COP	2.51	2.64	2.8	2.8	2.8	2.72	2.72	2.72	2.8	2.8
Baseline electric resistance usage (kWh)	9,226	10,054	7,030	5,531	5,416	10,164	10,630	14,460	11,605	9,361
DHP energy usage (kWh)	3,682	3,812	2,514	1,978	1,937	3,741	3,913	5,323	4,151	3,421
DHP savings (kWh)	5,544	6,242	4,515	3,552	3,478	6,423	6,717	9,137	7,454	5,940
Savings @ $0.14/kWh	$776	$874	$632	$497	$487	$899	$940	$1,279	$1,044	$832
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating (kWh)										**4,502**
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating										**$630**

* Site #1 is not included in the average calculations, as it is not a cold-climate model.

** The HSPF is adjusted by a factor of 0.9 to account for climate conditions for central New Hampshire.

Table 4-3 presents the savings for the same systems against a baseline of standard efficiency ASHPs. The baseline HSPF is the minimum federal efficiency standard for residential ASHPs, adjusted for Concord, NH[9], weather conditions using the climate factors utilized for the DOE savings calculation tool HeatCalc. These adjustment factors are further described in Section 4.3.2.

[9] The Laconia, NH, weather station is the closest to the monitored sites and it was utilized for TMY3 weather data normalization. Concord, NH, is the closest weather station utilized by HeatCalc.

Table 4-3. Estimated Weather Normalized Heating Season (Sept 15 – May 31) Usage & Savings Compared with Standard ASHP Baseline

Site #	1*	2	3	4	5	6	7	8	9	Average* (Sites 2–9)
System mfg.	Fujitsu	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	N/A
Model	30RLX	FE09NA	FE12NA	FE12NA	FE12NA	FE18NA	FE18NA	FE18NA	FE12NA	N/A
Heat cap. (Btu/h)	37,500	10,900	13,600	13,600	13,600	21,600	21,600	21,600	13,600	16,263
Rated HSPF	9.5	10.0	10.6	10.6	10.6	10.3	10.3	10.3	10.6	10.4
Adjusted HSPF**	8.55	9.00	9.54	9.54	9.54	9.27	9.27	9.27	9.54	9.4
Avg heating COP	2.51	2.64	2.8	2.8	2.8	2.72	2.72	2.72	2.8	2.8
Baseline rated HSPF	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7
Baseline adjusted HSPF***	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1
Baseline ASHP usage (kWh)	6,173	6,727	4,703	3,700	3,623	6,801	7,113	9,675	7,765	6,263
DHP energy usage (kWh)	3,682	3,812	2,514	1,978	1,937	3,741	3,913	5,323	4,151	3,421
DHP savings (kWh)	2,491	2,915	2,189	1,722	1,686	3,059	3,199	4,352	3,614	2,842
Savings @ $0.14/kWh	$349	$408	$306	$241	$236	$428	$448	$609	$506	$398
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating (kWh)										**2,154**
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating										**$302**

*Site #1 is not included in the average calculations, as it is not a cold-climate model.

**The DHP HSPF is adjusted by a factor of 0.9 to account for climate conditions for central New Hampshire.

***The adjusted HSPF for baseline ASHPs is adopted from the DOE HeatCalc adjusted ratings for standard ASHPs and is an adjustment factor of approximately 0.66.

4.3.4 Calculated Btu and Oil Heat Savings

Although most efficiency programs do not currently support fuel switching, there is mounting pressure to do so because concerns over meeting climate change goals are increasing. Also, programs funded with proceeds from carbon/GHG credits and other programs not funded through system benefit charges may consider promoting fuel switching to electric with DHPs. Table 4-4 presents the calculated savings associated with oil-fired heat, using the Btu content of #2 fuel oil and a 78% average system efficiency, including distribution losses. The Btu savings could also be used to calculate the savings for other fuels and heating systems if the efficiency and fuel Btu content are known.

Table 4-4. Estimated Weather Normalized Heating Season (Sept 15 – May 31) Savings Compared with Oil Heat Baseline

Site #	1*	2	3	4	5	6	7	8	9	Average (Sites 2–9)
System mfg.	Fujitsu	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	N/A
Model	30RLX	FE09NA	FE12NA	FE12NA	FE12NA	FE18NA	FE18NA	FE18NA	FE12NA	N/A
Heat cap. (Btu/h)	37,500	10,900	13,600	13,600	13,600	21,600	21,600	21,600	13,600	16,263
Rated HSPF	9.5	10.0	10.6	10.6	10.6	10.3	10.3	10.3	10.6	10.4
Adjusted HSPF**	8.55	9.00	9.54	9.54	9.54	9.27	9.27	9.27	9.54	9.4
Avg heating COP	2.51	2.64	2.8	2.8	2.8	2.72	2.72	2.72	2.8	2.8
Oil heat baseline output (MMBtu)	31.48	34.31	23.98	18.87	18.48	34.68	36.27	49.34	39.60	31.94
Avg. oil heating system efficiency	78%	78%	78%	78%	78%	78%	78%	78%	78%	78%
Oil heat baseline fuel usage (MMBtu)	40.4	44.0	30.7	24.2	23.7	44.5	46.5	63.3	50.8	40.9
Oil baseline fuel displaced (gal)	291	317	222	174	171	321	335	456	366	295
Oil baseline fuel cost savings @ $3.70	$1,077	$1,173	$820	$645	$632	$1,186	$1,241	$1,687	$1,354	$1,092
DHP usage (MMBTU)	12.56	13.01	8.58	6.75	6.61	12.77	13.35	18.16	14.16	11.67
DHP energy usage (kWh)	3,682	3,812	2,514	1,978	1,937	3,741	3,913	5,323	4,151	3,421
DHP energy cost @$0.14	$516	$534	$352	$277	$271	$524	$548	$745	$581	$479
Net fuel cost savings	$561	$640	$468	$368	$361	$662	$693	$942	$773	$613
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating (MMBtu)										**22.19**
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating										**$465**

* Site #1 is not included in the average calculations, as it is not a cold-climate model.

** The DHP HSPF is adjusted by a factor of 0.9 to account for climate conditions for central New Hampshire.

4.3.5 Natural Gas Potential Savings

Although natural gas is not used or currently available at the participant sites, the potential for natural gas savings was considered. Using the above Btu methodology, and an energy cost of $1.50 per therm (the current average residential price in Massachusetts), potential natural gas savings were calculated and are presented in Table 4-5.

Table 4-5. Estimated Weather-Normalized Heating Season (Sept 15 – May 31) Savings Compared with Natural Gas Heat Baseline

Site #	1*	2	3	4	5	6	7	8	9	Average* (Sites 2–9)
System	Fujitsu	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	N/A
Model	30RLX	FE09NA	FE12NA	FE12NA	FE12NA	FE18NA	FE18NA	FE18NA	FE12NA	N/A
Heat cap. (Btu/h)	37,500	10,900	13,600	13,600	13,600	21,600	21,600	21,600	13,600	16,263
Rated HSPF	9.5	10.0	10.6	10.6	10.6	10.3	10.3	10.3	10.6	10.4
Adjusted HSPF**	8.55	9.00	9.54	9.54	9.54	9.27	9.27	9.27	9.54	9.4
Avg heating COP	2.51	2.64	2.8	2.8	2.8	2.72	2.72	2.72	2.8	2.8
NG heat baseline output (MMBtu)	31.48	34.31	23.98	18.87	18.48	34.68	36.27	49.34	39.60	31.94
Avg. NG heating system efficiency	80%	80%	80%	80%	80%	80%	80%	80%	80%	80%
NG baseline usage (MMBtu)	39.3	42.9	30.0	23.6	23.1	43.3	45.3	61.7	49.5	39.9
NG baseline fuel displaced (therm)	393	429	300	236	231	433	453	617	495	399
NG baseline cost @ $1.50/therm	$590	$643	$450	$354	$346	$650	$680	$925	$742	$599
DHP Usage (MMBTU)	12.56	13.01	8.58	6.75	6.61	12.77	13.35	18.16	14.16	11.67
DHP energy usage (kWh)	3,682	3,812	2,514	1,978	1,937	3,741	3,913	5,323	4,151	3,421
Energy cost @$0.14	$516	$534	$352	$277	$271	$524	$548	$745	$581	$479
Net fuel cost savings	$75	$110	$98	$77	$75	$126	$132	$180	$161	$120
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating (MMBtu)										**21.41**
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating										**$91**

* Site #1 is not included in the average calculations, as it is not a cold-climate model.

** The DHP HSPF is adjusted by a factor of 0.9 to account for climate conditions for central New Hampshire.

5 AIR-CONDITIONING LOAD IMPACTS & LOAD SHAPE

The potential for load building associated with incentive programs supporting DHPs is a concern for program administrators and regulators. The issue focuses on the possibility that efficiency program participants who install DHPs to replace or displace electric resistance or fossil fuel heat will also use the units to cool spaces that were not previously cooled, thereby building cooling season peak loads.

In terms of A/C load impacts, the participants in this study fall into three categories:

1. **Replacement of window A/C units** – Six participants removed and/or discontinued using less efficient window A/C units upon installation of the DHP(s). Although in some cases the DHPs were rated for higher cooling capacities, and in those cases the improved efficiency and the fact that the systems modulate to meet the load would argue against load building.
2. **Installed as an alternative to central A/C** – Two participants engaged a contractor with the objective of installing central A/C in the home. And one participant who replaced a window unit was considering having a disabled central A/C system repaired. The three participants reported that the contractor explained the advantages of DHPs and the possibility of participating in the NHEC program. The decision was made to install the DHPs instead of a new central system. The improved efficiency again argues against cooling season load-building.
3. **Displacement of heating and introduction of incidental cooling** – One participant installed a DHP in a sunroom as an “experiment.” After successful testing, the system was used to replace radiant floor heating and to introduce cooling. In this case, some cooling season load-building is occurring.

Our interviews with the homeowners as well as an analysis of the monitored data, led to a conclusion that there is no summer load-building occurring with eight of the nine participant homes. Rather, if it can be assumed that if usage patterns are similar for DHPs as for either of the two baselines (window A/C units, and standard central A/C), the increased efficiency results in A/C savings for eight of the nine participant homes.

However, several factors should be considered when making the above assumptions, and when estimating the cooling savings for DHPs:

- DHPs are operated differently from either central or window A/C. A remote control is used to select the operating mode. Central A/C is operated by a conventional or programmable thermostat. Window units are operated by an on/off switch typically with a combination of power settings and integral thermostats.
- Participant and market actor interviews reported that DHP purchasers often initially contacted installation contractors seeking to add central A/C to their homes. For those moving ahead with a DHP installation, it can be assumed that alternatively they would

have chosen to install central A/C, or perhaps window A/C if the installation costs of a central system presented a barrier.

- Participant interviews reported that the homeowners manually choose when to operate the DHP for cooling, rather than rely on a cooling setpoint.
- The modulating nature of DHPs allows them to provide cooling in part-load conditions without cycling on and off.
- DHPs typically offer better zone control than that offered by central A/C. The ability to cool occupied rooms, rather than an entire building or floor, may result in additional savings beyond those calculated using weather-based analysis. The planned introduction of multi-zone cold-climate DHPs will further the zoning advantages for both heating and cooling.
- DHPs are much quieter than most window A/C units.
- Compared with window A/C, DHPs often offer improved occupant comfort.

Many of the above points could be used to argue for or against additional A/C savings for the installed DHPs. However, on balance these factors support a conclusion that there is no significant summer load-building, and even in the heating-dominant climate of the study region some cooling savings are obtained.

5.1.1 Savings Calculations

This project was unsuccessful in its attempt to calculate savings against the two identified baselines utilizing a methodology similar to that used for heating, (i.e., based on normalized weather data and the estimated efficiency levels of the units). It was determined that we could not reliably use that methodology due to several factors:

- With the DHPs sized for NH heating loads, the cooling loads are very small, and it proved difficult to identify periods when active cooling was taking place based on efforts to correlate weather data and modulations at minimum loads from long-term interval monitoring records. In fact, utilizing weather data for cooling impacts in the studied region likely provides inaccurate results for any user-controlled cooling system.
- A review of the cooling season data demonstrated that the DHP usage patterns were more random than they were correlated to weather conditions, pointing again to homeowner manual control. The team deemed it inappropriate to select very limited time periods when cooling occurred and attempted to correlate and normalize to weather data.

We did, however, use two alternative methodologies to identify and estimate potential cooling savings: using the monitored total summer usage to calculate baseline usage and savings, given identical operational parameters. And the ENERGY STAR calculation tool, which uses cooling degree days for the region and system efficiency to estimate equivalent full-load operating hours (EFLH) and resulting savings.

Monitored Energy Usage Methodology

The calculations presented in Tables 5-1 and 5-3 utilize the energy usage logged during June, July, and August, along with the installed unit SEER to calculate savings compared with both standard central A/C units and window A/C units. These two cooling scenarios were calculated because both types of cooling can be found in the region, and both represent typical baselines found throughout the Forum sponsor territories and are therefore helpful in informing program planning, design, and evaluation.

Assumptions:

- All DHP usage in June, July, and August is associated with cooling.
- May and September have some cooling conditions, but in reviewing the data we could recognize very little usage associated with cooling in the shoulder seasons.
- Site 9 had no A/C and no intention of installing A/C prior to installation of the DHP.
- Baseline efficiency levels are the current federal minimum standards
  - 13 SEER for central A/C
  - 9.7 EER for window A/C

Table 5-1 presents the calculated savings associated with a central A/C baseline. Table 5-3 presents the savings for a window A/C baseline.

ENERGY STAR Calculation Methodology

Primarily as a way to cross-reference the results and to illustrate the results of using a widely accepted tool, we also used an ENERGY STAR Calculator to estimate cooling. The assumptions are the same as for the previous methodology. Table 5-2 presents the results for central A/C. An ENERGY STAR-calculated result is not presented for a window A/C baseline, as the calculator does not include a reasonable methodology for comparing DHPs and window A/C. It is interesting to note that the average savings for all sites is fairly comparable, but the range of savings is much greater using the actual logged data. This is further evidence that user decisions are a major driver of A/C usage in this climate.

Table 5-1. Metered Data Calculated Cooling Savings Compared with Central A/C Baseline*

Metered kWh					kWh	Calculated kWh		Savings @
Site	SEER	June	July	August	Total	Baseline	Saved	$0.14
#1	17.5	183	200	185	568	765	197	$27.51
#2	26	70	15	6	91	182	91	$12.74
#3	23	131	213	171	515	910	395	$55.36
#4	23	52	75	25	152	269	117	$16.34
#5	23	149	214	119	482	852	370	$51.81
#6	20.2	154	180	7	341	530	189	$26.44
#7	20.2	187	274	222	683	1061	378	$52.97
#8	20.2	81	100	63	244	379	135	$18.92
#9	23	18	119	19	156	0	-156	-$21.84
Average savings							**191**	**$26.70**
Weighted average savings per ton (12,000 Btu) of cooling							**153**	**$21.49**

Table 5-2. ENERGY STAR Methodology – Calculated Cooling Savings Compared with Central A/C Baseline*

ENERGY STAR Calculated kWh Usage & Savings					Savings @
Site	SEER	Baseline	Installed	Savings (kWh)	$0.14
#1	17.5	1066	792	274	$38.36
#2	26	267	133	133	$18.66
#3	23	355	201	154	$21.61
#4	23	355	201	154	$21.61
#5	23	355	201	154	$21.61
#6	20.2	533	343	190	$26.60
#7	20.2	533	343	190	$26.60
#8	20.2	533	343	190	$26.60
#9	23	0	201	-201	-$28.14
Average savings				**138**	**$19.28**
Weighted average savings per ton (12,000 Btu) of cooling				**101**	**$14.17**

*Sites 6 and 7 had planned to install central A/C prior to installing DHPs

Table 5-3. Metered Data Calculated Cooling Savings Compared with Window A/C Baseline*

Site	Metered kWh				kWh	Calculated kWh		Savings @
	SEER	June	July	August	Total	Baseline	Saved	$0.14
#1	17.5	183	200	185	568	1025	457	$63.96
#2	26	70	15	6	91	244	153	$21.42
#3	23	131	213	171	515	1220	705	$98.76
#4	23	52	75	25	152	360	208	$29.15
#5	23	149	214	119	482	1142	660	$92.44
#6	20.2	154	180	7	341	710	369	$51.71
#7	20.2	187	274	222	683	1423	740	$103.57
#8	20.2	81	100	63	244	508	264	$37.00
#9	23	18	119	19	156	0	-156	-$21.84
Average savings							**378**	**$52.91**
Weighted average savings per ton (12,000 Btu) of cooling							**299**	**$41.84**

* Sites 1, 2, 3, 4, 5, and 8 removed or disabled window A/C units following installation of DHPs

Given the limited cooling season in central NH, and the lack of any significant cooling season load-building evidenced in this study, it is reasonable to conclude that cooling season load-building is not a significant factor in promoting DHP installations. It is also evident that cooling savings are likely obtained even in heating-dominant climates. For regions with greater cooling loads, it can be argued that DHP installations will nearly always replace existing less-efficient A/C units, or will be installed as an efficient option to other proposed options, and DHP programs will reduce cooling season loads rather than build them.

5.1.2 Replaced Cooling Equipment Disposition

A related consideration for cooling loads is the disposition of replaced equipment, especially window A/C units, which are easily installed in another location. Of the five participants who had replaced window A/C units, one reported "gifting" the replaced units to their adult children. The other four had either disposed of the units or had retained them, but reported that they were no longer used. Although there is likely some load-building effect from the continued use of replaced equipment, this issue is present in many efficiency measures and is a policy consideration regarding the disposition of replaced equipment. Many program administrations require the disposal of replaced equipment, and some offer window A/C trade-in programs that could be expanded to DHPs.

5.2 Cooling Season Load Shape

This section presents an average energy consumption load shape for the metered DHPs. In order to calculate a savings load shape, the consumption load shape for the baseline condition—presumably an ASHP, central AC, or window-mount A/C—needs to be established and compared against the DHP load shape for similar climatic conditions.

The monitored cooling load shape was calculated from the monitored power data, normalized with TMY3 weather data. Figures 5-1 plots the average metered (not weather-normalized) load shape, per ton of cooling, for the nine systems. Figure 5-2 plots the calculated cooling season load shapes, per ton of cooling, weather-normalized with the TMY-3 data. Figure 5-3 plots the weather-normalized load shape, per ton of cooling, for each system. The June through August 1:00–5:00 p.m. non-holiday weekday timeframe targeted by the NE-ISO for load reduction is highlighted. The average weather-normalized kW demand per ton (12,000 Btu/hr) of cooling between 1:00 and 5:00 p.m., June through August, is estimated to be 0.135 kW.

Figure 5-1. Metered Average Load Shape for June, July, and August per Ton of Cooling – All Monitored DHPs

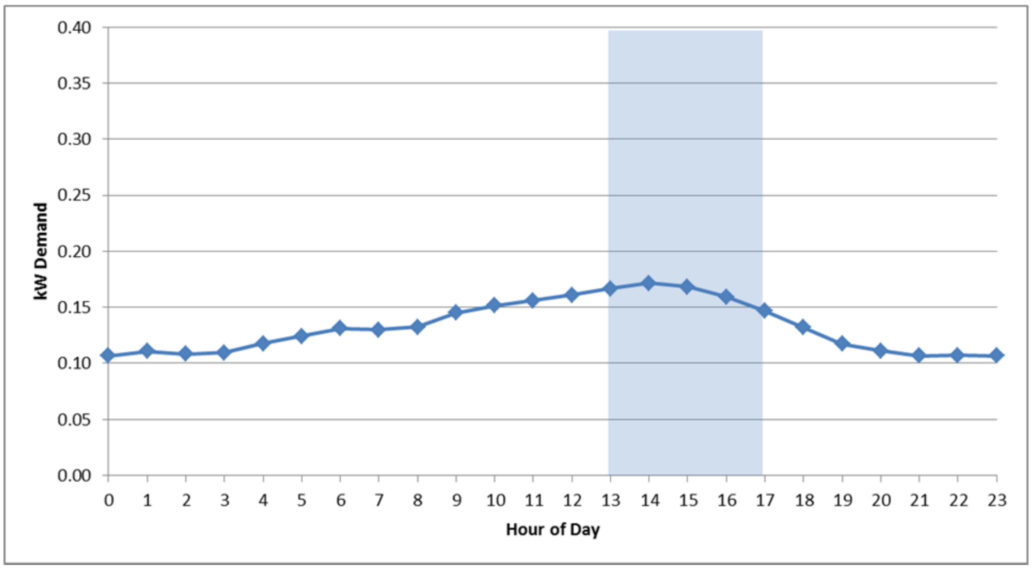


Figure 5-2. Weather-Normalized Average Load Shape for June, July, and August per Ton of Cooling – All Monitored DHPs

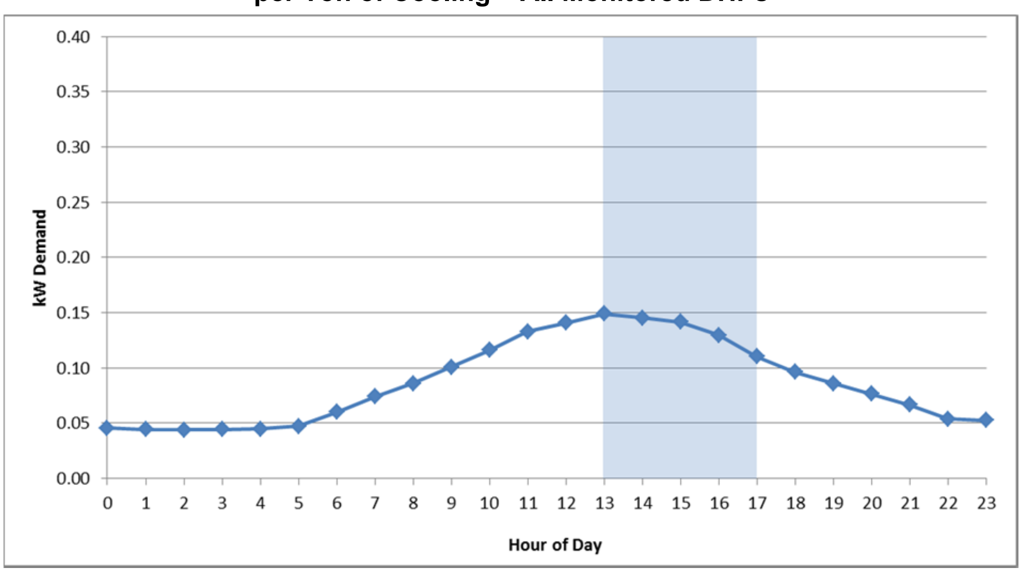
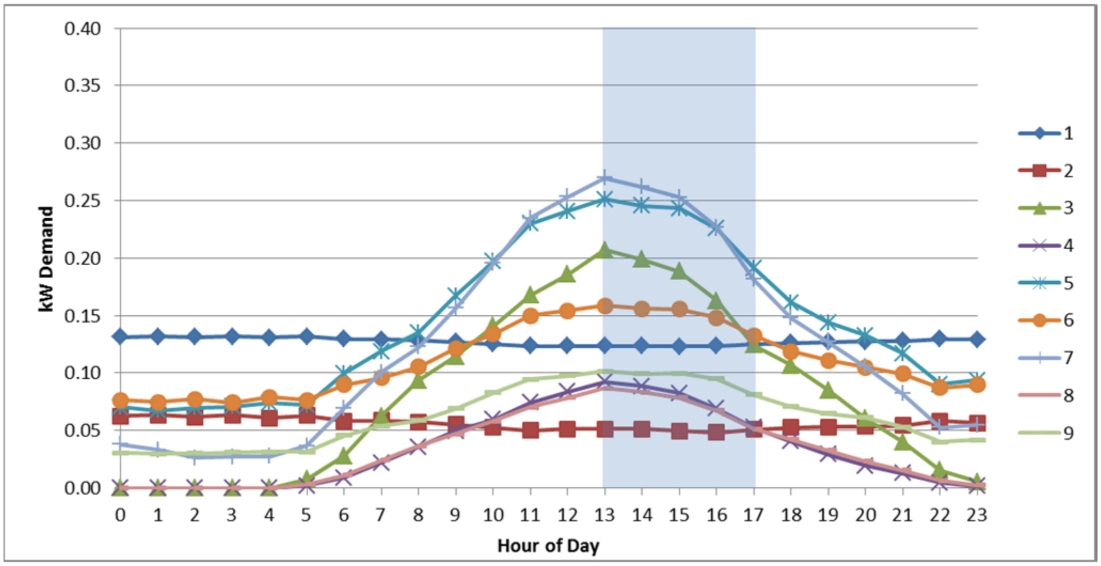
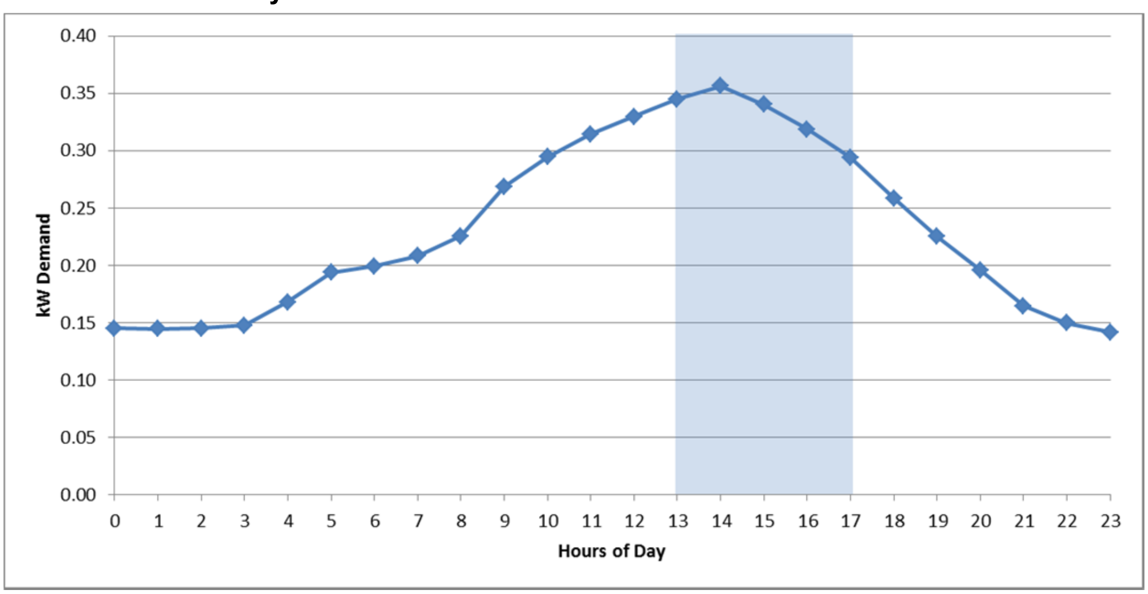


Figure 5-3. Weather-Normalized Average Load Shape for June, July, and August per Ton of Cooling – All Monitored DHPs



In order to illustrate the load shape for cooling during hot-weather periods, the demand was plotted for days when the OAT reached a minimum of 90°F at least once. Figure 5-4 plots the load shape restricted to those days.

Figure 5-4. Average Metered Load Shape for June, July, and August per Ton of Cooling, Restricted to Days When OAT Reached a Minimum of 90°F – All Monitored DHPs

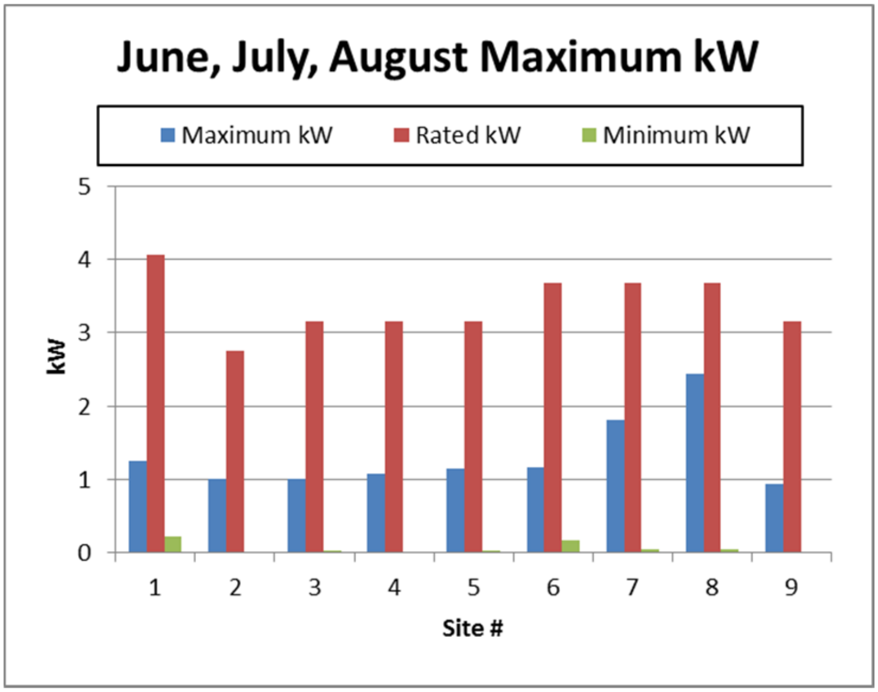


Sites 1 and 2 recorded very flat cooling load shapes. The metered data was re-checked and found to be recorded accurately. It is clear that both systems were "idling" in standby and/or operating at a very low setting for long periods of time. Some intermittent evening and nighttime cooling was recorded, but the load was small and had little effect on average values.

An additional observation is that the continuously modulating nature of inverter driver DHPs, especially when sized for heating in NH, results in average peak loads far below the maximum kW rating of the units. Figure 5-5 presents the maximum rated and maximum/minimum metered kW for the cooling season. Metered peak demands for all seasons are presented in Section 9.

Appendix C includes individual summer load shape charts for each of the nine sites.

Figure 5-5. Cooling Season Maximum kW Demand



6 PERFORMANCE RATINGS AND ESTIMATED COEFFICIENT OF PERFORMANCE

Heat pump and A/C equipment is rated using standard test procedures under laboratory conditions. The standard ratings are useful for comparing systems and for approximating performance in the field. The laboratory conditions are matched in the field for very brief periods, if at all. However, field-collected data that represents the components of rating procedures can be used to approximate standard ratings and correlate performance levels. For this project, we utilized the field monitored data to estimate COP to correlate with the adjustments to HSPF that were used to estimate savings associated with operation in central NH climate conditions, and to compare with COP estimates reported in other studies.

DHPs are rated for performance using two standard testing methodologies developed by the Air-Conditioning, Heating and Refrigeration Institute (AHRI): Heating Seasonal Performance Factor (HSPF); and Seasonal Energy Efficiency Ratio (SEER). Coefficient of performance (COP) can be considered a basic component of both HSPF and SEER, but it is a measurement of steady state performance. The three terms are defined by the AHRI as follows:

- **HSPF** – The total space heating output during the space heating season, expressed in Btus, divided by the total electrical energy consumed by the heat pump system during the same season.[10]
- **SEER** – The total heat removed from the conditioned space during the annual cooling season, expressed in Btus, divided by the total electrical energy consumed by the air conditioner or heat pump during the same season.
- **COP** – The ratio of the average rate of space heating (or cooling) delivered to the average rate of electrical energy consumed by the heat pump. These rate quantities must be determined from a single test or, if derived via interpolation, must be tied to a single set of operating conditions.

For this study we analyzed the monitored data and reviewed the reported homeowner operating conditions to identify operating conditions that were conducive to correlating to rated COP performance levels, and we are reporting those results. However there were several challenges to estimating COP from the logged data, and as a result, we did <u>not</u> utilize the estimated COP to calculate DHP savings.

6.1 Challenges to Field Performance Calculations

Recent DHP testing performed by Ecotope, in association with the National Renewable Energy Laboratory (NREL) for the Northwest Energy Efficiency Alliance (NEEA), performed laboratory testing and field testing of DHPs with a goal of determining COP values. The difficulty of producing valid results is detailed in the study reports. The following is an excerpt from the NEEA/Ecotope <u>Ductless Heat Pump Impact & Process Evaluation: Lab-Testing Report</u>:

[10] This study utilized an adjusted HSPF rating, as detailed in Section 4.

"One important finding is learning how challenging it is to replicate the ratings for variable speed equipment. There are a total of eight heating tests and seven cooling tests which must be exactly replicated if one expects to match literature data. Further, the critical values in the rating calculation come from 'intermediate' speed compressor operation. The intermediate speed is generally the basis for nominal input, output, and COP at a particular testing temperature."[11]

Field operating conditions never result in a "single set of operating conditions." DHPs by their very nature do not operate under steady-state conditions. The operational parameters that present challenges include:

- **Inverter driven modulation** – DHPs are controlled by an inverter and variable speed drive combination. Compressor power usage and output is continuously variable. In the lab, they are "forced" into a steady-state for COP measurements. The decision to monitor the DHP installations for an extended time period required that power be monitored at 10 minute intervals in order to conserve logger storage space. Although supply and return temperatures remain fairly constant, the power usage varies – even over the course of just a few minutes, making correlating ΔT to the metered interval data for power difficult.
- **User controls** – The great majority of DHPs in service are controlled by handheld remote controls, as are the nine included in this study. Users have a wide variety of options, including fan speeds, setpoints, dehumidification modes, etc.
- **Fan speed/air-flow** – Due to the difficulty of monitoring within occupied homes, no attempt was made to monitor supply air flow. In fact, monitoring air flow has been a struggle for all of the DHP studies reviewed. With the assistance of Mitsubishi technical representatives we were able to make assumptions regarding fan speed and correlated air flow from the data collected. In addition, according to Mitsubishi, when laboratory testing for COP, the highest fan speed is always utilized for the test.
- **Supply air louvers** – DHPs incorporate louvers that are adjustable with the remote control. For field monitoring in customer homes, the adjustment of the louvers can have a significant impact on the monitoring of the supply air temperature, due to both direction and turbulence, which tends to mix air right at the delivery point.
- **Participant site heating configurations** – Only site 9 includes a space heated solely by the monitored DHP. The other sites include heating contribution from additional DHPs, and a variety of space and central heating systems. This may have contributed to inconsistent data, as there were many periods, especially during colder weather when both the recorded ΔT and kW were very low.
- **Very small cooling loads** – Being sized for heating loads in central NH, the units are oversized for the cooling loads encountered. It was found to be very difficult to calculate cooling COP from the monitored data, with calculated COP values ranging from very low

[11] NEEA/Ecotope, Ductless Heat Pump Impact & Process Evaluation: Lab-Testing Report, pg. 28.

to very high. It is the project team's belief that the very low power draws and low return/supply air ΔTs indicate periods of dehumidification only, standby, and modulation down to the lower limits of the units. This makes it difficult to assign proper values to the COP calculations.

6.2 Estimated COP Results

Despite the challenges outlined above, by identifying monitored results that could be associated with conditions that were closer to steady state and also represented normal operating conditions, we have estimated COP ranges for several different temperature ranges. The data utilized comes from sites 6 and 9, as the metered data from those sites represents the most consistent operational patterns. In addition, the DHP metered at site 9 is installed in a space that is served by no other heating system.

6.2.1 Estimated Heating COP

Table 6.1 presents monitored data and estimated COP ranges during the heating season. The COP for three fan speeds was calculated for each data point used[12]. A fourth fan speed, termed "powerful" is available. It must be manually selected, and the participants reported using auto settings, and sometimes manually selecting low or medium settings.

System operational characteristics and discussions with Mitsubishi personnel guided an interpretation of the fan speeds that were likely operational given the performance characteristics for the given time period. The shaded cells represent the data points we utilized to develop the estimated COPs. The estimated COP values represent the average of the calculated COPs presented in the shaded cells within each temperature bin.

Note that selecting the "high" fan speed setting exclusively for calculations when OATs are extremely low would significantly increase the estimated COP. Although the DHPs operate at these cold temperatures, the output is reduced, and a lower fan speed may be selected either manually or automatically to maintain the proper supply air temperature range. However, the metering procedure was unable to definitively identify fan speed at any particular time.

Mitsubishi Technical Representative and mechanical engineer Joseph Cefaly expressed the following concern regarding calculating COP from field-collected data: "Field COP testing is inherently several orders of magnitude more crude than the UL/ETL/AHRI certification testing that the manufacturer goes through to list equipment in the AHRI database (www.ahridirectory.org/). In the laboratory environment, the measurements are extremely accurate and the number of variables recorded is several times that which is captured during

[12] Heating energy produced by the heat pumps was estimated using the standard engineering calculation of (ΔT) x (specific heat of air) x (volume of air). Note: for the specific heat of air, a standard factor of 1.08 (BTUs to heat a cubic foot of air 1 degree F) was utilized.

field testing."[13] An example of an AHRI test report completed by Intertek, an AHRI-certified testing lab, is included in Appendix D.

[13] Email correspondence, Joseph Cefaly, March 2014, regarding the utilization of field-collected data for COP calculations, referencing this study and a concurrent study performed in New York State by Stephen Winter Associates.

Table 6-1. Estimated Heating COP Site 9

Site #9 Heating COP - Mitsubishi MSZ-FE12NA										
Day	Outside Air Temp (°F)	Return Air Temp (°F)	Supply Air Temp (°F)	Delta T (°F)	Amps	kW	Low CFM 166	Med CFM 240	High CFM 399	Estimated COP
2/5/2013 21:07	19.7	65.5	104.0	38.4	7.3	1.6	1.25	1.81	3.00	
2/5/2013 22:07	19.0	65.4	103.5	38.1	6.6	1.4	1.39	2.01	3.34	
2/5/2013 23:07	19.0	63.9	95.9	32.0	8.2	1.8	0.94	1.35	2.25	
2/6/2013 0:07	19.0	64.9	104.3	39.4	7.0	1.5	1.34	1.93	3.22	
2/6/2013 1:07	18.7	63.9	99.4	35.5	7.6	1.7	1.11	1.61	2.67	
2/6/2013 2:07	19.0	64.2	101.9	37.7	8.1	1.8	1.12	1.61	2.68	
Average	**19.1**	**64.6**	**101.5**	**36.8**	**7.5**	**1.6**	**1.18**	**1.70**	**2.83**	**2.27**
2/6/2013 13:07	34.7	65.9	94.1	28.2	6.0	1.3	1.11	1.61	2.68	
2/6/2013 14:07	35.3	67.3	99.1	31.8	3.8	0.8	2.01	2.91	4.84	
2/6/2013 15:07	33.3	66.1	89.9	23.8	4.8	1.1	1.19	1.72	2.86	
2/6/2013 16:07	30.0	66.7	85.0	18.4	3.8	0.8	1.14	1.65	2.74	
2/6/2013 17:07	27.3	66.3	85.4	19.1	3.9	0.9	1.15	1.67	2.78	
2/6/2013 18:07	26.3	65.3	97.8	32.6	6.3	1.4	1.23	1.78	2.96	
Average	**31.2**	**66.3**	**91.9**	**25.6**	**4.8**	**1.1**	**1.28**	**1.85**	**3.08**	**2.46**
2/7/2013 2:07	12.7	64.3	90.0	25.7	7.7	1.7	0.80	1.16	1.92	
2/7/2013 3:07	12.0	65.2	87.9	22.7	7.1	1.6	0.77	1.11	1.84	
2/7/2013 4:07	11.3	63.7	87.2	23.5	7.8	1.7	0.72	1.04	1.73	
2/7/2013 5:07	10.0	64.6	88.9	24.2	7.3	1.6	0.80	1.15	1.91	
2/7/2013 6:07	10.0	64.6	88.1	23.4	7.1	1.6	0.79	1.15	1.91	
2/7/2013 7:07	10.0	61.7	81.3	19.7	8.1	1.8	0.58	0.84	1.39	
Average	**11.0**	**64.0**	**87.2**	**23.2**	**7.5**	**1.6**	**0.74**	**1.07**	**1.78**	**1.42**
2/8/2013 9:07	11.3	66.1	85.2	19.1	6.4	1.4	0.72	1.03	1.72	
2/8/2013 10:07	12.0	65.2	83.8	18.6	7.6	1.7	0.58	0.84	1.40	
2/8/2013 11:07	12.0	65.2	102.0	36.9	7.4	1.6	1.19	1.72	2.85	
2/8/2013 12:07	12.7	64.5	95.9	31.4	7.7	1.7	0.97	1.40	2.33	
2/8/2013 13:07	14.0	65.8	103.8	38.1	7.1	1.6	1.28	1.85	3.08	
2/8/2013 14:07	16.0	65.3	98.8	33.5	7.4	1.6	1.08	1.57	2.61	
Average	**13.0**	**65.4**	**94.9**	**29.6**	**7.3**	**1.6**	**0.97**	**1.41**	**2.34**	**1.87**
2/13/2013 22:07	25.0	66.7	101.2	34.5	7.6	1.7	1.09	1.57	2.62	
2/13/2013 23:07	25.0	65.1	100.7	35.6	7.2	1.6	1.19	1.72	2.86	
2/14/2013 0:07	25.0	65.5	101.3	35.8	5.7	1.3	1.50	2.16	3.60	
2/14/2013 1:07	24.3	64.5	99.9	35.3	7.1	1.6	1.18	1.71	2.84	
2/14/2013 2:07	25.0	65.3	101.4	36.1	6.2	1.4	1.40	2.02	3.36	
2/14/2013 3:07	25.0	64.1	98.8	34.7	7.5	1.7	1.10	1.59	2.64	
Average	**24.9**	**65.2**	**100.6**	**35.3**	**6.9**	**1.5**	**1.23**	**1.77**	**2.95**	**1.98**
2/14/2013 10:07	34.7	66.0	91.9	25.9	4.5	1.0	1.38	2.00	3.32	
2/14/2013 11:07	34.0	66.2	96.2	30.1	4.3	0.9	1.67	2.41	4.01	
2/14/2013 12:07	36.0	67.2	97.8	30.6	3.3	0.7	2.20	3.18	5.29	
2/14/2013 13:07	36.7	67.2	89.6	22.3	3.4	0.8	1.55	2.25	3.74	
2/14/2013 14:07	37.7	66.6	94.9	28.3	3.3	0.7	2.05	2.97	4.93	
2/14/2013 15:07	37.3	66.3	96.0	29.7	3.8	0.8	1.87	2.71	4.50	
Average	**36.1**	**66.6**	**94.4**	**27.8**	**3.8**	**0.8**	**1.76**	**2.55**	**4.24**	**2.85**
2/15/2013 13:07	44.0	67.0	89.3	22.3	2.2	0.5	2.37	3.42	5.69	
2/15/2013 14:07	45.7	66.7	93.6	27.0	2.8	0.6	2.32	3.35	5.57	
2/15/2013 15:07	45.3	66.4	95.2	28.8	3.1	0.7	2.20	3.19	5.30	
2/15/2013 16:07	46.0	66.3	95.7	29.4	3.1	0.7	2.30	3.33	5.53	
2/15/2013 17:07	44.3	66.9	91.7	24.7	2.6	0.6	2.28	3.29	5.47	
2/15/2013 18:07	38.0	67.0	93.9	27.0	2.7	0.6	2.41	3.48	5.79	
Average	**43.9**	**66.7**	**93.2**	**26.5**	**2.7**	**0.6**	**2.31**	**3.34**	**5.55**	**3.73**

6.2.2 Estimated Cooling COP

Table 6.2 presents monitored data and estimated COP ranges estimated for site #6 during the cooling season. As with heating, the fourth fan speed, termed "powerful," is not used in the calculations. We have less confidence in the calculated values for cooling COP, as it was very difficult to recognize data that reasonably fit into typical parameters for estimating COP. There is little demand for cooling in central NH, and the units are typically operating under low part-load conditions. At many points, the ΔT values across return and supply air are very low. The systems may be idling due to reaching their minimum cooling thresholds, or they may be providing dehumidification only.

The shaded cells represent the data points we utilized to develop the average COPs. The estimated COP values represent the average of the calculated COPs presented in the shaded cells within each temperature bin.

As stated above, identifying enough appropriate data for accurately estimating cooling COP was not possible, and the estimated cooling COP was not used in the cooling savings estimates. Table 6.2 is presented only to provide an example of the data collected. The resulting estimated COP should **not** be considered accurate for use in predicting DHP cooling performance.

Table 6-2. Estimated Cooling COP

Site #6 Cooling COP - Mitsubishi MSZ-FE18NA										
Day	Outside Air Temp (°F)	Return Air Temp (°F)	Supply Air Temp (°F)	Delta T (°F)	Amps	kW	Low CFM 388	Med CFM 469	High CFM 628	Estimated COP
7/2/2013 10:07	61.0	75.2	69.6	5.6	0.97	0.21	3.23	3.90	5.22	
7/2/2013 11:07	61.7	73.9	68.7	5.2	1.18	0.26	2.46	2.97	3.98	
7/2/2013 12:07	63.0	72.9	68.0	4.8	0.97	0.21	2.78	3.36	4.50	
7/2/2013 13:07	63.0	71.9	67.2	4.7	0.96	0.21	2.70	3.26	4.37	
7/2/2013 14:07	63.0	71.0	66.7	4.3	0.97	0.21	2.46	2.97	3.98	
7/2/2013 15:07	63.0	76.9	71.6	5.3	0.96	0.21	3.08	3.72	4.98	
Average	**62.4**	**73.6**	**68.6**	**5.0**	**1.0**	**0.2**	**2.77**	**3.35**	**4.49**	**3.54**
7/4/2013 9:07	82.7	85.1	78.6	6.5	3.51	0.77	1.03	1.25	1.67	
7/4/2013 10:07	85.0	82.2	74.2	8.0	2.38	0.52	1.87	2.26	3.03	
7/4/2013 11:07	88.0	88.1	84.7	3.4	2.29	0.50	0.82	1.00	1.33	
7/4/2013 12:07	84.0	85.9	80.7	5.1	2.31	0.51	1.24	1.50	2.00	
7/4/2013 13:07	88.0	82.9	75.0	8.0	2.27	0.50	1.95	2.36	3.16	
Average	**85.5**	**84.9**	**78.7**	**6.2**	**2.6**	**0.6**	**1.35**	**1.64**	**2.19**	**1.91**
7/5/2013 12:07	89.0	76.5	71.6	4.9	2.07	0.46	1.32	1.60	2.14	
7/5/2013 13:07	90.0	75.5	71.0	4.5	1.83	0.40	1.37	1.65	2.21	
7/5/2013 14:07	90.0	74.7	70.5	4.2	1.84	0.40	1.26	1.53	2.05	
7/5/2013 15:07	91.0	73.9	70.1	3.9	1.90	0.42	1.14	1.37	1.84	
7/5/2013 16:07	91.0	76.6	71.3	5.3	2.18	0.48	1.35	1.63	2.19	
7/5/2013 17:07	89.3	92.0	89.7	2.3	1.58	0.35	0.82	0.99	1.33	
Average	**90.1**	**78.2**	**74.0**	**4.2**	**1.9**	**0.4**	**1.23**	**1.48**	**1.98**	**1.73**
7/6/2013 14:07	84.7	76.9	71.5	5.4	1.52	0.33	1.99	2.40	3.22	
7/6/2013 15:07	86.0	75.8	70.7	5.1	1.50	0.33	1.88	2.27	3.04	
7/6/2013 16:07	85.0	75.7	70.5	5.3	1.58	0.35	1.86	2.24	3.01	
7/6/2013 18:07	81.7	86.9	80.8	6.1	2.06	0.45	1.66	2.01	2.69	
7/6/2013 19:07	78.3	88.4	84.0	4.4	1.72	0.38	1.44	1.74	2.32	
Average	**83.1**	**80.8**	**75.5**	**5.3**	**1.7**	**0.4**	**1.75**	**2.12**	**2.83**	**2.47**
7/7/2013 18:07	79.7	91.4	86.1	5.3	1.46	0.32	2.04	2.47	3.31	
7/7/2013 19:07	77.7	89.8	84.7	5.2	1.26	0.28	2.29	2.77	3.71	
7/7/2013 20:07	73.7	86.6	78.4	8.2	1.57	0.35	2.93	3.54	4.74	
7/7/2013 21:07	73.0	89.0	84.0	5.0	1.58	0.35	1.79	2.16	2.90	
7/7/2013 22:07	72.7	88.2	85.0	3.2	1.34	0.29	1.34	1.62	2.18	
7/7/2013 23:07	72.7	84.9	77.9	7.0	1.27	0.28	3.08	3.72	4.98	
Average	**74.9**	**88.3**	**82.7**	**5.7**	**1.4**	**0.3**	**2.24**	**2.71**	**3.63**	**2.86**
7/15/2013 1:07	69.3	77.6	72.2	5.4	1.42	0.31	2.14	2.59	3.47	
7/15/2013 2:07	68.0	76.4	71.6	4.8	1.20	0.26	2.25	2.72	3.64	
7/15/2013 3:07	68.0	75.5	71.1	4.4	1.14	0.25	2.14	2.59	3.46	
7/15/2013 4:07	67.3	81.4	75.8	5.6	0.91	0.20	3.44	4.16	5.57	
7/15/2013 6:07	72.7	84.7	79.6	5.2	0.91	0.20	3.18	3.84	5.14	
7/15/2013 7:07	77.7	82.2	75.2	7.0	1.40	0.31	2.78	3.36	4.50	
Average	**70.5**	**79.6**	**74.2**	**5.4**	**1.2**	**0.3**	**2.59**	**3.13**	**4.20**	**3.31**

6.3 Estimated COP Conclusions

As previously discussed, COP is difficult to measure in the field, and it is particularly difficult to measure for DHPs. However, we have been able to estimate the COP under a variety of temperature ranges for two of the monitored systems. Due to the factors discussed in Sections

6.1 and 6.2, we have more confidence in the estimated heating COP values than we do in the cooling values.

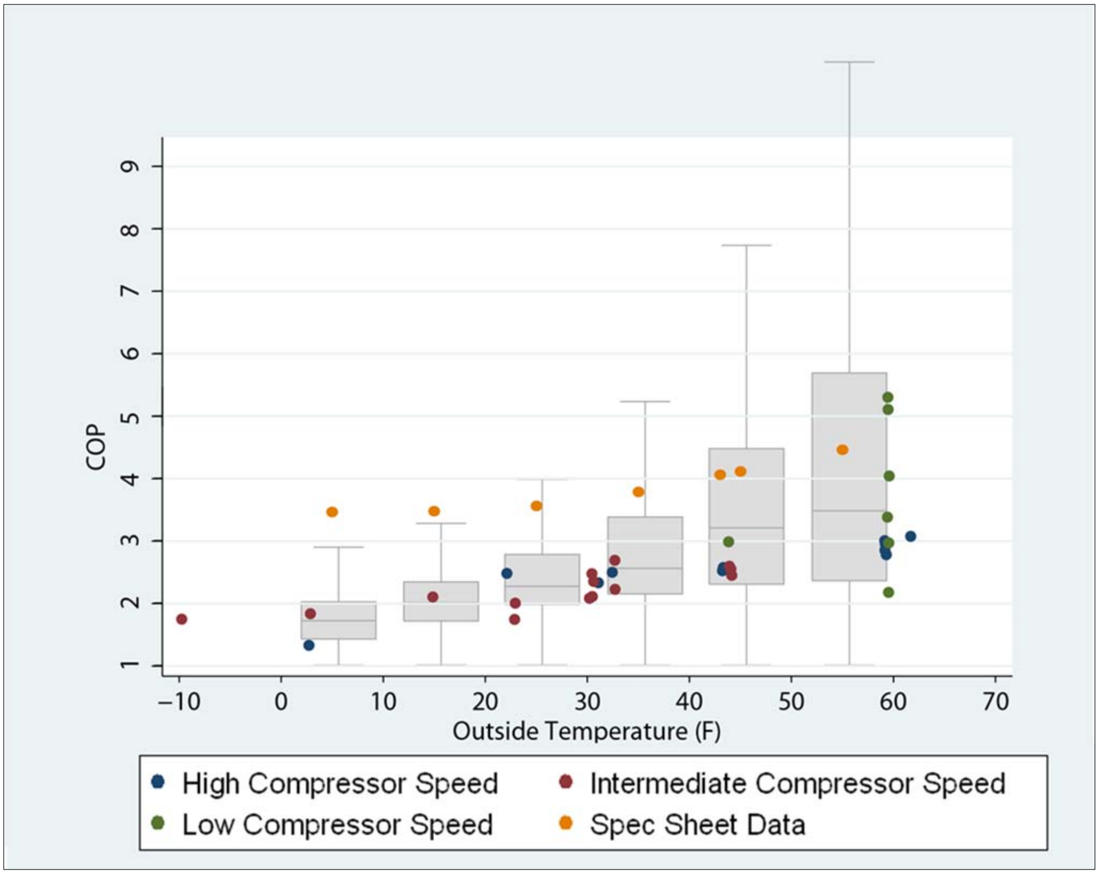
Heating COP

It is estimated that the heating COP ranges from approximately 1.4 @ 11°F to 3.7 at 44°F. This is in line with calculated COP estimates from the recent Ecotope/NREL studies, which tested DHPs in situ and under laboratory conditions and resulted in estimated average heating COP ranges of 2.3 at 7°F to 3.8 at 47°F associated with several microclimates in the Pacific Northwest. The lower COP estimated at cold temperatures for this study may be associated with the inability to identify the actual fan speed, as discussed in section 6.2.1. All monitored/calculated COP values are somewhat lower than the manufacturer published data. The DHPs studied are capable of heating at both higher and lower temperatures than were covered by these findings, but we did not observe enough operation at the extreme ranges to estimate COP. At 20°F, which represents the average January temperature in Laconia, NH, we calculated a COP ranging from 2.0 to 2.4.

In our opinion it would not be accurate, or fair, to use these, or any, field-calculated performance values in comparison with other models' published COP figures. However, it is fair to assume that actual in situ DHP heating COP levels, for any given temperature range, are somewhat lower than those measured under controlled laboratory steady-state conditions.

Figure 6.1 is a chart from the NEEA/Ecotope field study report which plots the calculated COP with the manufacture published COP ratings (spec sheet data.) For all corresponding data points the field measured COP is lower. The fact that the differential is relatively constant suggests that adjustment factors could be applied.

Figure 6.1. NEEA/Ecotope Field Study COP Values



Cooling COP

As stated previously, we have much less confidence in using the logged data to estimate cooling COPs. The systems are operating at low part-loads, and attempts to calculate COP led to widely divergent values. The data in Table 6-2 is presented to illustrate what data was collected and how it is used to calculate COP. It represents the most consistent cooling data from our sample. The NEEA/Ecotope laboratory steady-state testing of the same model DHP reported cooling COPs in the 3.0–5.0 range when OATs were 80°F–90°F and supply air temperatures were ranged between 60°F–65°F.

6.4 Defrost Cycle

All heat pumps incorporate a defrost cycle to protect the outdoor components from damage and to allow heating to continue during cold OATs. Under certain operating conditions, dependent on both temperature and humidity levels, frost accumulates on the heat exchanger coil, and a defrost cycle is triggered to melt accumulated frost. The unit's efficiency during defrost operation is taken into account in HSPF calculations, but not in simple COP calculations or COP performance curves.

Although the exact defrost cycle operation varies by manufacturer, a reverse cycle strategy is used for all DHPs. When operating in defrost mode, warm refrigerant vapor is delivered by the compressor to the outdoor condenser/evaporator coils, and the indoor fan is turned off, as heat cannot be delivered. The defrost cycle reduces the overall performance of the heat pump because the compressor is using energy but no heat is being delivered.

The National Renewable Energy Laboratory (NREL) in association with Ecotope tested a Mitsubishi FE12NA model similar to those encountered in this project and determined COP penalties under certain conditions. The results of the NREL testing are presented in Table 6-3. The table clearly shows that the penalty increases as the OAT falls, but also that the penalty varies considerably with fan speed. As DHPs are used for heating below -3°F (some current models, including the one tested here, and several of the models monitored in this study can deliver heat at OATs approaching -20°F), the defrost penalty continues to increase.

Table 6-3. NREL Tested Defrost Penalty

OAT	Fan Speed	Defrost Cycle Time (Min)	Defrost Time	Defrost COP Penalty
35°F	M	90	3.3	1.4%
27°F	H	79	2	1.3%
17°F	H	91	3.8	1.8%
17°F	M	90	3.3	1.4%
7°F	H	31	2.2	4.7%
7°F	L	23	2.5	3.9%
-3°F	H	47	10.2	11.5%

7 HOMEOWNER DHP OPERATIONAL PRACTICES

A stated goal of this project is to determine common operational practices of homeowners who heat and/or cool with DHPs. Because DHPs are typically operated by a remote control, not dissimilar to a television remote control, human intervention likely plays a larger role than is experienced with conventional central systems that are controlled by automatic thermostats.

For the systems we monitored, we interviewed the homeowners twice, once during logger installation, and a second time following review of the logged data. This section summarizes our findings.

7.1 General Heating and Cooling Usage

With one exception, all of the homeowners use their systems for both heating and cooling. This includes those users who purchased their DHPs with the original intention of cooling only. The exception is one homeowner who installed their DHPs principally to provide cooling and dehumidification for reasons associated with a family medical issue. They did use the DHPs for heating during the study period but have since reverted to a combination of kerosene, propane, and wood pellet usage, in an effort to lower electric bills. They did not perform a comparative cost analysis of all of the heat sources to determine the net effect. We did monitor one of their DHPs during the time period they used it for heating, and it was operating within anticipated performance levels.

One other homeowner reported using one of two DHPs for cooling only. The DHP used for cooling only is installed in a bedroom that receives enough heat from the other living spaces.

7.2 Control Systems

The DHPs we studied are all supplied with a handheld remote control as standard equipment. An optional wall-mounted electronic programmable thermostat is available that will allow the DHP to be controlled much like a conventional system.

None of the participants had elected to install a wall-mount thermostat, and all controlled their systems via the remote control. Operational trends include:

- **Setpoints** – All of the participants chose to select setpoints during active heating and cooling. Exceptions included limited time periods when only dehumidification was selected, and when systems were turned off. The setpoints varied widely from 62°F – 72°F for heating, and from 70°F – 76°F for cooling.
- **Fan speed settings** – The DHPs installed offer the option of selecting an automatic fan speed setting, or the selection of three or four speed settings. Predominantly, the participants select the auto setting. Exceptions include:
  - A site #1 participant reported that they typically selected medium fan speed during the heating season. This was also the participant who expressed dissatisfaction with the system performance.

- Selecting a low or medium setting during cooling season when the user believes that the cooling demand is low.
- Selecting a reduced setting due to airflow noise when the fan is operating on high or powerful.
- All participants were asked if comfort issues associated with cool air resulted in changing fan speeds or other adjustments. There were no comfort issues associated with air flow.

- **Set-back & set-forward** – None of the participants utilize an automatic set-back or set-forward feature for their DHPs. More than one-half stated that they manually selected a lower heating setpoint for nighttime operation, and/or when the home is unoccupied. Only one reported manually selecting a high cooling setpoint for unoccupied periods.
- **Interactions with other system controls** – Most of the participants report selecting the DHP(s) as their primary heating system. All select the DHP(s) as their primary cooling system. For heating, participants with central thermostatically controlled heating systems, typically set the central system thermostat at a lower setpoint than the DHP setting. This varies from a single degree (this thermostat is remote from the DHP) to over a 10°F differential. It is important to note that this procedure was learned through experience, as in most cases the participant initially intended the system as supplementary but with experience came to rely on the DHP as the primary heating system. Exceptions to this include:
  - The one participant (site #1) who expressed dissatisfaction with the DHP has recently installed electric baseboard heating and relies on kerosene, propane, and electric space heaters. We have recommended that he reconnect with the installer to investigate the operation of the DHP.
  - The participant who installed the systems for health reasons has reverted back to relying on conventional heating systems after being "shocked" by the first electric bill during the heating season. We suggested that they consider all fuel costs in assessing which systems to prioritize. The DHP was used for heating during the study period.
  - Many of the participants have a variety of space heaters: kerosene direct-vent units, propane unvented heaters, wood/pellet stoves, and propane fireplaces. In many cases these are selected based on local space needs, while the DHP remains the primary heat source.

8 HOMEOWNER SATISFACTION

All but two of the participants were very satisfied with the performance of their DHP systems, both from comfort and energy usage standpoints. The following subsections summarize the reported satisfaction level of the participants.

8.1 Heating Performance

All but two of the participants were extremely satisfied with the ability of the DHPs to heat the space in which they were installed, and in many cases the entire house. In most cases, heating performance surpassed expectations, as participants learned to rely on the DHPs as a primary rather than supplementary heat source. Most of the participants currently utilize conventional heat sources when OATs are very low and the DHPs have difficulty maintaining setpoints. The two exceptions include:

- **Site #1 –** The participant has not been satisfied with the heating performance of the DHP installed. The installing contractor has returned to assess the operation of the unit, but he reported no issues. Our monitoring did reveal somewhat erratic results, but it is difficult to discern whether they are associated with system faults or control selections. The participant relies on several types of space heaters and during December of 2012 installed electric baseboard to contribute to the heating of the home. The DHP is not a cold-climate unit. It is also notable that the unit is very oversized (30,000 Btu/h) for the 280 ft^2 living room in which it is installed. The expectation was that the unit would heat most/all of the 1,500 ft^2 house, which was constructed in 1950. Within the last 10 years the home has been insulated with blown-in cellulose, was caulked and weather-stripped, and had its windows replaced.
- **Site #4 –** The participants installed the DHPs primarily to provide cooling and dehumidification in order to maintain recommended indoor conditions related to a family member's illness. Although they were satisfied with the ability of the units to heat their home, they have recently reverted to kerosene, propane, and biomass pellet fuel after experiencing "sticker shock" when reviewing the associated electric bill. It was suggested that they compare the cost of all fuels to assess the net effect. The metered DHP was used for heating during the study period.

8.2 Cooling Performance

All participants expressed satisfaction with the cooling performance of the installed systems. As the monitoring revealed, with the central NH climate, the systems are sized for heating loads, and thereby they provide adequate cooling while operating under part-load conditions for the vast majority of the time. Some of the participants, notably sites #1 and #2, use their DHPs very little for cooling.

8.3 Other Comfort Considerations

A stated goal of the project was to investigate various comfort issues associated with fan-forced heating and cooling. Of particular interest was the sensory effect that moving temperate air might have on occupant comfort, and whether or not there were adjustments made to operating modes such as fan speed or louver position. All participants were asked about overall comfort, and particularly if the airflow from the DHP had any negative impacts on comfort. All responded that they had experienced no negative comfort issues associated with the units, and that they have not needed to adjust air flow, furniture placement, etc.

Another comfort issue is sound. We did receive comments that, at times, occupants would use the remote control to switch from the auto fan speed and manually select a lower fan speed, as the sound level from higher fan speeds was an annoyance. One of our firm's conference rooms is served by a DHP, and the staff makes similar adjustments during meetings.

9 ADDITIONAL SITE-MONITORED DATA

This section presents summary tables and graphs of the data that was logged at the nine sites as well as general observations related to the presented data.

Recording of minimum and maximum amperage, which has been converted to kW, reveals several aspects related to system operation:

- During all seasons, the systems are off, or on standby, for some time periods.
- During the heating season, the minimum load may be associated with both user operational choice and freeze protection when OATs drop below the minimum rated value.
- As expected for the regional climate conditions, heating loads are higher than cooling loads.
- No systems were recorded to operate at the maximum rated load. However, all systems, with the exception of site #1, were reported to satisfy heating demands for the spaces in which they were installed, except when temperatures dropped below the rated full-output capabilities.
- The power logger for site #6 stopped recording data during the last month of data logging.
- The DHP installed at site #1 has a rated heating output nearly 50% higher than the systems installed at site #s 6, 7, and 8, yet the rated and monitored load is only approximately 12% higher. However, the systems installed at sites #6, #7, and #8 are rated to operate at much lower OATs than is the system installed at site #1. This and the fact that there is no single mandated methodology for calculating rated output may explain the inconsistencies.

9.1 Metered Site Data – Comparison of Usage Patterns

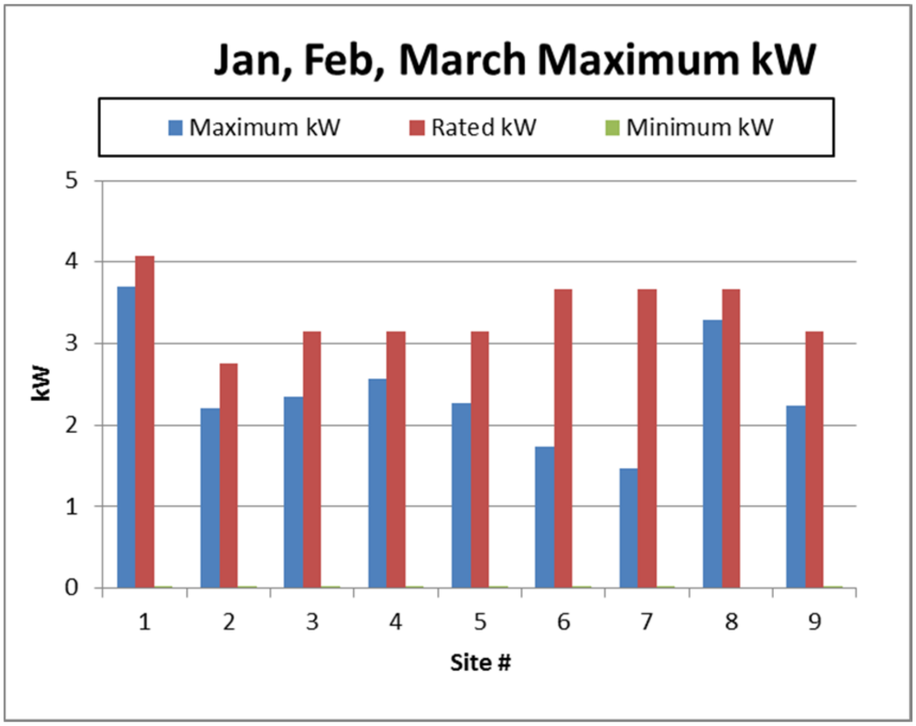
The following tables and charts present additional data associated with the usage patterns for the monitored systems, as well as utility supplied usage data spanning a 2-year period. Table 9-1 presents the maximum rated and recorded loads.

Table 9-1. System Seasonal Maximum and Minimum Loads (kW)

	April - May		June, July, August		September		Jan, Feb, March		
Site #	Maximum kW	Minimum kW	Maximum kW	Minimum kW	Maximum kW	Minimum kW	Maximum kW	Minimum kW	Rated kW
1	2.53	0.23	1.26	0.23	1.07	0.00	3.70	0.00	4.1
2	2.19	0.01	1.01	0.00	1.38	0.00	2.21	0.01	2.8
3	1.64	0.01	1.01	0.03	1.51	0.01	2.34	0.01	3.2
4	2.13	0.00	1.09	0.00	0.52	0.00	2.56	0.00	3.2
5	1.79	0.01	1.16	0.03	1.28	0.03	2.27	0.01	3.2
6	1.73	0.17	1.16	0.18	N/A	N/A	1.73	0.00	3.7
7	1.57	0.04	1.82	0.05	1.39	0.04	1.47	0.02	3.7
8	3.10	0.05	2.44	0.06	1.33	0.00	3.29	0.00	3.7
9	1.83	0.00	0.95	0.00	1.09	0.00	2.23	0.01	3.2

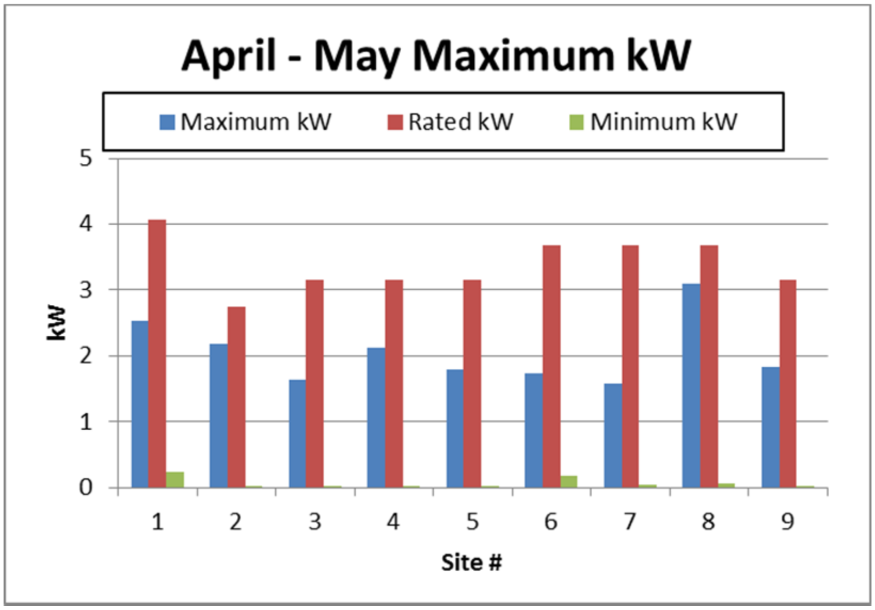
The maximum winter heating season load is illustrated in Figure 9-1. It demonstrates that several of the systems operate near their full rated loads during the heating season. The minimum loads are barely registering, as the systems shut down at extreme low temperatures to protect against compressor damage, and/or the user selects an alternate heating source during certain temperature conditions.

Figure 9-1. Heating Season Maximum Load



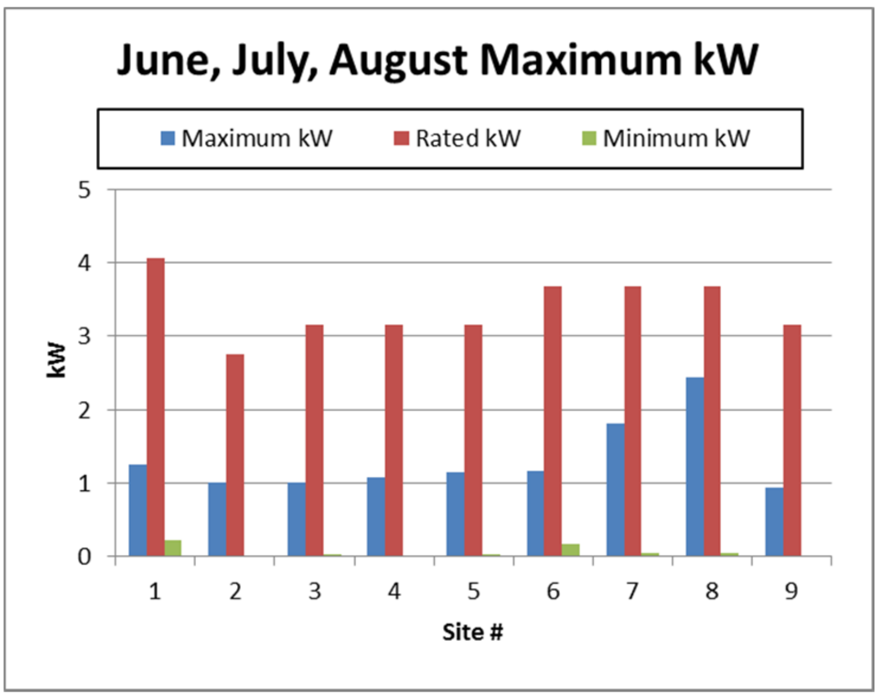
For the spring shoulder season, presented in Figure 9-2, the maximum load drops significantly for most of the systems. Nearly all of this load is heating, rather than cooling, load.

Figure 9-2. Shoulder Season Maximum Load



For the summer cooling season, the maximum load decreases dramatically, as shown in Figure 9-3, for all of the systems, as central NH is a heating-dominant climate.

Figure 9-3. Maximum Load Summer Cooling Season



Energy usage (kWh) was also recorded and is presented in Table 9-2. Notes associated with this table include:

- Data logging began in early February for sites #8 and 9.
- Data logging could not begin for site #6 until March due to participant scheduling of site visits.
- At site #4, although the occupants reported discontinuing use of the DHP for heating due to "sticker shock" associated with the electric bill, the data reveals that the system was utilized for heating during the monitored period.
- Sites #2, 3, 5, 7, and 8 have an additional DHP installed. Site #4 has two additional DHPs installed. Therefore, the usage should not be viewed as the total heating or cooling usage for those sites.
- Heating is the dominant driver of the energy usage for all nine systems.

Table 9-2. DHP Monitored Usage

Monthly DHP Nine Month Consumption (kWh)										Total
Location	January	February	March	April	May	June	July	August	September	
Site #1	543	558	250	176	183	183	200	185	145	2,424
Site #2	458	775	599	286	156	70	15	6	20	2,386
Site #3	281	447	345	251	159	131	213	171	154	2,151
Site #4	251	312	230	161	111	52	75	25	28	1,245
Site #5	423	347	198	70	95	149	214	119	132	1,746
Site #6	0	0	495	334	173	154	180	7	75	1,418
Site #7	721	584	563	416	247	187	274	222	145	3,362
Site #8	0	710	632	337	129	81	100	63	61	2,114
Site #9	0	623	414	265	48	18	119	19	38	1,544

Figure 9-4 presents the utility-monitored energy usage for the participant sites for an 8-month period (Jan–Aug) for 2012 and 2013. Here are the notes associated with this table:

- According to the national weather service, for Plymouth NH, there was a 13% increase in heating degree days and a 3% decrease in cooling degree days, from 2012 to 2013. Figure 9-5 presents a similar comparison but with the heating season weather normalized for the 2 years for the time period from the start of metering through May. The result is a slight change in total usage values.
- Site #1 was not occupied for most of 2012.
- Site #8 changed ownership at the time of the DHP installation.

Figure 9-4. 2012–2013 Utility Metered Data

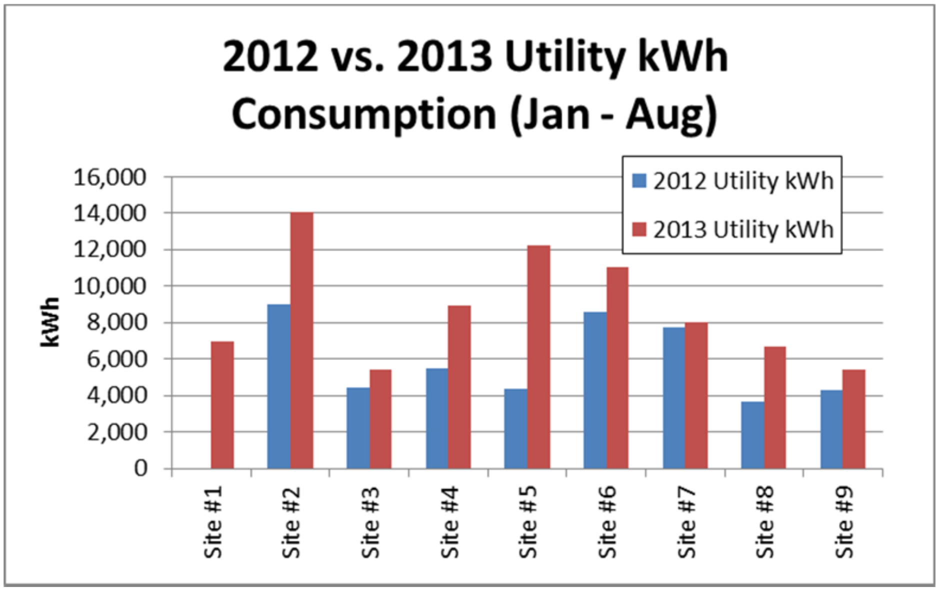


Figure 9-5. 2012–2013 Utility Metered Data – Weather Normalized (2 years)

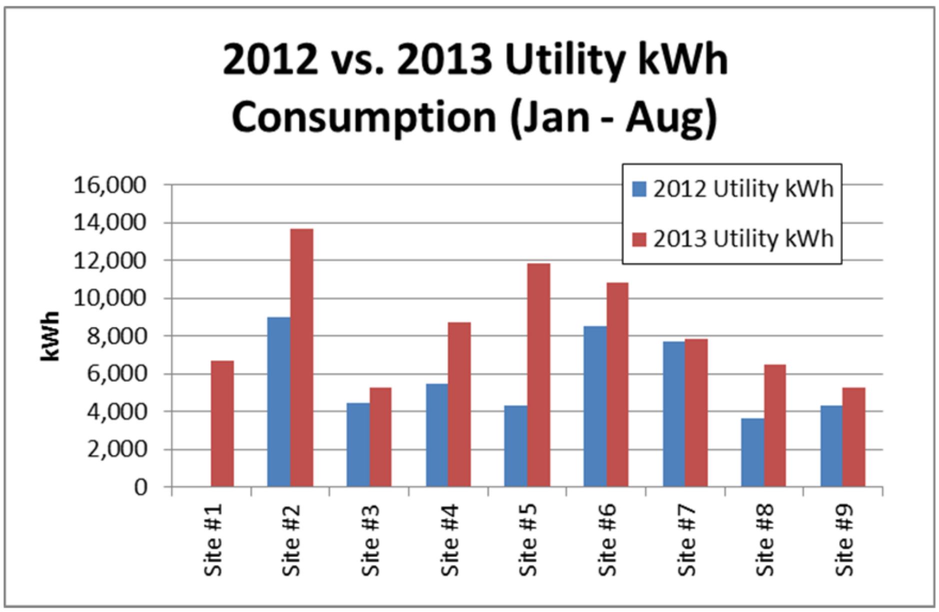
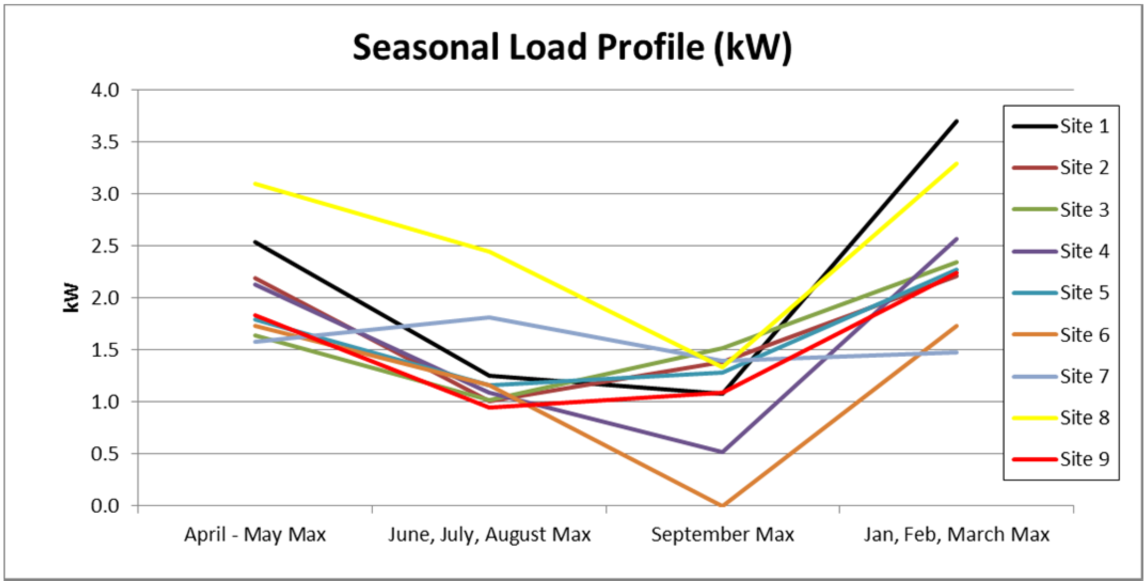


Figure 9-6 charts the profile for peak recorded load for each season. The seasonal load profile for the nine sites is largely consistent. Site #6 includes no September data due to logger failure.

Figure 9-6. Load Profile



9.2 Representative DHP Usage Patterns

This section presents data from a single monitored site in order to illustrate the relationship of the DHPs' load to weather conditions and overall utility metered usage data. We chose site #3 as it represents an installation that has fewer variables and is operated in what might be considered standard fashion by the occupants. The DHP is installed in the living room of an in-law apartment, and it is the only DHP in the apartment. A second DHP is installed in the main home, and both are served by a central boiler. The occupant typically selects setpoints around 72°F for heating and 75°F for cooling, selecting auto operation of the fan.

Figure 9-7 charts DHP usage along with total utility usage data for the same time periods. It is clear that the DHP is a major contributor to the overall load. It is significant to note that the main house has a second DHP and the utility-metered data is associated with the entire residence.

Figure 9-7. Utility Meter and DHP Usage Data

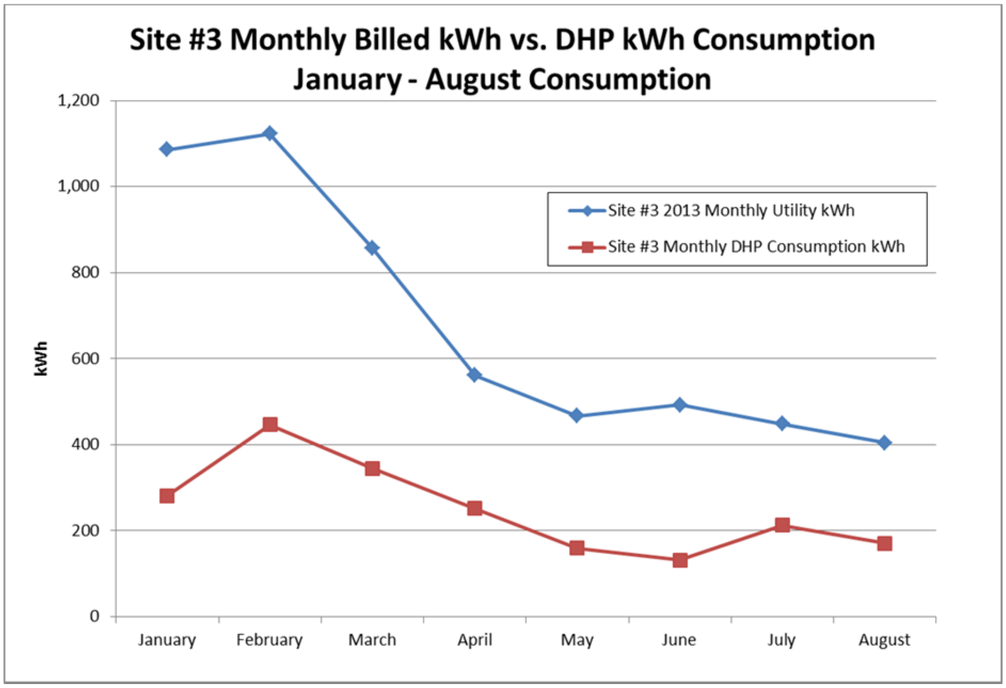


Figure 9-8 presents the same data overlaid with the heating and cooling degree days. This illustrates how closely associated both the DHP and total usage are related to weather conditions.

Figure 9-8. DHP and Total Usage Charted with Degree Day Totals

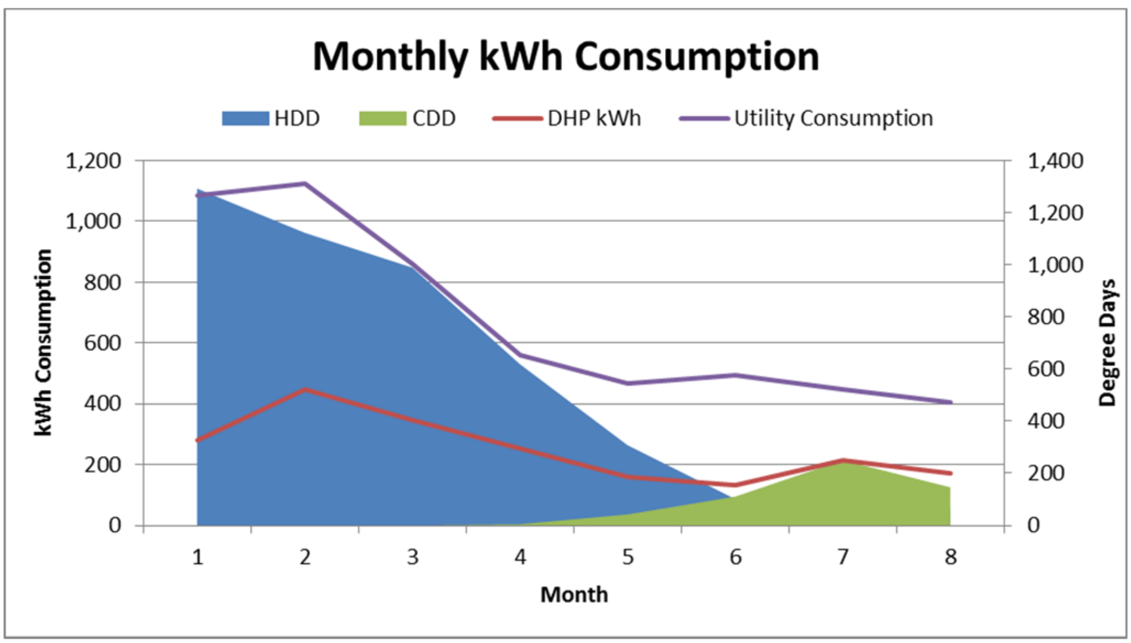


Figure 9-9 illustrates how the systems modulate in order to deliver consistent temperature. The OAT is cold and dropping rather steadily over 2 days from 20°F to just below 0°F. The supply air temperature is between 80°F and 90°F for nearly the entire period. The amperage, however, ranges from 2 to 7 amperes and is constantly modulating. A standard heat pump, or even a boiler, would cycle many times under these conditions.

Figure 9-9. Load Profile Charted with Supply and OATs

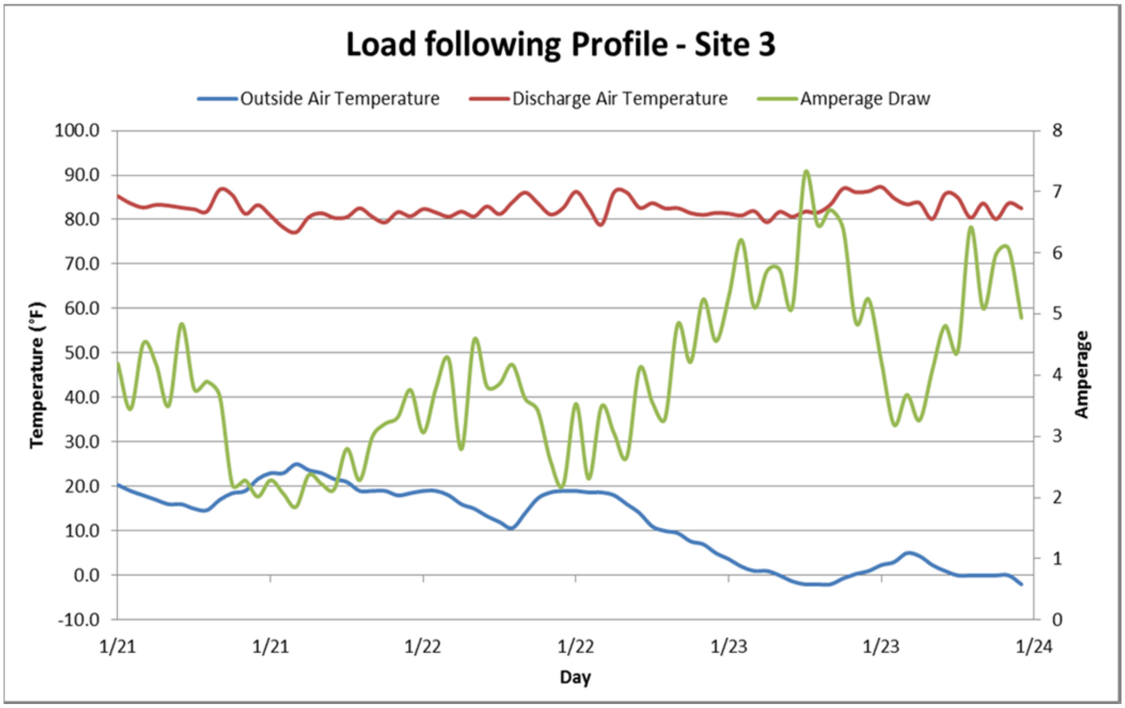


Figure 9-10 charts a single winter day's DHP load shape against the OAT and utility data from newly installed NHEC "smart" meters. The load on the DHP can clearly be seen as responding to the temperature changes. The variation in load is not as dramatic as the OAT swing due to the ability of building mass to moderate such swings.

Figure 9-10. January Single Day Load Shape

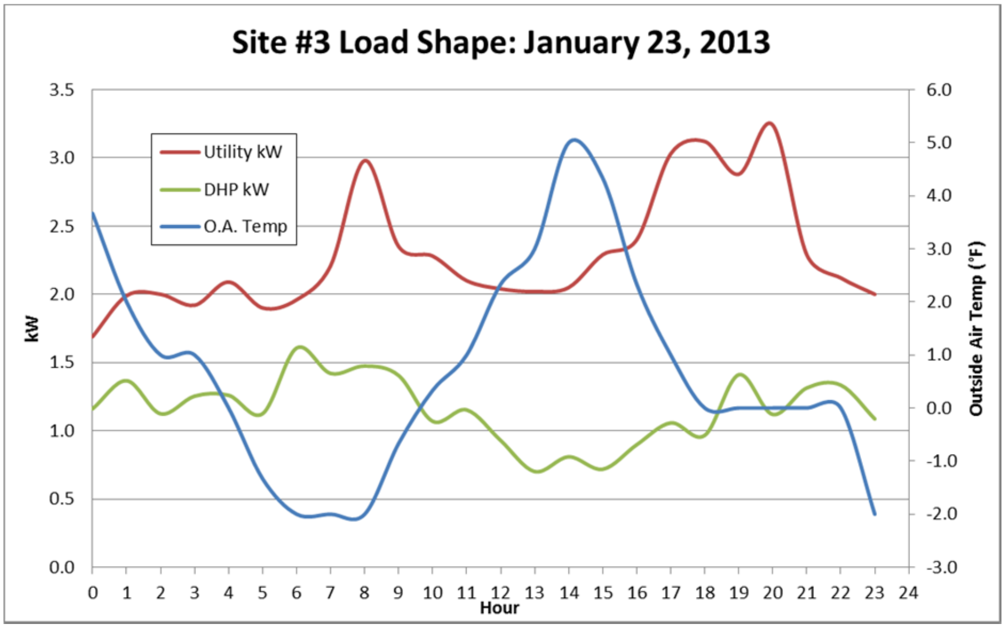


Figure 9-11 charts the system load in amperes along with the OAT for a 7-day period in January where the OAT ranged from -5°F to 25°F.

Figure 9-11. Heating Season Load and Temperature over 7 Days

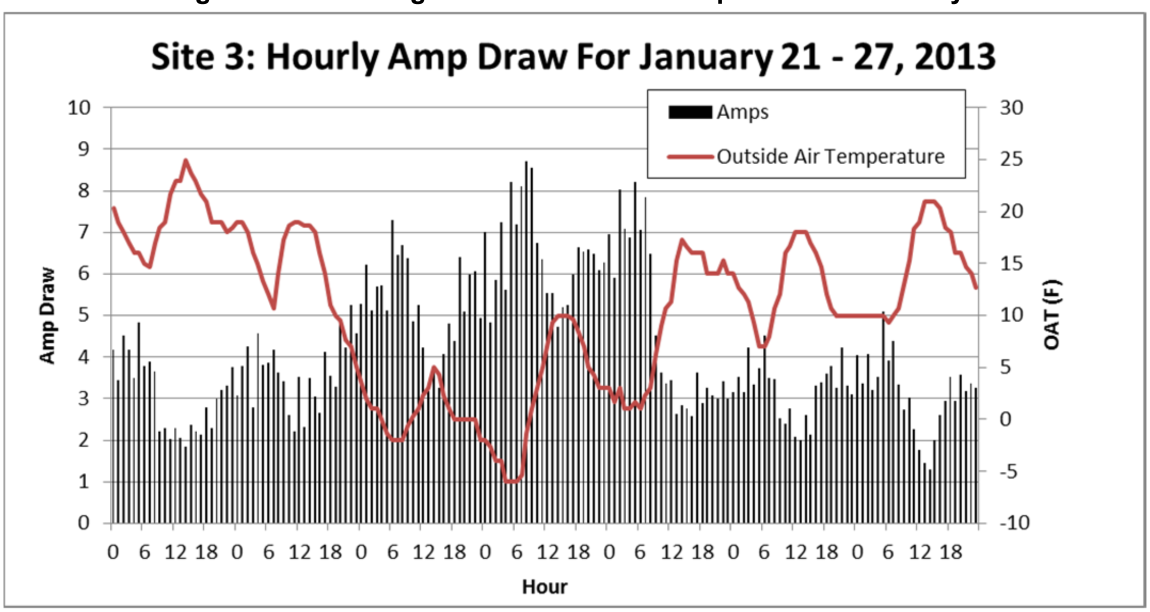
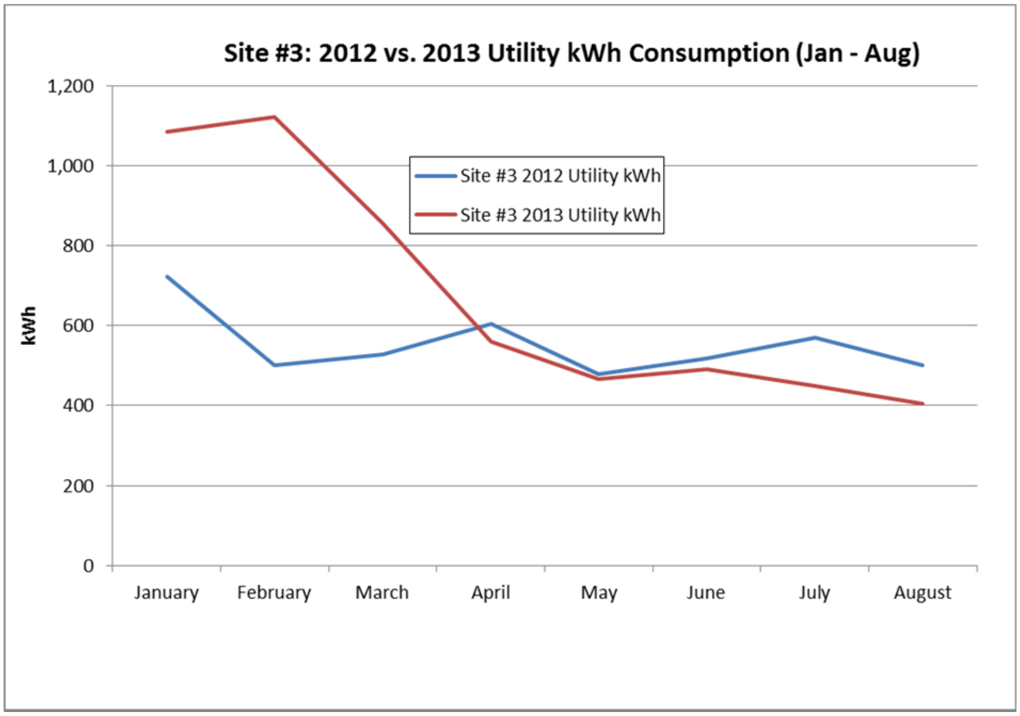


Figure 9-12 charts the utility monitored energy usage for 2012 and 2013. Two important factors are illustrated:

1. The effect of switching fuels from heating oil to the DHPs (one in the apartment and one in the main house) during the heating season

2. A small decrease in usage during the cooling season, as the DHPs replaced less efficient window A/C units

Figure 9-12. Site #3 Utility Metered Usage



10 Consolidated Edison Impact Evaluation

Prior to the data logging performed for this study, Consolidated Edison (Con Edison) performed an impact evaluation of its Residential Electric HVAC Program. The evaluation included the monitoring of twenty-five DHPs installed in residences. ERS led the evaluation, with Navigant Consulting performing the M&V for the DHPs. Con Edison has graciously offered to share the data and results of the evaluation in order to assist in assessing DHP performance.

As of this writing, Con Edison has not yet approved the final study results, and we are unable to fully disclose the results. However, the following bullet points summarize methodologies and conclusions that can be shared at this time. When the full report becomes available, it will be included with the appendices.

10.1 Methodology

- **Phone survey** – Participants were asked to provide a schedule of A/C use depending on time of the day for hot, warm, and mild temperature types of days. The evaluation team determined different day types which were dictated by the high and average outside air temperature (OAT) for the day. For each day type, the participants provided a schedule with temperature setpoints for their units. The responses were used to predict run time for sampled sites.
- **On-site M&V approach** – The twenty-five DHP sites were monitored from July 15 through March 1, 2013. Power and indoor temperature were monitored.
- Billing data was disaggregated to estimate DHP usage.
- Ecotope NEEA study results were utilized to assign/extrapolate performance characteristics for the DHPs.
- Cooling energy model for a typical meteorological year (TMY) using TMY3 data in EnergyPlus software
- Operational modes were assigned based on performance curves and temperature data, and run time was calculated for each unit using the operational modes assigned and categorized by run time and average daily temperatures.
- Total kWh savings was calculated for each data point using the benchmark power curve and assuming a SEER 13 baseline.
- Hourly savings values were used to produce total kWh savings for a typical year.

10.2 Conclusions

- Savings are estimated to be an average 756 kWh per DHP.
- Eighty percent of the savings in Con Edison territory are associated with cooling. Heating usage is less than anticipated, but this may be due to programmatic focus on cooling, as well as the availability of both natural gas and district steam heating.

- Actual run times are shorter than predicted when utilizing the program's deemed EFLHs value.
- The verified peak-demand impact of DHPs is lower than program deemed predicted impacts due primarily to the systems operating more than anticipated under part-load conditions as they modulate in response to indoor and outdoor temperatures. An additional contributing factor was determined to be the unpredictable nature of users heating and cooling single spaces on an as-needed basis, in contrast with central A/C systems, which are typically used to meet a certain thermostat setpoint for most or all hours of the day.
- Program administrators should consider incentivizing cold-climate DHPs separately from other heat pumps and specifically targeting participants who will use them for heating.

Upon release of the evaluation report, with Con Edison's permission, summary results will be added as an appendix to this report.

11 Savings Calculations Methodologies

The Phase I report for this project published the algorithms currently used by several efficiency programs for calculating savings for DHPs, as well as a recommended algorithm. All of the algorithms utilize SEER and HSPF along with estimated full-load hours. The recommended algorithm from the phase I report is presented here for convenience:

The recommended algorithm for computing savings per DHP is:

$$\Delta kWh = \left(\frac{kBtu/hr_{base,c}}{SEER_{base}} - \frac{kBtu/hr_{ee,c}}{SEER_{ee}}\right) \times EFLH_c + \left(\frac{kBtu/hr_{base,h}}{HSPF_{base}} - \frac{kBtu/hr_{ee,h}}{HSPF_{ee}}\right) \times EFLH_h + \Delta kWh_{duct}$$

where,

ΔkWh	= Gross annual energy savings
ΔkWh_{duct}	= Gross annual energy savings from elimination of duct losses
$kBtu/hr_{base,c}$	= Nominal capacity of baseline unit for cooling
$kBtu/hr_{base,h}$	= Nominal capacity of baseline unit for heating
$kBtu/hr_{ee,c}$	= Nominal capacity of energy-efficient unit for unit for cooling
$kBtu/hr_{ee,h}$	= Nominal capacity of energy-efficient unit for heating
$SEER_{base}$	= Seasonal energy efficiency ratio of baseline unit
$SEER_{ee}$	= Seasonal energy efficiency ratio of energy-efficient unit
$HSPF_{base}$	= Heating season performance factor of baseline unit
$HSPF_{ee}$	= Heating season performance factor of energy-efficient unit
$EFLH_h$	= Equivalent full-load hours for heating
$EFLH_c$	= Equivalent full-load hours for cooling

At this point in time there is no identifiable reason to adopt a different algorithm. However, there are three factors which should be considered:

1. As discussed, SEER and HSPF are calculated for several regions but are published for only one. Calculations would be more accurate if manufacturers would supply ratings for the efficiency program service territory. HeatCalc, a downloadable spreadsheet tool, includes a calculator to adjust published HSPF ratings for the local climate. As an example, a published HSPF of 10.6 is adjusted to 7.6 for the Boston area.[14] However, the HeatCalc adjustment factors were developed before the market availability of cold-climate DHPs and they include assumptions that electric resistance heat is used below 17°F. As such, this adjustment factor unfairly penalizes cold-climate DHPs. For this study we decided to

[14] HeatCalc software tool. www.eia.gov/neic/experts/heatcalc.xls

adopt a 10% penalty against the rated HSPF to adjust for the northern New England climate, and the results correlate with the metered/calculated heating COP. However, more research is needed to obtain accurate ratings for individual regions.

2. Estimating equivalent full-load hours (EFLH) is difficult for DHPs because they constantly modulate output, and thereby power demand over a wide range. And, perhaps more importantly, they are typically controlled by homeowner use of a remote control rather than a conventional thermostat. EFLH estimates should be adjusted with the knowledge gained from research efforts and impact evaluations.

3. **Fuel switching** – The above algorithm assumes that the baseline is either another heat pump or electric resistance heat. Fuel switching is now being supported in some programs and is expected to be progressively allowed in other jurisdictions. For fuel switching, it is advisable to utilize the same algorithm, substituting the appropriate performance factors and ratings. For oil-fired boilers and furnaces for example, the typical AFUE (averaging 80%–85% for existing systems), adjusted for system losses would be utilized. Adjusting for jacket and distribution losses results in an overall average system efficiency for oil-fired systems of approximately 78%.

DHPs offer both user manual control and local zone heating and cooling. The factors introduced are widely discussed in this report and offer challenges for any algorithm-based approach. As much as possible, evaluation results should be used to apply adjustment factors to algorithmic methodologies.

12 CONCLUSIONS AND RECOMMENDATIONS

Given the small participant sample, this study provides a wealth of information regarding the performance of DHPs in a heating-dominant climate; the performance of cold-climate models rated to operate at temperatures well below 0°F; and the usage patterns associated with systems that incorporate features that allow homeowners convenient direct control over their operation. Included in the overall knowledge gained are purchasing decisions, comfort decisions, operational choices, savings perceptions, summer load impacts, and system performance characteristics. Of particular interest is the observation that participant operational usage of the systems evolved following their initial experiences with the systems, as most owners who initially considered their DHPs as supplemental heating systems began to rely on their systems as primary heat sources.

We also are making several recommendations associated with M&V approaches and using the knowledge gained through recent and current research efforts, as well as through impact evaluation, to make better use of DHP ratings for predicting savings.

12.1 Conclusions

The conclusions resulting from this primary research are summarized in the subsections that follow.

12.1.1 Significant Heating Savings are Achieved Compared with Electric and Fuel Oil Baselines

The monitoring of heating performance for 4 months of the heating season and extrapolating weather-normalized performance for an entire heating season demonstrates that the systems are capable of delivering significant energy and cost savings in the New England climate, as shown in the tables below. The estimated savings for the eight cold-climate DHPs average approximately $832 per heating season compared with an electric resistance heat baseline, and approximately $398 compared with a standard efficiency air-source heat pump (ASHP). Savings associated with an oil heat baseline, which is the actual baseline for a majority of the participant sites, are also impressive at an average of $613 per heating season (September 15 – May 31).

Tables 12-1 through 12-3 present the weather-normalized estimated heating season energy usage and savings compared with the three baselines:

1. Electric resistance baseboard heat
2. An ASHP that meets minimum federal efficiency standards
3. An oil-fired boiler with an average system efficiency (includes distribution losses) of 78%

A weighted average savings is also calculated for each baseline at 1 ton of heating (12,000 Btu) to allow for the simple calculation of average savings for different sizes of DHPs.

Potential savings associated with a natural gas baseline were also estimated. Due to the current price of natural gas, the savings are small. The estimated natural gas savings are presented in Section 4.3.5.

Table 12-1. Monitored DHP Normalized Heating Season Usage & Savings Compared with Electric Resistance Baseline

Site #	1*	2	3	4	5	6	7	8	9	Average* (Sites 2-9)
System mfg.	Fujitsu	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	Mitsubishi	N/A
Model	30RLX	FE09NA	FE12NA	FE12NA	FE12NA	FE18NA	FE18NA	FE18NA	FE12NA	N/A
Heat cap. (Btu/h)	37,500	10,900	13,600	13,600	13,600	21,600	21,600	21,600	13,600	16,263
Rated HSPF	9.5	10.0	10.6	10.6	10.6	10.3	10.3	10.3	10.6	10.4
Adjusted HSPF**	8.55	9.00	9.54	9.54	9.54	9.27	9.27	9.27	9.54	9.4
Avg heating COP	2.51	2.64	2.8	2.8	2.8	2.72	2.72	2.72	2.8	2.8
Baseline electric resistance usage (kWh)	9,226	10,054	7,030	5,531	5,416	10,164	10,630	14,460	11,605	9,361
DHP energy usage (kWh)	3,682	3,812	2,514	1,978	1,937	3,741	3,913	5,323	4,151	3,421
DHP savings (kWh)	5,544	6,242	4,515	3,552	3,478	6,423	6,717	9,137	7,454	5,940
Savings @ $0.14/kWh	$776	$874	$632	$497	$487	$899	$940	$1,279	$1,044	$832
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating (kWh)										**4,502**
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating										**$630**

* Site #1 is not included in the average calculations as it is not a cold-climate model.

** The HSPF is adjusted by a factor of 0.9 to account for climate conditions for central New Hampshire.

Table 12-2. Monitored DHP Normalized Heating Season Savings Compared with Standard ASHP Baseline

Site #	1*	2	3	4	5	6	7	8	9	Average*
Baseline ASHP energy usage (kWh)	6,173	6,727	4,703	3,700	3,623	6,801	7,113	9,675	7,765	6,263
DHP energy usage (kWh)	3,682	3,812	2,514	1,978	1,937	3,741	3,913	5,323	4,151	3,421
DHP energy savings (kWh)	2,491	2,915	2,189	1,722	1,686	3,059	3,199	4,352	3,614	2,842
Savings @ $0.14/kWh	$349	$408	$306	$241	$236	$428	$448	$609	$506	$398
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating										**2,154**
Sites 2-9 weighted average savings per ton (12,000 Btu) of heating										**$302**

* Site #1 is not included in the average calculations as it is not a cold-climate model.

Table 12-3. Monitored DHP Normalized Heating Season Savings Compared with Fuel Oil Baseline

Site #	1*	2	3	4	5	6	7	8	9	Average*
Baseline #2 fuel oil displaced (gallons)	291	317	222	174	171	321	335	456	366	295
Baseline #2 fuel oil cost @ $3.70	$1,077	$1,173	$820	$645	$632	$1,186	$1,241	$1,687	$1,354	$1,092
DHP energy usage (kWh)	3,682	3,812	2,514	1,978	1,937	3,741	3,913	5,323	4,151	3,421
DHP energy usage cost @ $0.14/kWh	$516	$534	$352	$277	$271	$524	$548	$745	$581	$479
Net savings	$561	$640	$468	$368	$361	$662	$693	$942	$773	$613
Weighted average savings per ton (12,000 Btu) of heating										**$465**

* Site #1 is not included in the average calculations as it is not a cold-climate model.

12.1.2 The Systems Perform Well at Extremely Cold Temperatures

All of the systems monitored performed well at cold temperatures, with all but one of the systems producing effective heat well below 0°F. Eight of the nine systems continued producing heat down to their lowest minimum outdoor temperature limit of -18°F, although full output is not maintained when outdoor temperatures are below -5°F. The remaining system is a larger (30,000 Btu/h) DHP (site #1) that continued to deliver heat down to its minimum operational limit of 0°F.

12.1.3 Additional Conclusions

In addition to potential savings and cold-climate performance, we were able to formulate several other conclusions regarding the performance of DHPs, as follows:

- **Cold-weather performance is critical for heating in the study area climate zone**. Both customer satisfaction and monitored performance were substantially lower for the DHP installed at site #1, which was not among the cold-climate systems, but rather a system with a 17°F full output low temperature rating, and a 0°F operational limit. The participant recently added electric baseboard to the same area.
- **Published HSPF and COP ratings can be misleading.** Because of the modulating nature of DHPs and variations in climate conditions, the standard rating methods are not necessarily good predictors of field performance. However, applying adjustment factors to the ratings improves the ability to use the ratings for predicting savings. For this study, a factor of 0.9 (10% reduction in the rating) was applied to the published HSPF rating. Published HSPF ratings are typically based on the climate conditions in "AHRI Zone IV" which extends as far north as coastal southern New England. Details of heat pump rating systems and their adjustment factors are found in Section 4.
- **DHPs typically displaced conventional heating.** No heating systems were uninstalled due to the installation of the DHPs. However, all systems fully or partially displaced heating produced by non-heat pump sources. The fuels and systems displaced included central oil boilers, vented kerosene and propane space heaters, unvented propane space heaters, and biomass pellet stoves. Table 3-3 provides details of displaced heating.
- **DHPs installed for supplemental heating often become a primary heating system with owner experience over time.** The participants progressively tended to rely on the installed DHPs as the primary heating system, as they learned the benefits through experience. Most participants reported an initial intent to utilize the systems to supplement the heat from installed central systems, but with positive experiences began to rely on the systems as the primary heat source.
- **For cooling, DHPs both replaced and displaced less-efficient A/C systems.** Six of the participants replaced standard window-installed A/Cs. One of those six was also considering the repair of an unused central A/C system prior to installing the DHPs. In

addition, two participants had contacted an HVAC dealer requesting a quotation for installing central A/C. The same dealer proposed and installed the DHPs as a heating and cooling alternative to the A/C only central system. One participant installed the DHP for heating and cooling, where there were no existing or proposed cooling systems. Table 3-4 provides details of replaced and displaced cooling systems.

- **Cooling season load-building is not a significant factor.** In all but one case, the DHPs installed replaced window A/C units, or were purchased instead of installing central A/C. With the increased efficiency of DHPs it can be concluded that summer peak load building is not significant, and that even in the central NH climate, some cooling savings are achieved.
- **Average summer load shape is coincident with New England ISO targeted peak periods**. Although the cooling loads in central New Hampshire are relatively small, the peak demand and the peak savings associated with cooling are coincident with the 1 p.m. to 5 p.m. weekday time periods identified a peak demand periods by the New England ISO. Section 5.2 provides cooling load shape charts and details.
- **Purchase decisions varied, but were often associated with A/C.** The project participants decided to install DHPs for a variety of reasons, including: replacing fossil fuel space heaters, replacing window A/Cs, supplementing central heating systems, dehumidification, and even experimenting to assess the savings potential. Eight of the nine participants either replaced existing cooling systems or purchased the DHPs as an alternative to a standard cooling system.
- **Users prefer simple remote control operation**. All participants control their DHPs with handheld remote controls. None have installed the optional wall-mount thermostats. Most users select a heating or cooling setpoint depending on the season and select auto for the fan speed. No participants reported utilizing automatic set-back features, and if any set-back/set-forward settings are selected it is done manually for specific individual time periods. None of the participants utilize any of the special heating or cooling settings, such as "economy," available with the remote control. Some participants reported selecting the fan speed rather than utilizing the auto setting at certain times, due mostly to sound levels. Dehumidification modes are sometimes selected during cooling season.
- **Comfort levels are high**. With the exception of site #1, participants were universally enthusiastic about the comfort levels achieved. None reported experiencing any negative comfort effects from conditioned air being blown directly on them. This can be attributed at least partly to proper placement and installation of the fan-coil units.
- **Typical usage varies by climate zone and fuel availability**. The monitoring in NH and the New York City area demonstrates that DHPs may be used very differently depending on the climate zone, as well as other factors. The preliminary findings of the Con Edison impact evaluation include a conclusion that 80% of the savings in the territory are attributable to cooling. This is in direct contrast with the findings of this study, which determined that the great majority of the savings are associated with heating. In addition

to climatic differences, the Con Edison program places a promotional emphasis on cooling, and both natural gas and district steam heat are prevalent.

12.2 Recommendations

The following recommendations for evaluation methodologies, program implementation strategies, and further study to close knowledge gaps are in addition to the recommendations made in Phase 1 of this study.

- **EM&V methodologies** – Phase 1 of this project proposed algorithms and methodologies for calculating energy savings associated with DHPs. The report also cautioned about the difficulty of assigning simple deemed values for systems that have highly variable usage and performance patterns based on climate conditions and occupant intervention. The data collected during this phase further reinforces those concerns and informs the following recommendations:
  - **Utilize standard ratings, recognizing the limitations**. SEER and HSPF, the ratings utilized for cooling and heating performance, respectively, are based on strict operational parameters under several steady-state laboratory conditions. The AHRI methodology for calculating HSPF includes coefficients for six heating zones, which are differentiated from the climate zones utilized for energy codes. However, the HSPF ratings are typically published only for zone IV, which includes coastal southern New England, New Jersey, Virginia, Kentucky, Kansas, etc. (A map of the zones is presented in Section 4, Figure 4-1). The OAT covered by the rating is 17°F – 47°F, which is appropriate for that climate zone. When predicting performance for regions north and south of zone IV, heating performance will be inaccurate.
  - **An accurate savings tool for DHPs is needed.** HeatCalc, a DOE-supported downloadable spreadsheet tool, includes a calculator to adjust published HSPF ratings for the local climate.[15] However, the adjustment factors were formulated prior to the introduction of cold-weather performing heat pumps and assume that electric resistance coils contribute part or all of the heating at colder temperatures. Updating the tool to be consistent with the cold-weather performance of cold-climate DHPs would provide program administrators, as well as market actors, a relatively simple tool for estimating DHP savings.
  - **Utilize performance monitoring and billing analysis to assist in predicting savings.** Standards organizations, as well as the heat pump industry, recognize the limitations of COP, SEER, and HSPF for calculating the performance of continuously modulating DHPs. Field studies and impact evaluation efforts that include monitoring and/or billing analysis should be used to further inform and adjust savings calculations for DHPs. Impact evaluation sponsors should allocate enough elapsed time for DHPs to be evaluated over a minimum of three seasons. An added

[15] www.eia.gov/neic/experts/heatcalc.xls.

benefit of evaluating performance over three or more seasons, or for multiple years, is that it allows for the capture of changes in usage that typically take place as the users gain experience with the systems.

- **Program implementation strategies** – The following recommendations are related to promoting DHPs as components of efficiency program portfolios.
  - **Stay current with DHP advances.** Program administrators should work to stay current with this advancing technology. As this is being written, at least one manufacturer is in the process of introducing yet another increase in efficiency for DHPs, as well as larger and multi-head units that perform at the low temperatures currently reached only by single-head units.
  - **Promote DHPs appropriate for the climate zone**. For heating-dominant climates, program administrators should consider restricting program participation to the installation of systems that will operate at near full-load conditions at the design temperature for the region. Customer disappointment and savings snapback are likely if support is given to DHPs that perform marginally in the lower ranges of the regional OATs.
  - **Consider DHPs for fuel switching**. In jurisdictions where incentives are allowed, DHPs are excellent candidates for fuel switching from oil heat. Where incentives are not allowed, DHP performance on a direct fuel cost basis is attractive compared to oil heat. When climate change is considered, replacing fossil fuel systems with DHPs becomes more attractive. Most climate change studies suggest that replacing fossil fuel systems with efficient electric systems powered by clean generation and/or renewable energy sources is a necessary component of meeting long-term climate goals. A recent European study concluded that "achieving an 80% GHG reduction across the economy will likely require massive electrification of space heating, water heating, and personal transportation while simultaneously de-carbonizing the power sector" (i.e., 95%–100% reliance on renewable, nuclear, and/or fossil fuels with carbon capture and storage).[16] A study of GHG emission reduction options for the state of California reached similar conclusions.[17] However, even before the grid is decarbonized, an efficient DHP would result in lower carbon emissions than an efficient gas furnace or boiler under many scenarios. For example, a DHP with a seasonal average COP of 2.7 that receives its power from a 45%-efficient natural gas power plant and a grid with marginal line losses of 10%. The delivered efficiency of the heat—from power plant to home heat—is 110% (0.45 × 0.9 × 2.7). In contrast, even a very efficient gas home heating system (condensing furnace or boiler, coupled with

[16] European Climate Foundation, *Roadmap 2050: Practical Guide to a Prosperous, Low-Carbon Europe*, Volume 1, April 2010, p. 6. See www.roadmap2050.eu .

[17] Price, Snuller, Energy and Environmental Economics, "Meeting California's Long-Term Greenhouse Gas Reduction Goals," prepared for Hydrogen Energy International, November 2009.

an efficient distribution system) will typically not be more than about 90% efficient. Since both are ultimately using gas, the DHP will produce approximately 20% lower carbon emissions.

- **Closing knowledge gaps** – It is hard to close all the knowledge gaps on rapidly advancing technologies. But if efficiency programs are to meet goals, knowledge of products, applications, and usage patterns is critical. Key points include:
  - **Information sharing** – A lot of work is being performed right now on DHP performance. This study focused attention on user operational experiences and equipment performance, but only for one part of the country. The planned EM&V Forum meta-study of DHP research, combined with other efforts, is intended to further increase access to more comprehensive assessment of DHP under various baseline scenarios.
  - **Control options** – Control options for DHPs include both programmable wall-mounted thermostats and hand-held remote controls. As noted, all of the DHPs monitored for this study were controlled by hand-held remote controls. The hand-held remote controls utilize a thermistor (an electrical resistor that varies with temperature) inside the DHP return air stream to monitor room temperature while the remote wall-mounted unit bypasses the built-in thermistor, sensing temperature at the thermostat location. Although it was not analyzed as part of this study, the fact that these control options operate differently poses the question of how performance might be impacted. Comparative studies of DHP performance with the two control types would inform decision-making for program administrators, market actors, and homeowners.
  - **Controls integration** – This study revealed that from nine participants came nine different methodologies for controlling their DHPs in relation to other heating systems. The methods can best be described as "work-arounds." Especially with the advancement of "smart" controls, and Web-accessible thermostats, program implementers and evaluators should work with the industry to identify advantageous methodologies for controlling multiple systems, in order to encourage optimized control of the systems.
  - **Commercial markets** – To date, most of the focus on promoting DHPs, especially recently introduced high efficiency/low temperature models, has been on the residential market. The time is right to increase the penetration of high efficiency DHP in the small/medium commercial market. In addition to the ability to perform at high efficiency levels, DHPs are able to solve difficult heat/cooling zone issues; avoid simultaneous heating and cooling; isolate ventilation and conditioning systems; condition specialty areas such as server rooms; and provide variable control for areas of variable occupancy such as conference rooms, etc.
  - **Performance ratings** – Replacements for SEER and HSPF may not be available soon. A reasonable goal would be for manufacturers to supply the SEER and HSPF ratings

appropriate for the efficiency program territories. Although it is understandable that the industry desires to publish one set of numbers, efficiency programs need performance metrics for the local climate in order to accurately predict savings. Although the DOE-supported tool HeatCalc includes a calculator to adjust published HSPF and SEER ratings for the local climate, it too is inaccurate for predicting the heating performance of cold-climate DHP.

13 SUMMARY

Just as solid-state LED technology is changing the lighting market faster than predicted, DHPs are advancing at a rate faster than was thought possible a short time ago. Especially in the area of cold-weather performance, the advancement has been remarkable and well received by markets and efficiency program administrators. Less than 5 years ago, some manufacturers of cold-climate heat pumps were struggling due to performance failures and system maintenance costs. Now, with performance and reliability issues resolved, manufacturers are focusing on the next level of performance and systems that reach broader markets. Evaluators and implementers should work together to assure that customer satisfaction levels are met and that DHPs continue to advance the market.

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Appendices:

Appendix A – DHP Specification Sheets

Appendix B – Participant Surveys

Appendix C – Single-Site Cooling Load Shapes

Appendix D – DHP Performance Test Report

Appendix E – AHRI Testing procedure

Appendix A

DHP Specification Sheets

Submittal Data: System 30RLX
ASU30RLX & AOU30RLX

FUJITSU

Job Name:		Location:	
Purchaser:			
Engineer:			
Submitted To:		For:	❑ Reference ❑ Approval ❑ Construction
Submitted By:			
Unit Designation:	Schedule No.	Model No.:	

Capacities:

Cooling	30,600 BTU/h
Outdoor Design Temp. *F° DB/WB*	95/75
Heating	32,000 BTU/h
Outdoor Design Temperature *F° DB/WB*	47/43
HSPF	9.5
SEER	17.5
EER *Cooling/Heating*	10.0/10.7
Voltage/Frequency/Phase	208-230/60/1

Indoor Unit:

Noise Level Cooling *db (A) - H/ M/ L/ Q*	49/ 42/ 37/ 33
Noise Level Heating *db (A) - H/ M/ L/ Q*	49/ 42/ 37/ 33
Weight	31 lbs.

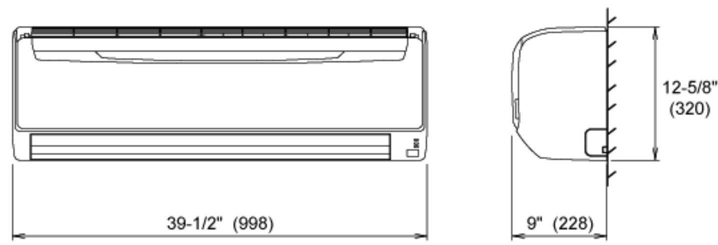
Outdoor Unit:

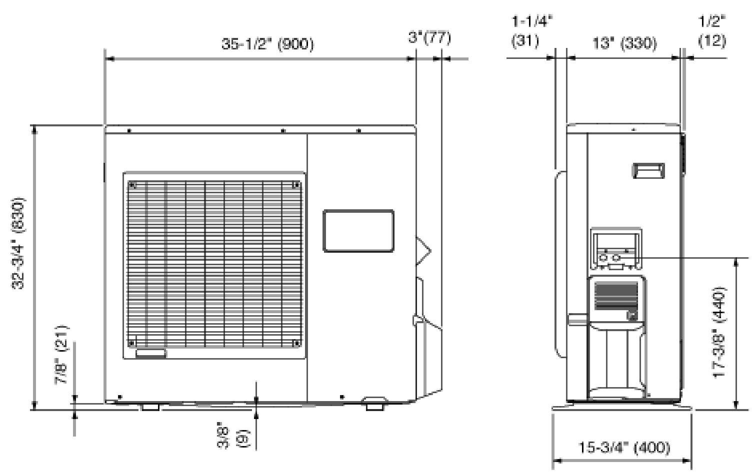
Noise Level *Cooling/Heating*	54/55
Recommended Fuse Size	30A
Min. Ampacity	13.1A
Running Current *Cooling*	13.2A
Running Current *Heating*	10.5A
Weight	137 lbs.

Refrigerant Piping:

Max Ht. Difference	98 ft.
Max Total or Combined Length	164 ft.
Discharge Vapor Line (O.D.)	3/8 in.
Suction (O.D.)	5/8 in.

OUTLINE AND DIMENSIONS





- Six year compressor warranty
- Two year parts warranty
- Digital wireless remote control
- 4-way automatic louvers
- Min Heat (50F heating set point)
- Built in Low Ambient
- Auto Restart/ Reset
- 24 hour timer
- Optional Wired Remote Control
- Dry mode
- Refrigerant R410A
- Quiet Mode

Notes:

MITSUBISHI ELECTRIC

COOLING & HEATING

M-SERIES

SUBMITTAL DATA: MSZ-FE18NA & MUZ-FE18NA

18,000 BTU/H WALL-MOUNTED HEAT PUMP SYSTEM

Job Name:	Location:	Date:
Purchaser:	Engineer:	
Submitted to:	For ☐Reference ☐Approval ☐Construction	
System Designation:	Schedule No.:	

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GENERAL FEATURES

- Highly energy-efficient system with quiet operation
- Updated sleek, compact indoor unit design
- Includes Nano Platinum and Anti-allergy Enzyme filters
- "Powerful Mode" function permits system to temporarily run at a lower/higher temperature with an increased fan speed, which quickly brings the room to the optimum comfort level
- Hand-held Wireless Remote Controller
- Limited warranty: five years parts and seven years compressors

ACCESSORIES

Outdoor Unit

☐ Base Heater (MAC-642BH-U)
☐ Three-pole Disconnect Switch (TAZ-MS303)
☐ Air Outlet Guide (MAC-856SG)
☐ Mounting Base (DSD-400N)
☐ Mounting Pad (ULTRILITE1)
☐ Drain Socket Assembly (MAC-860DS)
☐

Indoor Unit

☐ Condensate Pump (SI3100-230; 230V)
☐ Replacement Anti-allergy Enzyme Filters (MAC-2310FT-E; 2/set)

Controller Options

☐ Wireless Wall-mounted Remote Controller Kit (MHK1)*
☐ Portable Central Controller (MCCH1)*
☐ Outdoor Air Sensor (MOS1)*
☐ Wired Wall-mounted Controller (PAR-31MAA requires MAC-333IF)*
☐ Simple MA Remote Controller (PAC-YT53CRAU requires MAC-333IF)*
*See Submittal for information on each option.
☐ System Control Interface (MAC-333IF)
☐ Remote Temperature Sensor (M21-JKO-307)
☐ Lockdown Bracket for Hand-held Controller (RCMKP1CB)

R410A INVERTER

Specifications are subject to change without notice.

SPECIFICATIONS : MSZ-FE18NA & MUZ-FE18NA

Cooling*
Rated Capacity . . . . . 18,000 Btu/h
Minimum to Maximum Capacity Range . . . . . 8,200 - 25,200 Btu/h
SEER . . . . . 20.2 Btu/h/W
EER . . . . . 14.2 Btu/h/W
Total Rated Input . . . . . 1,270 W

Heating at 47° F*
Rated Capacity . . . . . 21,600 Btu/h
Minimum to Maximum Capacity Range . . . . . 7,500 - 29,700 Btu/h
HSPF . . . . . 10.3 Btu/h/W
COP 4.11
Total Rated Input . . . . . 1,540 W

Heating at 17° F*
Rated Capacity . . . . . 11,700 Btu/h
Rated Total Input . . . . . 1,240 W
COP 2.77
Maximum Capacity** . . . . . 21,600 Btu/h
Maximum Total Input . . . . . 2,620 W

Heating at 5° F*
Maximum Capacity** . . . . . 21,600 Btu/h

*** Rating Conditions per AHRI Standard**
Cooling | Indoor: 80º F (27º C) DB / 67º F (19º C) WB
Cooling | Outdoor: 95º F (35º C) DB / 75º F (24º C) WB
Heating at 47ºF | Indoor: 70º F (21º C) DB / 60º F (16º C) WB
Heating at 47ºF | Outdoor: 47º F (8º C) DB / 43º F (6º C) WB
Heating at 17º F | Indoor: 70º F (21º C) DB / 60º F (16º C) WB
Heating at 17º F | Outdoor: 17º F (-8º C) DB / 15º F (-9º C) WB
Heating at 5º F | Indoor: 70º F (21º C) DB / 60º F (16º C) WB
Heating at 5º F | Outdoor: 5º F (-15º C) DB / 5º F (-15º C) WB

**Maximum Capacity is at full speed and performance for INVERTER-driven System.

Electrical Requirements
Power Supply . . . . . 208 / 230V, 1-Phase, 60 Hz
Breaker Size . . . . . 20 A

Voltage
Indoor - Outdoor S1-S2 . . . . . AC 208 / 230V
Indoor - Outdoor S2-S3 . . . . . DC +/- 24V
Indoor - Remote Controller . . . . . MKH1 DC 3V
PAR-31MAA DC 12V
PAC-YT53CRAU DC 12V

OPERATING CONDITIONS

		Indoor Intake Air Temp.	Outdoor Intake Air Temp.
Cooling	Maximum	90º F (32º C) DB 73º F (23º C) WB	115º F (46º C) DB
	Minimum	67º F (19º C) DB 57º F (14º C) WB	14º F (-10º C) DB
Heating	Maximum	80º F (27º C) DB 67º F (19º C) WB	75º F (24º C) DB 65º F (18º C) WB
	Minimum	70º F (21º C) DB 60º F (16º C) WB	-13º F (-25º C) DB -15º F (-26º C) WB**

** System cuts out at -18º F (-28º C) to avoid thermistor error, but recovers from cutout operation and automatically restarts at -13º F (-25ºC).

Indoor Unit
MCA . . . . . 1 A
Blower Motor (ECM) . . . . . 0.76 F.L.A.
Airflow
Cooling (Lo - Med - Hi - Powerful) . . . . 388 - 469 - 628 - 738 Dry CFM
347 - 420 - 562 - 661 Wet CFM
Heating (Lo - Med - Hi - Powerful) . . . . 388 - 469 - 628 - 738 Dry CFM
Sound Pressure Level
Cooling (Lo - Med - Hi - Powerful) . . . . . 34 - 41 - 49 - 53 dB(A)
Heating (Lo - Med - Hi - Powerful) . . . . . 32 - 41 - 49 - 52 dB(A)

DIMENSIONS	UNIT INCHES / MM
W	43-5/16 / 1,116
D	9-3/8 / 238
H	12-13/16 / 325

Weight . . . . . 37 lbs. / 17 kg
Moisture Removal . . . . . 2.7 pt./h
External Finish . . . . . Munsell No. 1.0Y 9.2 / 0.2
Field Drainpipe Size O.D. . . . . . 5/8" / 15.88 mm

Outdoor Unit
Compressor . . . . . DC Inverter-driven Twin Rotary
MCA . . . . . 17.1 A
Fan Motor (ECM) . . . . . 0.93 F.L.A.
Sound Pressure Level
Cooling . . . . . 55 dB(A)
Heating . . . . . 55 dB(A)

DIMENSIONS	INCHES / MM
W	33-1/16 + 3-3/16 / 840 + 81
D	13 / 330
H	34-5/8 / 880

Weight . . . . . 119 lbs. / 54 kg
External Finish . . . . . Munsell No. 3.0Y 7.8 / 1.1
Refrigerant Type R410A
Refrigerant Pipe Size O.D.
Gas Side . . . . . 5/8" / 15.88 mm
Liquid Side . . . . . 3/8" / 6.35 mm
Max. Refrigerant Pipe Length . . . . . 100' / 30 m
Max. Refrigerant Pipe Height Difference . . . . . 50' / 15 m
Connection Method . . . . . Flared

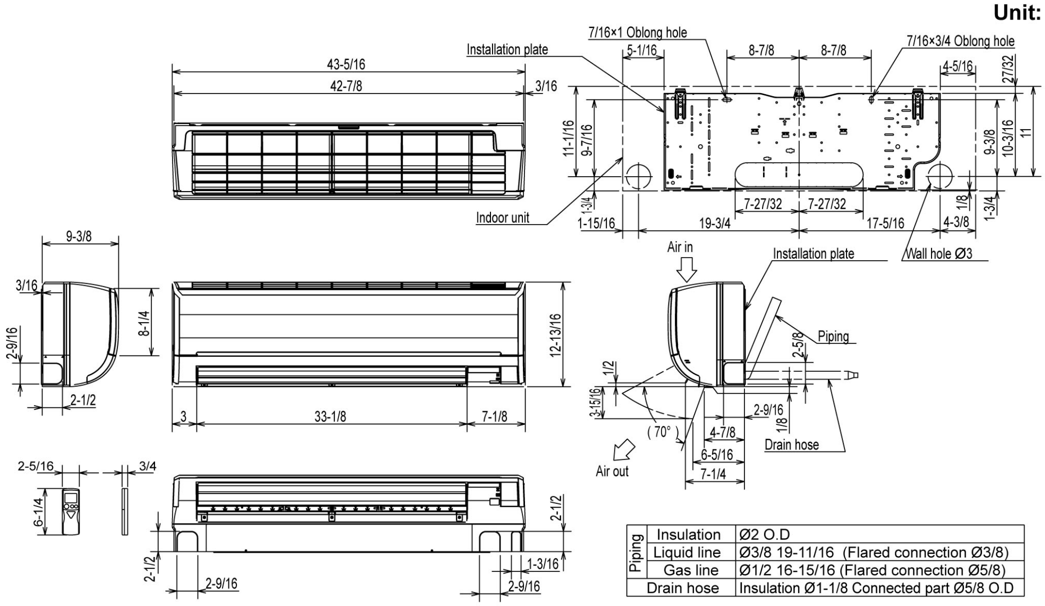
Notes:

Specifications are subject to change without notice.

DIMENSIONS: MSZ-FE18NA & MUZ-FE18NA

MSZ-FE18NA

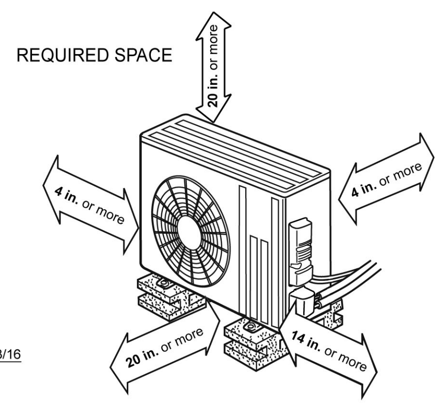
Unit: inch

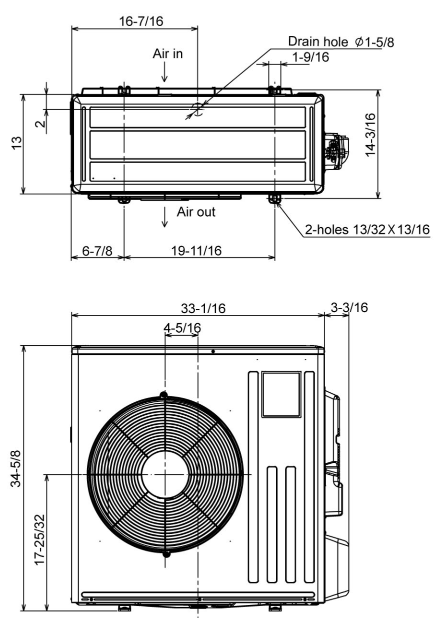


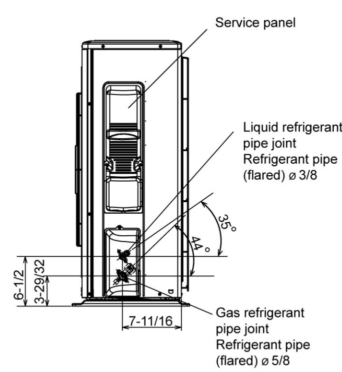
Piping	Insulation	Ø2 O.D
	Liquid line	Ø3/8 19-11/16 (Flared connection Ø3/8)
	Gas line	Ø1/2 16-15/16 (Flared connection Ø5/8)
Drain hose		Insulation Ø1-1/8 Connected part Ø5/8 O.D

MUZ-FE18NA

Unit: inch







Unitary Small HP
AHRI Standard 210/240
Certification applies only when the complete system is listed with AHRI.

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FORM# MSZ-FE18NA / MUZ-FE18NA - 201311

Specifications are subject to change without notice.

MITSUBISHI ELECTRIC

COOLING & HEATING

Mr.SLIM

Split-ductless A/C and Heat Pumps

SUBMITTAL DATA: MSZ-FE12NA-8 & MUZ-FE12NA1

12,000 BTU/H WALL-MOUNTED HEAT PUMP SYSTEM

Job Name:	Location:	Date:
Purchaser:	Engineer:	
Submitted to:	For ☐Reference ☐Approval ☐Construction	
System Designation:	Schedule No.:	

GENERAL FEATURES

- Highly energy-efficient system that features 92% heating capacity at 5°F, 75% at -4°F, and 58% at -13°F
- Updated sleek, compact indoor unit design
- Includes Standard, Platinum Deodorizing, and Anti-allergy Enzyme filters for a complete air purifying system
- "Powerful Mode" function permits system to temporarily run at a lower/higher temperature with an increased fan speed, which quickly brings the room to the optimum comfort level
- Integrated i-see Sensor automatically adjusts the unit's operation according to temperature differences detected between the floor and the intake air, ensuring optimum comfort and energy usage
- Hand-held Wireless Remote Controller
- Base heater is available as an option
- Limited warranty: five years on parts and defects and seven years on compressors

OPTIONAL ACCESSORIES

Outdoor Unit

☐ Base Heater (MAC-640BH-U)
☐ Drain Socket (MAC-851DS)

Indoor Unit

☐ Condensate Pump (SI3100-230; 230V)
☐ Replacement Platinum Deodorizing Filter (MAC-308FT)
☐ Replacement Anti-allergy Enzyme Filter (MAC-418FT; MERV 8)

Controller Options

☐ Wireless Remote Controller Kit (MHK1) with Remote Controller (MRCH1), Wireless Receiver (MIFH1), and cable (MRC1)*
☐ Setback down to 50°F when used with MRCH1 Remote Controller
☐ Portable Central Controller (MCCH1; for use with Wireless Remote Controller Kit MHK1)*
☐ Outdoor Air Sensor (MOS1; for use with Remote Controller (MRCH1), Wireless Remote Controller Kit (MHK1) and Portable Central Controller (MCCH1)*

*See Submittal for information on each option.

☐ Wall-mounted Wired Remote Controller (PAR-21MAA; req. MAC-3971F)
☐ MA Contact Terminal Interface (MAC-397IF)
☐ M-NET Control Adapter (MAC-399IF)
☐ Remote Temperature Sensor (M21-JKO-307)
☐ Lockdown Bracket for Hand-held Controller (RCMKP1CB)

Cooling*

Rated Capacity . . . . 12,000 Btu/h
Minimum Capacity . . . . 2,800 Btu/h
SEER . . . . 23.0 Btu/h/W
Total Input . . . . 930 W

Heating at 47° F*

Rated Capacity . . . . 13,600 Btu/h
Minimum Capacity . . . . 3,000 Btu/h
HSPF . . . . 10.6 Btu/h/W
Total Input . . . . 950 W

Heating at 17° F*

Rated Capacity . . . . 7,900 Btu/h
Rated Total Input . . . . 750 W
Maximum Capacity . . . . 13,600 Btu/h
Maximum Total Input . . . . 1,780 W

* Rating Conditions (Cooling) - Indoor: 80°F (27°C) DB, 67°F (19°C) WB; Outdoor: 95°F (35°C) DB, 75°F (24°C) WB.
(Heating at 47°F) - Indoor: 70°F (21°C) DB, 60°F (16°C) WB; Outdoor: 47°F (8°C) DB, 43°F (6°C) WB.
(Heating at 17°F) - Indoor: 70°F (21°C) DB, 60°F (16°C) WB; Outdoor: 17°F (-8°C) DB, 15°F (-9°C) WB.

Heating at 5° F

Maximum Capacity . . . . 12,500 Btu/h

Electrical Requirements

Power Supply . . . . 208 / 230V, 1-Phase, 60 Hz
Breaker Size . . . . 15 A

Voltage

Indoor - Outdoor S1-S2 . . . . AC 208 / 230V
Indoor - Outdoor S2-S3 . . . . DC 12-24V
Wireless Remote Controller . . . . DC

Indoor Unit: MSZ-FE12NA-8

ENERGY STAR

Wireless Remote Controller

Outdoor Unit: MUZ-FE12NA1

OPERATING RANGE

		Indoor Intake Air Temp.	Outdoor Intake Air Temp.
Cooling	Maximum	90°F (32°C) DB, 73°F (23°C) WB	115°F (46°C) DB
	Minimum	67°F (19°C) DB, 57°F (14°C) WB	14°F (-10°C) DB
Heating	Maximum	80°F (27°C) DB, 67°F (19°C) WB	75°F (24°C) DB, 65°F (18°C) WB
	Minimum	70°F (21°C) DB, 60°F (16°C) WB	-13°F (-25°C) DB, -15°F (-26°C) WB**

** System cuts out at -18°F (-28°C) to avoid thermistor error, but recovers from cutout operation and automatically restarts at -13°F (-25°C).

Indoor Unit

MCA . . . . 1 A
Fan Motor . . . . 0.76 F.L.A.
Airflow
Cooling (Lo - Med - Hi - Powerful) . . . . 162 - 226 - 381 - 410 Dry CFM
144 - 202 - 350 - 367 Wet CFM
Heating (Lo - Med - Hi - Powerful) . . . . 166 - 240 - 399 - 420 Dry CFM
Sound Pressure Level
Cooling (Lo - Med - Hi - Powerful) . . . . 22 - 33 - 43 - 45 dB(A)
Heating (Lo - Med - Hi - Powerful) . . . . 22 - 33 - 43 - 44 dB(A)

DIMENSIONS	UNIT INCHES / MM
W	31-7/16 / 799
D	10-1/8 / 257
H	11-5/8 / 295

Weight . . . . 27 lbs. / 12 kg
External Finish . . . . Munsell No. 1.0Y 9.2 / 0.2
Field Drainpipe Size O.D. . . . . 5/8" / 15.88 mm
Remote Controller . . . . Wireless

Outdoor Unit

Compressor . . . . DC Inverter-driven Twin Rotary
MCA . . . . 12 A
Fan Motor . . . . 0.56 F.L.A.
Sound Pressure Level
Cooling . . . . 48 dB(A)
Heating . . . . 49 dB(A)

DIMENSIONS	INCHES / MM
W	31-1/2 / 800
D	11-1/4 / 286
H	21-5/8 / 549

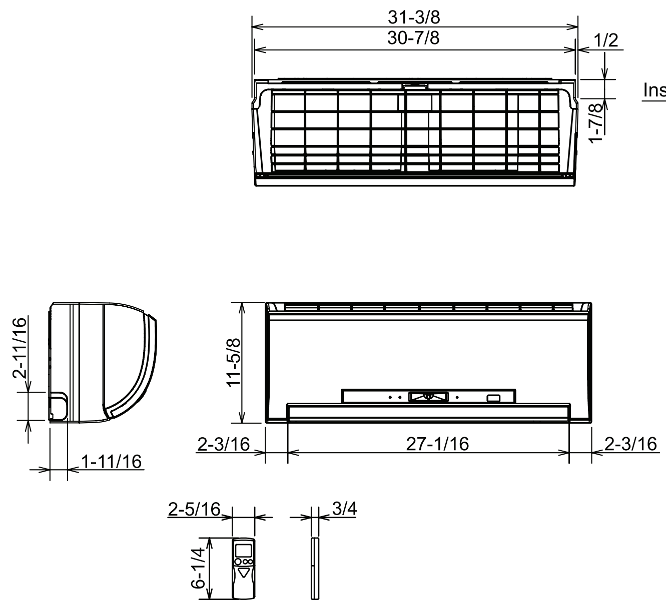
Weight . . . . 80 lbs. / 36 kg
External Finish . . . . Munsell No. 3.0Y 7.8 / 1.1
Refrigerant Type R410A
Refrigerant Pipe Size O.D.
Gas Side . . . . 3/8" / 9.52 mm
Liquid Side . . . . 1/4" / 6.35 mm
Max. Refrigerant Pipe Length . . . . 65' / 20 m
Max. Refrigerant Pipe Height Difference . . . . 40' / 12 m
Connection Method . . . . Flared

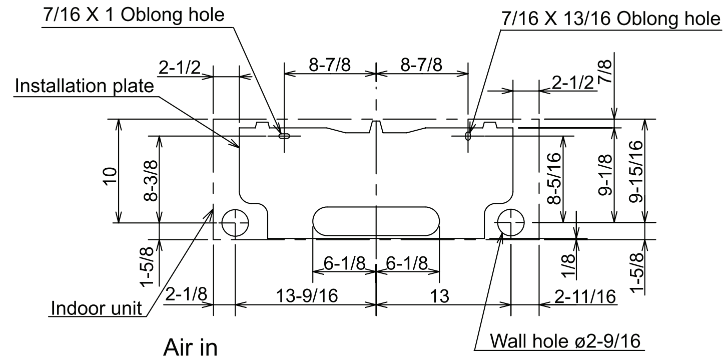
R410A INVERTER

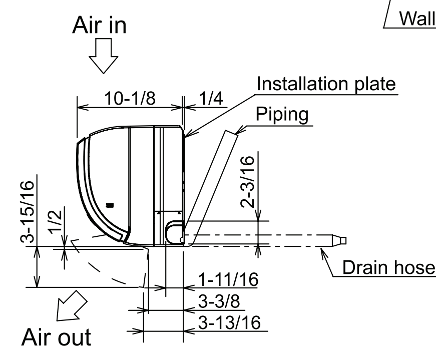
DIMENSIONS: MSZ-FE12NA-8 & MUZ-FE12NA1

MSZ-FE12NA-8

Unit: inch



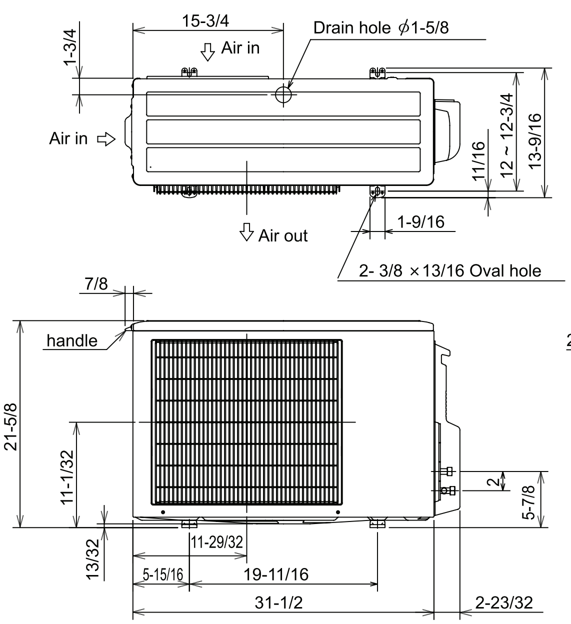


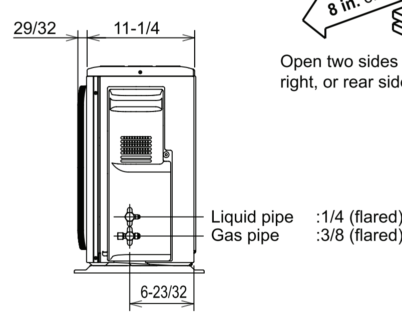


Piping	Insulation	ø1-3/8 O.D
	Liquid line	ø1/4 19-11/16 (Flared connection ø1/4)
	Gas line	ø3/8 16-15/16 (Flared connection ø3/8)
Drain hose		Insulation ø1-1/8 O.D Connected part ø5/8 O.D

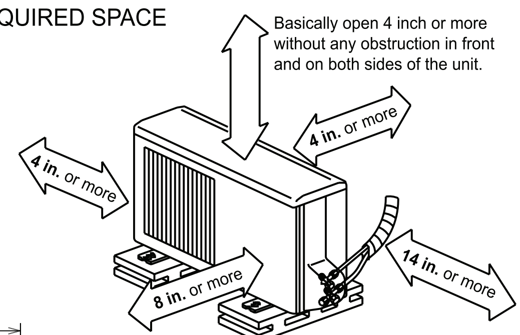
MUZ-FE12NA1

Unit: inch





REQUIRED SPACE



Open two sides of left, right, or rear side.

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AHRI Standard 210/240
Certification applies only when the complete system is listed with AHRI.

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FORM# MSZ-FE12NA-8 ~ MUZ-FE12NA1 - 201108

MITSUBISHI ELECTRIC

Mr.SLIM® Split-ductless A/C and Heat Pumps

SUBMITTAL DATA: MSZ-FE09NA & MUZ-FE09NA

9,000 BTU/H WALL-MOUNTED HEAT PUMP SYSTEM

Job Name: Location: Date:

Purchaser: Engineer:

Submitted to: For ☐ Reference ☐ Approval ☐ Construction

Unit Designation: Schedule No.:

GENERAL FEATURES

- Highly energy-efficient system that features 100% heating capacity at 5°F, 82% at -4°F, and 62% at -13°F
- Updated sleek, compact indoor unit design
- Includes Standard, Platinum Deodorizing, and Anti-allergy Enzyme filters for a complete air purifying system
- "Powerful Mode" function permits system to temporarily run at a lower/higher temperature with an increased fan speed, which quickly brings the room to the optimum comfort level
- Integrated i-see Sensor automatically adjusts the unit's operation according to temperature differences detected between the floor and the intake air, ensuring optimum comfort and energy usage
- Wireless remote controller
- Base heater is available as an option
- Limited warranty: five years on parts and defects and seven years on compressors

OPTIONAL ACCESSORIES

Outdoor Unit

- Base Heater (MAC-640BH-U)
- Drain Socket (MAC-851DS)

Indoor Unit

- M-NET Control Adapter (MAC-399IF)
- MA Contact Terminal Interface (MAC-397IF)
- Wired Remote Controller (PAR-21MAA; Requires MAC-3971F)
- Condensate Pump (SI3100-230; 230V)
- Replacement Platinum Deodorizing Filter (MAC-308FT)
- Replacement Anti-allergy Enzyme Filter (MAC-418FT; MERV 8)

Cooling*

Rated Capacity . . . . 9,000 Btu/h
Minimum Capacity . . . . 2,800 Btu/h
SEER . . . . 26.0 Btu/h/W
Total Input . . . . 580 W

Heating at 47° F*

Rated Capacity . . . . 10,900 Btu/h
Minimum Capacity . . . . 3,000 Btu/h
HSPF . . . . 10.0 Btu/h/W
Total Input . . . . 710 W

Heating at 17° F*

Rated Capacity . . . . 6,700 Btu/h
Rated Total Input . . . . 650 W
Maximum Capacity . . . . 12,500 Btu/h
Maximum Total Input . . . . 1,730 W

* Rating Conditions (Cooling) - Indoor: 80°F (27°C) DB, 67°F (19°C) WB; Outdoor: 95°F (35°C) DB, 75°F (24°C) WB.
(Heating at 47°F) - Indoor: 70°F (21°C) DB, 60°F (16°C) WB; Outdoor: 47°F (8°C) DB, 43°F (6°C) WB.
(Heating at 17°F) - Indoor: 70°F (21°C) DB, 60°F (16°C) WB; Outdoor: 17°F (-8°C) DB, 15°F (-9°C) WB.

Heating at 5° F

Maximum Capacity . . . . 10,900 Btu/h

Electrical Requirements

Power Supply . . . . 208 / 230V, 1-Phase, 60 Hz
Breaker Size . . . . 15 A

Voltage

Indoor - Outdoor S1-S2 . . . . AC 208 / 230V
Indoor - Outdoor S2-S3 . . . . DC 12-24V
Wireless Remote Controller . . . . DC

OPERATING RANGE

		Indoor Intake Air Temp.	Outdoor Intake Air Temp.
Cooling	Maximum	90°F (32°C) DB, 73°F (23°C) WB	115°F (46°C) DB
	Minimum	67°F (19°C) DB, 57°F (14°C) WB	14°F (-10°C) DB
Heating	Maximum	80°F (27°C) DB, 67°F (19°C) WB	75°F (24°C) DB, 65°F (18°C) WB
	Minimum	70°F (21°C) DB, 60°F (16°C) WB	-13°F (-25°C) DB, -15°F (-26°C) WB**

** System cuts out at -18°F (-28°C) to avoid thermistor error, but recovers from cutout operation and automatically restarts at -13°F (-25°C).

Indoor Unit: MSZ-FE09NA

Wireless Remote Controller

ENERGY STAR

Outdoor Unit: MUZ-FE09NA

Indoor Unit

MCA . . . . 1 A
Fan Motor . . . . 0.76 F.L.A.
Airflow
Cooling (Lo - Med - Hi - Powerful) . . . . 162 - 226 - 339 - 381 Dry CFM
144 - 202 - 307 - 343 Wet CFM
Heating (Lo - Med - Hi - Powerful) . . . . 166 - 240 - 367 - 381 Dry CFM
Sound Pressure Level
Cooling (Lo - Med - Hi - Powerful) . . . . 22 - 31 - 39 - 42 dB(A)
Heating (Lo - Med - Hi - Powerful) . . . . 22 - 31 - 40 - 42 dB(A)

DIMENSIONS	UNIT INCHES / MM
W	31-7/16 / 799
D	10-1/8 / 257
H	11-5/8 / 295

Weight . . . . 27 lbs. / 12 kg
External Finish . . . . Munsell No. 1.0Y 9.2 / 0.2
Field Drainpipe Size O.D. . . . . 5/8" / 15.88 mm
Remote Controller . . . . Wireless

Outdoor Unit

Compressor . . . . DC Inverter-driven Twin Rotary
MCA . . . . 12 A
Fan Motor . . . . 0.56 F.L.A.
Sound Pressure Level
Cooling . . . . 48 dB(A)
Heating . . . . 49 dB(A)

DIMENSIONS	INCHES / MM
W	31-1/2 / 800
D	11-1/4 / 286
H	21-5/8 / 549

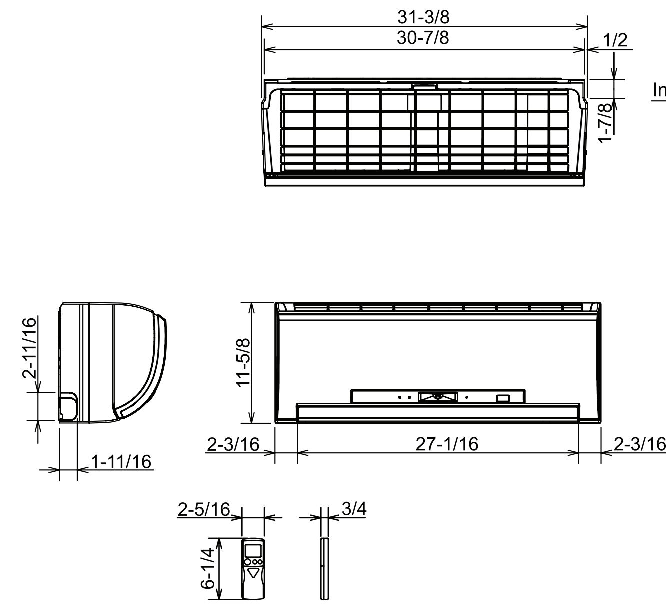
Weight . . . . 80 lbs. / 36 kg
External Finish . . . . Munsell No. 3.0Y 7.8 / 1.1
Refrigerant Type R410A
Refrigerant Pipe Size O.D.
Gas Side . . . . 3/8" / 9.52 mm
Liquid Side . . . . 1/4" / 6.35 mm
Max. Refrigerant Pipe Length . . . . 65' / 20 m
Max. Refrigerant Pipe Height Difference . . . . 40' / 12 m
Connection Method . . . . Flared

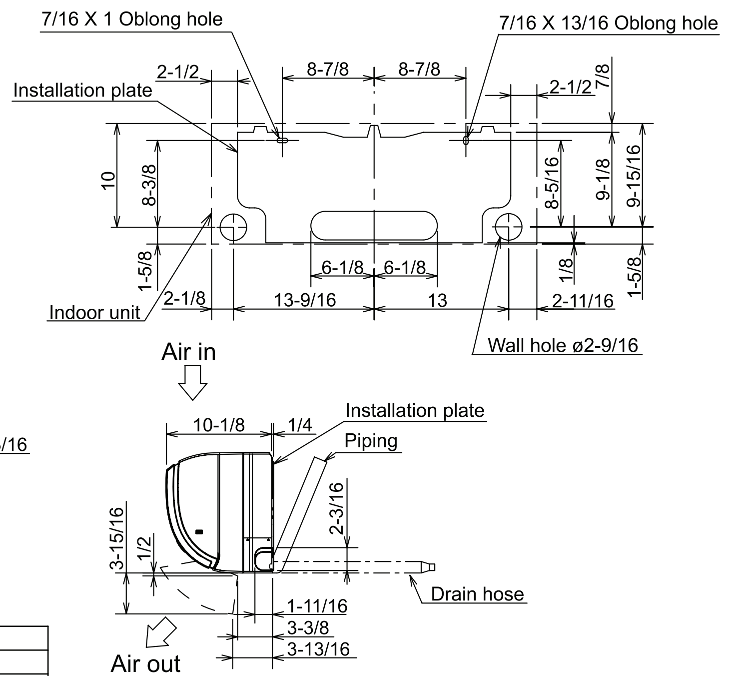
R410A INVERTER

DIMENSIONS: MSZ-FE09NA & MUZ-FE09NA

MSZ-FE09NA

Unit: inch

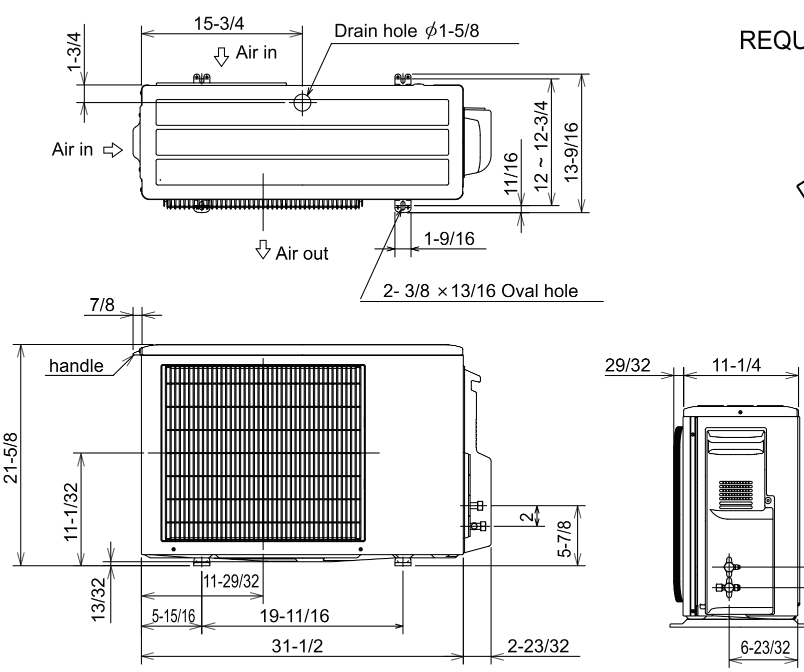


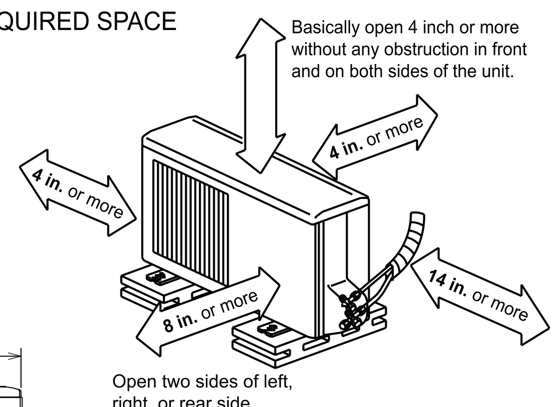


Piping	Insulation	ø1-3/8 O.D
	Liquid line	ø1/4 19-11/16 (Flared connection ø1/4)
	Gas line	ø3/8 16-15/16 (Flared connection ø3/8)
Drain hose		Insulation ø1-1/8 O.D Connected part ø5/8 O.D

MUZ-FE09NA

Unit: inch





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Suwanee, GA 30024
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www.mehvac.com

Specifications are subject to change without notice.

Appendix B

Participant Surveys

Appendix B – Participant Surveys – Site Details

Site Details

The following sections provide more detail regarding the sites visited, the DHPs installed, the conventional systems replaced/displaced, and the reported participant experiences operating the systems.

Site #1: Plymouth, NH

Site 1 is a single family, year-round residence, constructed in 1950. The house is a single story ranch style home with a living area of approximately 1,500 ft^2. Within the last ten years the home has been insulated with blown-in cellulose, caulked and weather-stripped, and the windows were replaced.

Site Specifics:

DHP monitored - Fujitsu AOU30RLX. Rated for 30,000 Btu/h cooling and 32,000 Btu/h heating.

Location – Living Room; 14'x20' – 280 ft^2

Usage – Heating and cooling

Systems Replaced:

5,000 BTU window A/C unit (remains installed, but is not used)

Heating fuels displaced:

Primary - Toyo kerosene vented unit heater; 22,000 Btu/h capacity; AFUE 87% installed in the living room

2-unvented "Mr. Buddy" propane space heaters using 5-20lbs "grill" tanks per year total

Wood stove using approximately 2 cord of hardwood per year

Home has no central heating system

DHP controls – remote control; no wall thermostat

Other thermostats - none

System Operation:

The DHP was purchased primarily for heating, with the expectation that the kerosene heater would be completely displaced, and that the entire home would be heated by the DHP. The DHP is currently used as follows:

Heating:

- The DHP heats the living room and contributes to other spaces during the day.

- During the night, the homeowner typically turns off the DHP and allows the area to cool, providing some heat from the kerosene heater.
- During the day, the remote control is typically set at 70°F with the fan speed set to medium.
- No setback settings are used during the night or unoccupied periods, as the system is turned off.

Cooling:

The DHP cools the living room and surrounding areas, replacing a 5,000 Btu window A/C unit.

The DHP is turned on for cooling when the outside temperature is above approximately 80°F

For cooling the remote control is typically set at 70°F with the fan speed set to low.

No set-forward setting is used.

Homeowner Comments:

The user of this system is the only homeowner surveyed who expressed disappointment in the performance of the system. The following was reported both during the logger installation and during the post-monitoring interview:

The homeowner expected the DHP to heat the entire house except during extremely cold conditions. For this reason a larger unit was selected.

The DHP does not maintain the desired temperature throughout the house and the propane and kerosene heaters are used more than desired.

The installer has returned to check system operation and reported no system faults.

In December 2013, 19′ of electric baseboard was added to the home to supplement the DHP heating.

Cooling performance is good and the system cools much more of the total house than the window A/C unit was able to serve.

Additional Observations:

It is noteworthy that this is the one participant site where the homeowner reported that the DHP was not providing the level of heating anticipated, and additional electric heating is being installed. We observe several factors that likely contribute to this situation:

The DHP is oversized for the space in which it is installed, although the homeowner was anticipating that the heat would migrate effectively throughout the house.

The unit is rated to operate at a minimum temperature of 0°F, and provides full output at 17°F and above. Temperatures drop well below these values in the region. At the time of install, DHPs rated over 20,000 Btu/h were not available in models that effectively provided heat below 0°F.

The residence was built in the 1950s, and although some weatherization has been performed, building envelope performance may be a significant factor.

The monitoring did reveal some unusual load patterns during the heating season. This could be attributable to the participant use of the DHP controls and the use of a variety of space heaters, but it is possible that the unit is not operating to specification.

The monitoring also showed that although the system was reported to not be able to meet the heating load, it never operated at its full rated power capacity of 4.1kW. The highest recorded load was 3.7kW, 10% below rated capacity. Recorded amperage revealed the same 10% differential. This is illustrated in Table 7.1.

Site #2: Gilmanton, NH

Site 2 is a single family year-round residence constructed in 1995 and is considered well-insulated. The house is a single story ranch style home with a finished basement. Total living area is approximately 2,000 ft^2. The home is also used to operate a small business, and is typically occupied the entire day.

Site Specifics:

DHP monitored - Mitsubishi model MUZFE09NA, rated for 9,000 Btu/h cooling and 10,900 Btu/h heating.

Location – Living Room; 16’x20’ – 320 ft^2

Other DHP – An identical unit is also installed in the finished basement.

Usage – Heating and cooling

Systems Replaced:

The two DHPs replaced five window A/C units of various sizes

Heating fuels displaced:

Two propane fireplace units

Home has no operational central heating system

DHP controls – remote control; no wall thermostat

Other thermostats – simple controls for fireplaces

System Operation:

The DHP was purchased primarily for heating, but is also used for cooling. The DHPs are now the primary heating and cooling systems for the home:

Heating:

- The monitored DHP heats the living room and contributes to other spaces during all times of heating demand.

- During the day, the remote control is typically set at 70-72°F with the fan speed set to "auto."
- The homeowner often manually selects a lower temperature at night.

Cooling:

The monitored DHP cools the living room and surrounding areas, replacing three window A/C units. Two other window units were replaced by an additional DHP that was not monitored.

For cooling the remote control is typically set at 75°F with the fan speed set to auto.

No set-forward setting is used.

Homeowner Comments:

The following was reported both during the logger installation. Homeowner has been unavailable for a post-monitoring interview:

The two DHPs installed now serve as the primary heating and cooling systems.

The propane fireplaces are used to supplement the heating both during extremely cold weather and also when the ambiance of a fire is desired.

Cooling performance is good and the two DHPs cool the entire house, replacing the five window A/C units.

Site #3: Meredith, NH

Site 3 is a year-round residence constructed in 1995 and is considered well-insulated. The house is a three story home which includes an in-law apartment. Total living area is approximately 6,500 ft^2. The monitored DHP serves the contiguous in-law apartment that encompasses 1,500 ft^2 of the 6,500 ft^2.

Site Specifics:

DHP monitored - Mitsubishi model MUZFE12NA1, rated for 12,000 Btu/h cooling and 13,600 Btu/h heating.

Location – Living Room; 12'x14' 168 ft^2

Other DHP – An identical unit is also installed in the living room of the main house.

Usage – Heating and cooling

Systems Replaced:

The monitored DHP replaced one window A/C unit of unknown size

Heating fuels displaced:

Home has an oil-fired boiler which serves as a central heating system for both the main house and the apartment.

DHP controls – remote control; no wall thermostat

Other thermostats – central heating system zone thermostat in the apartment living room.

System Operation:

The DHP were purchased for both heating and cooling. The DHPs now serve as the primary heating and cooling systems for the home. The central boiler is called for heating during cold periods when the DHPs are not able to maintain the set-point.

Heating:

- The monitored DHP heats the living room and contributes to other spaces within the apartment during all times of heating demand.
- During the day, the remote control is typically set at 70-72°F with the fan speed set to "auto."
- Set-back settings are not typically used.

Cooling:

The monitored DHP cools the living room and surrounding areas, replacing one window A/C unit.

For cooling the remote control is typically set at 75°F with the fan speed set to auto.

No set-forward setting is used.

Site #4: Northfield, NH

Site 4 is a single family year-round residence constructed during 2004 and is well insulated. The house is a wood frame single story ranch style home with a semi-finished basement and a total living area of 1,875 ft^2. The monitored DHP unit is a Mitsubishi MUZFE12NA1, which serves the living room of home. Two other identical DHPs serve the kitchen and the master bedroom

Site Specifics:

DHP monitored - Mitsubishi model MUZFE12NA1, rated for 12,000 Btu/h cooling and 13,600 Btu/h heating.

Location – Living Room; 15'x18' – 270 ft^2

Other DHPs – Two identical units are installed in the kitchen and master bedroom

Usage – Initially heating and cooling; now cooling only

Systems Replaced:

The monitored DHP replaced one 6,000 and one 10,000 Btu/hr window A/C units. These units were "gifted" to their adult children, and are in use elsewhere.

Heating fuels displaced:

Kerosene initially; none now.

DHP controls – remote control; no wall thermostat

Other thermostats – No central heating system. Each kerosene heater has its integral thermostat.

System Operation:

The DHPs were purchased primarily for cooling and humidity control associated with a family member's respiratory illness. The DHPs serve as the sole cooling systems for the home, but are not presently used for heating. Three direct vent kerosene heaters provide the majority of the heat, with contribution from a biomass pellet stove, and a propane fireplace.

Heating:

- The DHPs were used initially for heating, including our metering period. Heating usage has now been discontinued after "sticker shock" associated with the electric bill.
- Heat is now provided by three direct vent kerosene heaters, a pellet stove, and a propane fireplace.

Cooling:

The monitored DHP cools the living room, replacing one 10,000 Btu/hr window A/C unit.

The remote control is typically set at 64°F for health reasons. The fan speed is typically set to auto.

A small percentage of the time they use the DHP for dehumidification only, when they set the fan speed to medium.

No set-forward setting is used.

Homeowner Comments:

The following was reported both during the logger installation and during the post-monitoring interview:

The DHP systems were installed primarily for controlling heat and humidity, for health considerations, during the cooling season.

Following installation they used the three DHPs for heating, but have now discontinued this usage for heating after reviewing the electric bills. They did not indicate that they also considered the costs of the other fuels used.

They are very happy with the cooling performance and state that the units cool the entire house.

They are not aware of any cooling savings, stating that it was not the reason for the installations.

The replaced window air-conditioning units were "gifted" to their adult children, and are in use elsewhere.

Site #5: Tuftonboro, NH

Site 5 is a single family, year-round residence with a total living area of approximately 2,900 ft^2. The house is a wood frame, open concept, single story home with a partially finished basement. The DHP monitored is a Mitsubishi MUZFE12NA1, which serves the kitchen. Two additional DHPs serve the living room and bedroom.

Site Specifics:

DHP monitored - Mitsubishi model MUZFE12NA1, rated for 12,000 Btu/h cooling and 13,600 Btu/h heating.

Location – Kitchen/Dining; 540 ft^2

Other DHP – An identical unit is installed in the living room and a 9,000 Btu/hr unit is installed in a bedroom.

Usage – Heating and cooling

Systems Replaced:

The monitored DHP replaced one window A/C unit of unknown size

The home also had a central A/C system that was in disrepair and had not been used for some time. The owners considered repairing this system prior to deciding to install the DHPs.

Heating fuels displaced:

Home has a propane-fired boiler which serves as a central heating system.

A pellet stove is used during extremely cold weather.

DHP controls – remote control; no wall thermostat

Other thermostats – central heating system has a thermostat in the kitchen near the monitored DHP. Another thermostat is installed in the bedroom which is also served by a DHP.

System Operation:

The DHP was purchased for both heating and cooling. The DHPs now serve as the primary heating and cooling systems for the home. The central boiler, when operational, is called for heating during cold periods when the DHPs are not able to maintain the set-point. A pellet stove provides back-up heat.

Heating:

- The monitored DHP heats the kitchen and dining areas, and does not contribute substantially to other spaces which have their own DHPs.
- During the day, the remote control is typically set at 68°F with the fan speed set to "auto."
- Set-back settings are not used for the DHP.

- Boiler thermostats are now set to 50°F. Prior to installation of the DHPs these thermostats were set at 68°F with a 65°F nighttime set-back.

Cooling:

The monitored DHP cools the kitchen and dining area, replacing one window A/C unit of unknown size.

For cooling the remote control is typically set at 72°F with the fan speed set to auto.

No set-forward setting is used.

Homeowner Comments:

The following was reported both during the logger installation and during the post-monitoring interview:

- The owner was initially planning to install cooling only, but was attracted by the dual usage.

The three DHPs are now used as the primary heating and cooling source.

They are very happy with both heating and cooling performance and state that the units heat and cool the entire house well, except when it is "bitter cold" when they supplement with a pellet stove.

The propane-fired boiler recently failed, and after a delay was repaired. Owner reports greatly reduced propane usage due to the reliance on the DHP for primary heating.

The owner reports that pellet usage has been reduced 50% from 40 to 20 bags.

They are very pleased with the comfort level and have experienced no issues.

The replaced window air-conditioning units were "trashed."

The owner has not "bothered" to learn all the features of the remote control, including the set-back, set-forward features.

Site #6: Alton, NH

Site 6 is a single family, year-round residence built in 2005. The house is a wood frame, open concept, two-story colonial and is considered well insulated. Total living area is approximately 2,400 ft^2. The home has one DHP system installed which serves the kitchen and great room. The unit monitored is a Mitsubishi MUZFE18NA. The central heating system is an oil fired boiler.

Site Specifics:

DHP monitored - Mitsubishi model MUZFE18NA1, rated for 17,200 Btu/h cooling and 21,600 Btu/h heating.

Location – Kitchen/Great Room; 1250 ft^2

Other DHP – none

Usage – Heating and cooling

Systems Replaced: No existing systems were removed, but the homeowners were intending to install central A/C prior to receiving the DHP proposal.

Heating fuels displaced:

Home has an oil-fired boiler which serves as a central heating system.

DHP controls – remote control; no wall thermostat

Other thermostats – central heating system has a thermostat in the great room near the monitored DHP. Another thermostat is installed on the second floor.

System Operation:

The DHP was purchased for both heating and cooling. The DHP now serves as the primary heating and cooling system for the home. The central boiler is called for heating during cold periods when the DHP is not able to maintain the set-point.

Heating:

The monitored DHP heats the entire house except during periods of extreme cold when the central system contributes.

The remote control is typically set at 62°F with the fan speed set to "auto."

Set-back settings are not used for the DHP.

Boiler thermostats are now set to 58°F, making the DHP the primary heat source. Prior to installation of the DHPs these thermostats were set at 62°F.

They no longer use two electric space heaters due to the DHP performance.

Cooling:

The monitored DHP cools the entire house. No other A/C is installed or has been removed.

The house was not cooled prior to the installation of the DHPs, although the owners requested a proposal for central A/C, but alternatively decided to install the DHP.

For cooling the remote control is typically set at 72°F when the home is occupied and is manually raised to 76°F. The fan speed is set to auto.

No set-forward setting is used.

Homeowner Comments:

The following was reported both during the logger installation and during the post-monitoring interview:

- ❑ The owner was initially planning to install cooling and dehumidification only, but was attracted by the dual usage.

The DHP is now used as the primary heating and cooling source.

The owner reports “notable” heating oil savings.

The owner is very satisfied with system performance for both heating and cooling, stating that the central boiler is now used infrequently when outside temperatures are very low.

They are very pleased with the comfort level and have experienced no issues associated with airflow.

Site #7: Holderness, NH

Site 7 is a single family, year-round residence built in 1995/6. The house is a wood frame, open concept, 1 ½ story Cape style home and is considered well insulated, but the owner believes the insulation has deteriorated. Three windows and six patio doors were replaced in 2010. Total living area is approximately 1,600 ft^2. The DHP monitored is a Mitsubishi MUZFE18NA that is installed in the great room. Another similar 9,000 Btu/h DHP is installed in the master bedroom. The central heating system is an oil fired boiler.

Site Specifics:

DHP monitored - Mitsubishi model MUZFE18NA1, rated for 17,200 Btu/h cooling and 21,600 Btu/h heating.

Location – Great Room; 20’x30’ - 600 ft^2

Other DHP – One in bedroom used primarily for cooling

Usage – Heating and cooling

Systems Replaced: No existing systems were removed, but the homeowners were intending to install central A/C prior to receiving the DHP proposal.

Heating fuels displaced:

Home has an oil-fired boiler which serves as a central heating system.

DHP controls – remote control; no wall thermostat

Other thermostats – central heating system has a thermostat in the great room near the monitored DHP. Another thermostat is installed on the second floor.

System Operation:

The DHPs were purchased for both heating and cooling. The monitored DHP now serves as the primary heating and cooling system for the home. The second DHP is used almost exclusively for cooling in the bedroom. The central boiler system is called for heating during cold periods when the DHP is not able to maintain the set-point.

Heating:

The monitored DHP heats the entire house except during periods of extreme cold when the central system contributes.

The remote control is typically set at 69°F with the fan speed set to “auto.”

Set-back settings are not used for the DHP.

Boiler thermostats are now set to 68°F and 64°F for night setback making the DHP the primary heat source. Prior to installation of the DHPs these thermostats were set somewhat higher.

No space heaters are used.

Cooling:

The monitored DHP cools the entire house except the bedroom which has its own DHP. No other A/C is installed.

The house was not cooled prior to the installation of the DHPs.

For cooling the remote control is typically set at 75°F. The fan speed is set to auto.

No set-forward setting is used.

Homeowner Comments:

The following was reported both during the logger installation and during the post-monitoring interview:

The DHP is now used as the primary heating and cooling source.

The oil system now rarely fires, but is used for DHW.

The owner estimates heating oil savings of 160 gallons per heating season.

The owner is very satisfied with system performance for both heating and cooling, and states that the system exceeds his expectations.

They are very pleased with the comfort level and have experienced no issues associated with airflow.

Site #8: Sanbornville, NH

Site 8 is a single family, year-round residence built in 1986. The house is a wood frame, two-story colonial and is insulated to typical 1980s standards. No weatherization has been performed. Total living area is approximately 2,200 ft^2. The DHP monitored is a Mitsubishi MUZFE18NA that is installed in the living room. The home is also served by an oil fired hot water heating system.

Site Specifics:

DHP monitored - Mitsubishi model MUZFE18NA1, rated for 17,200 Btu/h cooling and 21,600 Btu/h heating.

Location – Kitchen/Foyer; 800 ft^2

Other DHP – One in bedroom used primarily for cooling

Usage – Heating and cooling

Systems Replaced: One 15,000 Btu/h window A/C unit. The unit is still in place, but reportedly not used and will be removed.

Heating fuels displaced:

Home has an oil-fired boiler which serves as a central heating system.

Wood stove fireplace insert.

DHP controls – remote control; no wall thermostat

Other thermostats – central heating system has a thermostat in the great room near the monitored DHP. Other thermostats are installed in a bedroom and a multi-use room.

System Operation:

The DHPs were purchased for both heating and cooling. The monitored DHP now serves as the primary heating and cooling system for the home. The central boiler system is called for heating during cold periods when the DHP is not able to maintain the set-point.

Heating:

The monitored DHP heats the entire house except during periods of extreme cold when the central system contributes.

The remote control is typically set at 72°F with the fan speed set to "auto."

Set-back settings are not used for the DHP.

Boiler thermostats are now set to 55°F and manually turned up when needed, making the DHP the primary heat source. Prior to installation of the DHPs these thermostats were set to 72°F.

The two bathrooms have electric heaters that are manually turned on when needed.

Cooling:

The monitored DHP cools the entire house.

A 15,000 Btu/h window air conditioner is in place but the owners say it is not used. No other A/C is installed or has been replaced.

The house was cooled prior to the installation of the DHP, but the present owners took possession just prior to the installation of the DHP.

For cooling the remote control is typically set at 72°F. The fan speed is set to auto.

No set-forward setting is used.

Homeowner Comments:

The following was reported both during the logger installation and during the post-monitoring interview:

The DHP is now used as the primary heating and cooling source and the owners report that it heats and cools the entire house.

The oil system is used during periods of extreme cold.

The owner cannot estimate savings as they moved into the house just prior to the DHP installation.

The owner is very satisfied with system performance for both heating and cooling.

They are very pleased with the comfort level and have experienced no issues associated with airflow, stating, “sometimes I stand in front of it and enjoy the heat flow.”

Site #9: Campton, NH

Site 9 is a single family, year-round residence built in 1850. A sunroom addition was constructed in 1995. The house is a wood frame, two-story colonial that is not well insulated or weatherized. Total living area is approximately 4,000 ft^2. The DHP monitored is a Mitsubishi MUZFE12NA that is installed in the sunroom. The sunroom was added about 20 years ago and is standard wood frame construction with fiberglass insulation. The home is also served by a central oil fired forced hot water heating system.

Site Specifics:

DHP monitored - Mitsubishi model MUZFE12NA1, rated for 12,000 Btu/h cooling and 13,600 Btu/h heating.

Location – Sunroom; 600 ft^2

Other DHP – None

Usage – Heating and cooling

Systems Replaced: Radiant slab heat that was served by the central boiler, is now disconnected.

Heating fuels displaced:

Home has an oil-fired boiler which serves as a central heating system.

DHP controls – remote control; no wall thermostat

Other thermostats – central heating system has a thermostat in the sunroom that is no longer functional, as it controlled the radiant heat.

System Operation:

The DHPs were purchased for both heating and cooling. The monitored DHP now serves as the sole heating and cooling system for the sunroom.

Heating:

The monitored DHP is used to heat the sunroom.

The remote control is typically set at 65°F with the fan speed set to “auto.”

Set-back settings are not used for the DHP.

Boiler thermostats are now set to 62°F. This setting has not changed with the installation of the DHP as it serves different spaces.

No electric heat or space heaters are installed.

Cooling:

The monitored DHP cools the sunroom.

No other cooling is installed in the house.

The house was not cooled prior to the installation of the DHP.

For cooling the remote control is typically set at 70°F. The fan speed is set to auto.

No set-forward setting is used.

Homeowner Comments:

The following was reported both during the logger installation and during the post-monitoring interview:

The owner is a contractor who installed the DHP as an “experiment.” He has been very impressed with the performance.

The DHP is used primarily to heat and cool the sunroom. During moderate heating conditions, the DHP also contributes additional heating to adjacent areas.

The owner has experienced no comfort issues associated with airflow.

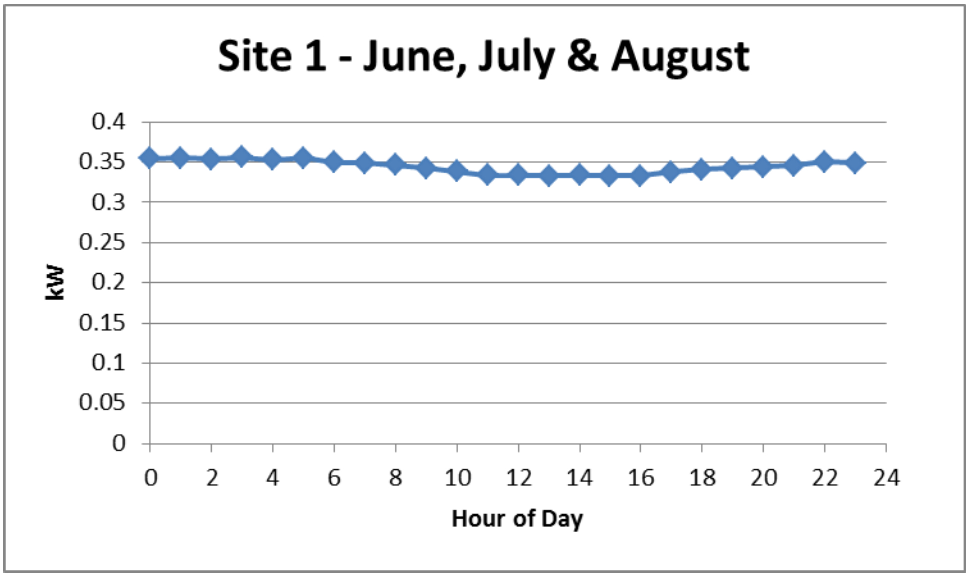
Appendix C

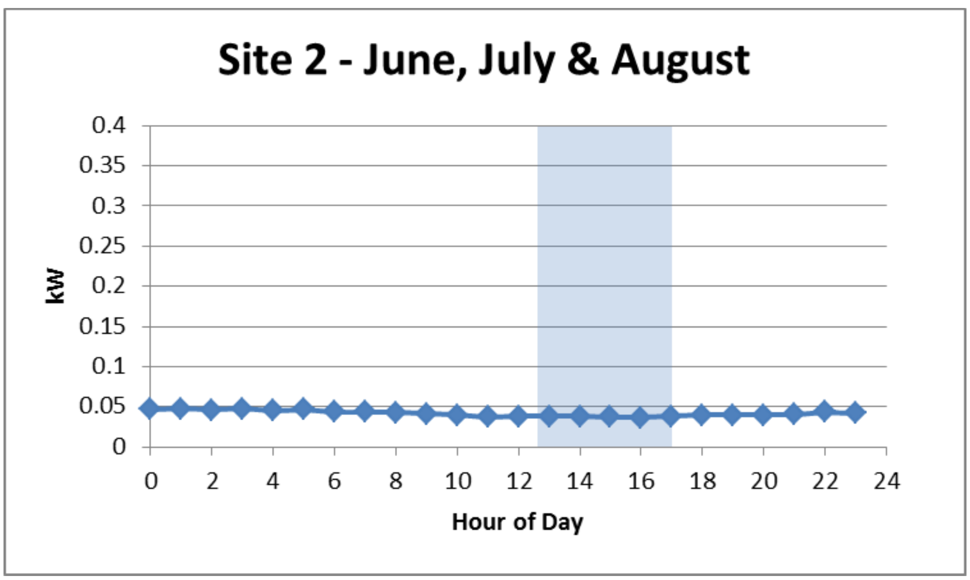
Single-Site Cooling Load Shapes

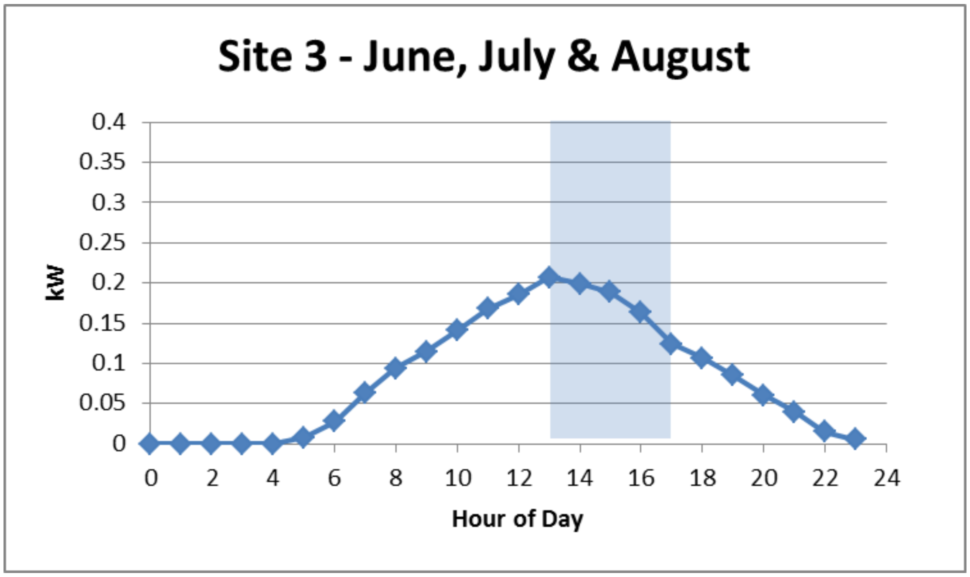
APPENDIX C

Single-Site Cooling Load Shapes

The following load shapes are for individual sites, per ton of cooling, normalized with TMY3 weather data. The load shapes are for average hourly demand. In order to establish a savings load shape, a similar load shape for the baseline technology would be developed for comparison. This data is also presented in Section 5 of the report.



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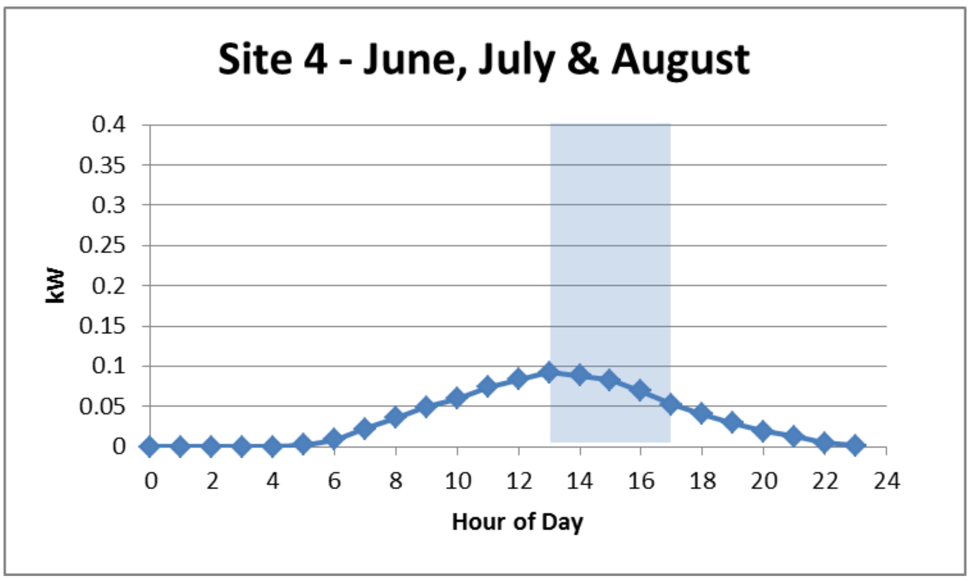
kW
0.4
0.35
0.3
0.25
0.2
0.15
0.1
0.05
0
0
2
4
6
8
10
12
14
16
18
20
22
24
Hour of Day


Site 4 - June, July & August
kW
0.4
0.35
0.3
0.25
0.2
0.15
0.1
0.05
0
0
2
4
6
8
10
12
14
16
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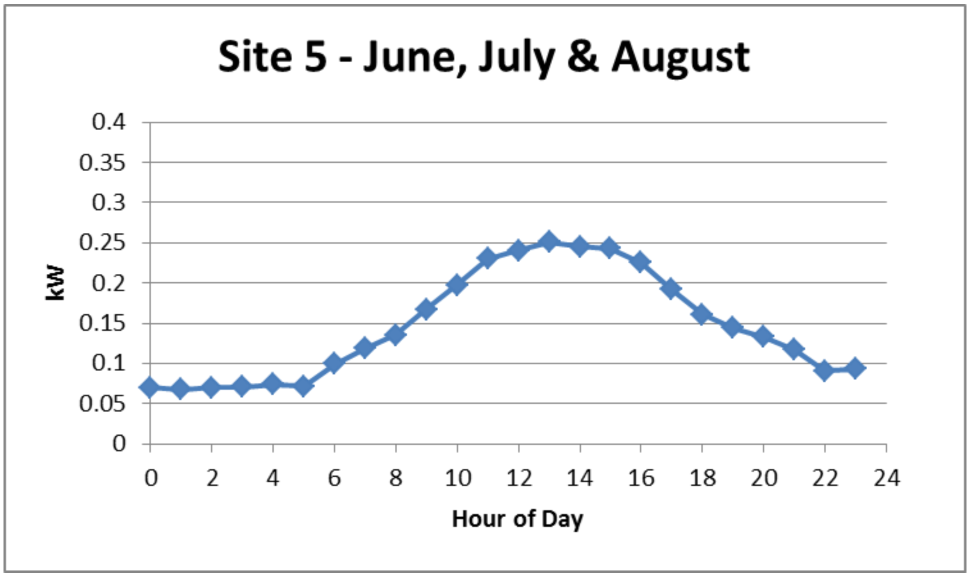
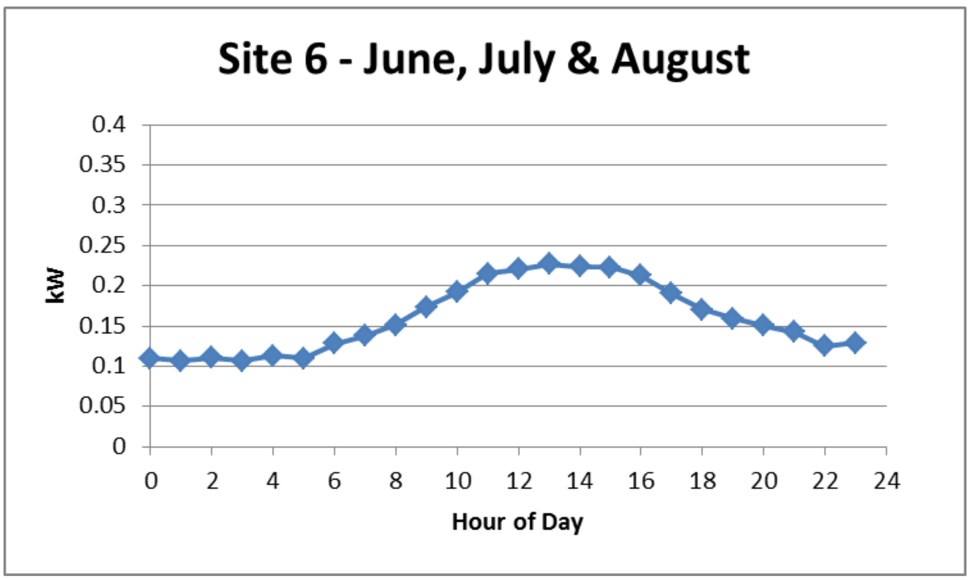
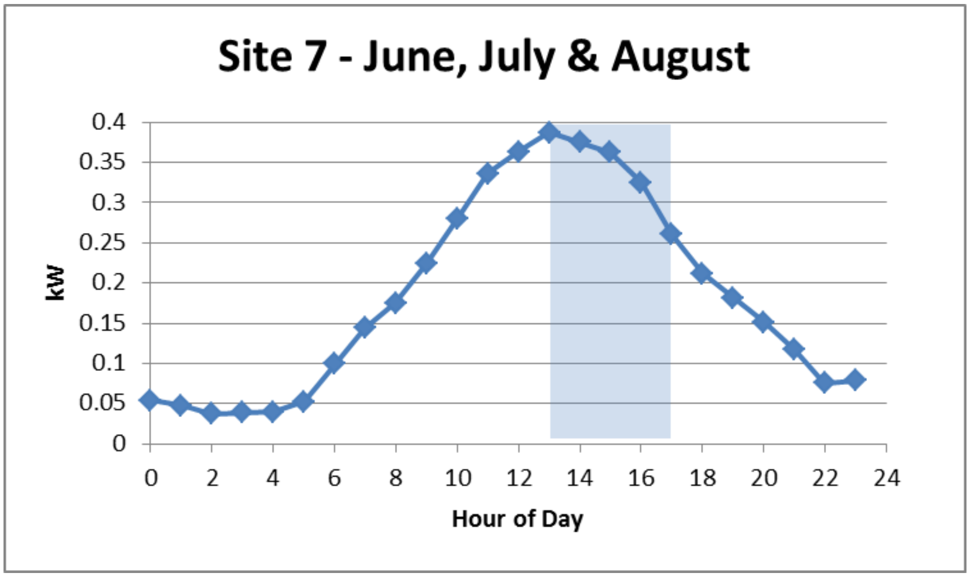
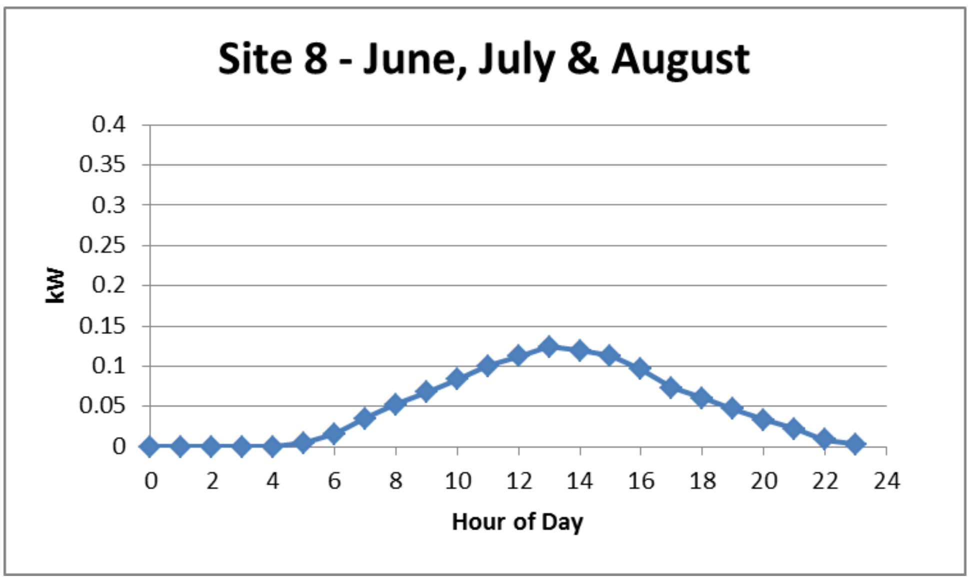
20
22
24
Hour of Day


Site 5 - June, July & August
kW
0.4
0.35
0.3
0.25
0.2
0.15
0.1
0.05
0
0 2 4 6 8 10 12 14 16 18 20 22 24
Hour of Day

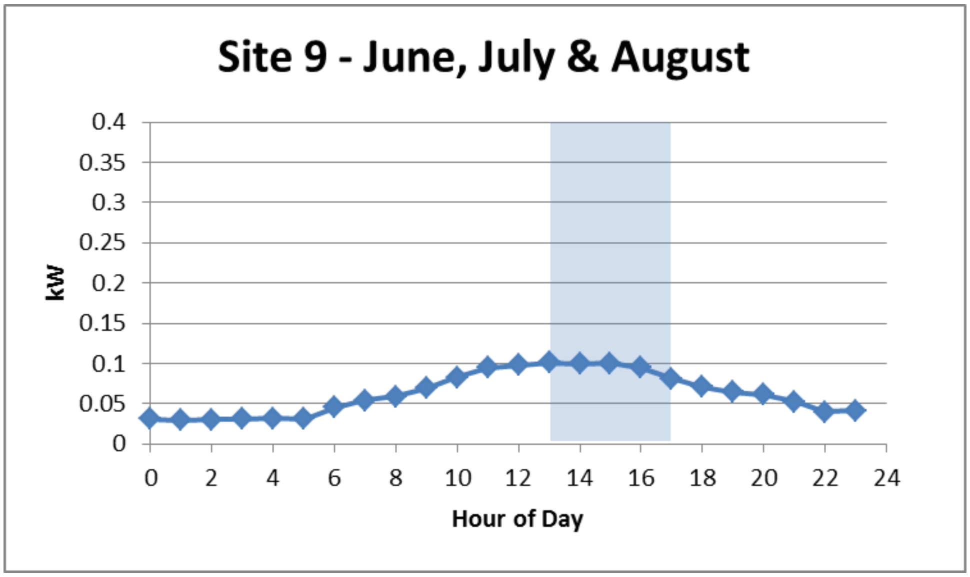

Site 6 - June, July & August
kW
0.4
0.35
0.3
0.25
0.2
0.15
0.1
0.05
0
0 2 4 6 8 10 12 14 16 18 20 22 24
Hour of Day



Site 7 - June, July & August
kW
0.4
0.35
0.3
0.25
0.2
0.15
0.1
0.05
0
0
2
4
6
8
10
12
14
16
18
20
22
24
Hour of Day



Site 8 - June, July & August
kW
0.4
0.35
0.3
0.25
0.2
0.15
0.1
0.05
0
0
2
4
6
8
10
12
14
16
18
20
22
24
Hour of Day


Site 9 - June, July & August
kW
0.4
0.35
0.3
0.25
0.2
0.15
0.1
0.05
0
0
2
4
6
8
10
12
14
16
18
20
22
24
Hour of Day

Appendix D

DHP Performance Test Report

Intertek
3933 US Route 11
Cortland, NY 13045-9717

AHRI Certification No.: USHP-10621-1
Intertek Report No.: 100016615CRT-290
Intertek Private Report No.: 100210869CRT-004

UNITARY AIR-CONDITIONER AND AIR-SOURCE UNITARY HEAT PUMP CERTIFICATION PROGRAM
A PROGRAM OF THE AIR-CONDITIONING, HEATING, AND REFRIGERATION INSTITUTE
CERTIFICATION TEST REPORT OF A HEAT PUMP

Mitsubishi Electric and Electronics USA, Inc. Attn: Paul L. Doppel 3400 Lawrenceville-Suwanee Road Suwanee, Georgia 30024	Air-Conditioning, Heating, and Refrigeration Institute Attn: Mr. Eric Chen 2111 Wilson Blvd., Suite 500 Arlington, VA 22201-3001

Manufacturer: Mitsubishi Electric and Electronics USA, Inc.		**AHRI Reference No.:** 3577498
Indoor Model No.: MSZ-FE12NA		**SERIAL NO.:** 0000629
Outdoor Model No.: MUZ-FE12NA		**SERIAL NO.:** 0003746
TYPE: HRCU-A-CB-O	**HSVTC:** No	**TEST DATE:** October 15 – 17, 2010

SUMMARY
COOLING CYCLE

SECTION NO *	TITLE OF TEST	TEST RUN	VOLTAGE VOLTS	CERTIFIED	MEASURED	RATING RATIO %	RESULT OF TEST #
6.1	STANDARD RATING	A_2	230	12,000	12,691	105.8	P
	ENERGY EFFICIENCY RATIO, EER II	A_2	230	12.90	13.32	103.3	P
	COOLING STEADY STATE, BTU/HR	B_2	230	12,900	13,550	-	-
	COOLING STEADY STATE, BTU/HR	E_V	230	-	7,606	-	-
	COOLING STEADY STATE, BTU/HR	B_1	230	-	3,868	-	-
	COOLING STEADY STATE, BTU/HR	F_1	230	-	4,299	-	-
6.1.2	SEASONAL ENERGY EFFICIENCY RATIO	-	230	23.00	**24.54	106.7	P
6.1.5	MINIMUM EXTERNAL PRESSURE						
	AIR FLOW, SCFM	A_2	-	420	344	-	-
	STATIC PRESSURE, IN. W.G.	A_2	-	-	0.00	-	-
	INDOOR MOTOR SPEED SETTING	A_2	230	High	-	-	-

*AHRI STANDARD 1230-2010 DOE Covered: Yes ** Cooling C_D = 0.25 (Not Performed)

CODE -
- **P** – Passed
- **NP** - Not Performed
- **ND** - No Decision
- **MD** - Model Discontinued
- **TBR** - To Be Recertified
- **SR** - See "Remarks"
- **RTM** - Returned To Manufacturer
- **F** - Failed
- **D** – Defective

II - Verified, however not a certified value under the AHRI Certification Program

SUMMARY
HEATING CYCLE

SECTION NO *	TITLE OF TEST	TEST RUN	VOLTAGE VOLTS	CERTIFIED	MEASURED	RATING RATIO %	RESULT OF TEST #
6.1	HEATING CAPACITY, BTU-HR (HIGH TEMP)	$H1_2$	230	13,600	13,731	101.0	P
	COEFFICIENT OF PERFORMANCE, COP	$H1_2$	230	4.20	4.24	-	-
	AIR FLOW, SCFM	$H1_2$	230	-	379	-	-
	STATIC PRESSURE, IN. W.G.	$H1_2$	230	-	0.00	-	-
	HEATING CAPACITY, BTU-HR (LOW TEMP)	$H3_2$	230	8,300	8,322	-	-
	COEFFICIENT OF PERFORMANCE, COP	$H3_2$	230	3.04	3.07	-	-
	FROST ACCULUMATION CAPACITY, BTU/HR	$H2_V$	230	-	5,666	-	-
	COEFFICIENT OF PERFORMANCE, COP	$H2_V$	230	-	4.71	-	-
	HEATING CAPACITY, BTU-HR (HIGH TEMP)	$H1_1$	230	-	4,194	-	-
	COEFFICIENT OF PERFORMANCE, COP	$H1_1$	230	-	6.03	-	-
	HEATING CAPACITY, BTU/HR (HI-HI TEMP)	$H0_1$	230	-	5,334		
	COEFFICIENT OF PERFORMANCE, COP	$H0_1$	230	-	8.27	-	-
6.1.2	HEATING SEASONAL PERFORMANCE FACTOR	-	230	10.50	**12.31	117.3	P

**AHRI STANDARD 1230-2010

** Heating C_D = 0.25 (Not Performed)
** Demand Defrost Factor = 1.0000

CODE

MD - Model Discontinued
NP - Not Performed
ND - No Decision

RTM - Returned to Manufacturer
TBR - To Be Recertified
SR - See "Remarks"

P - Passed
F - Failed
D – Defective

Remarks

The client supplied the sample, which Intertek received on September 16, 2010. Testing was conducted at Intertek, 3933 U.S. Route 11, Cortland NY 13045.

The following installation instructions were provided by the manufacturer and were considered adequate for their intended purpose:

- Mitsubishi Split-Type Air Conditioners Installation Manual.
- Mitsubishi Electric Split-Type Air Conditioners Indoor Unit Operating Instructions.

The condition of the test specimen was 'new'. It was subjected to Test Series E and met the requirements for certification of an air source unitary heat pump in accordance with AHRI STANDARD 1230-2010 and Section 8 of ANSI/ASHRAE Standard 37-2005.

The EER at test condition "A" was verified.

Test Supervised by:

Sam Ando

Sam Ando
Technician Team Leader

Report Approved by:

James A. Harvey
Associate Engineer

Copied by: cnv

COPY

TEST_NAME = Cool_A_CD	Test Date = 10/15/2010	Trade Name = MR.SLI
Manf. = MITSUBISHI ELECTRIC AN	Operator = P.M.MARCEY	ID Serial = 0000629
ARI No. = USHP-10621-1	Order No. = G100016615/290	OD Serial = 0003746
ID Model = MSZ-FE12NA	OD Model = MUZ-FE12NA	Type = HRCU-A-CB-0
Blower Model = "A-2" TEST	Test Voltage = 230	Recvd = 09/16/10
Rated Capacity = 12000 PM	Rated EER = 12.9 PM	Filter = YES
Rated Airflow = 420 PM	Rated SEER = 23.0 PM	Blower Speed = HIGH

ID Inlet DB	= 79.794 DegF
ID Inlet WB	= 67.128 DegF
ID Outlet DB	= 56.913 DegF
ID Outlet WB	= 55.802 DegF
*ID Static	= .0011357 In/H2o
ID B4 Nozzle	= -.029522 In/H2o
ID Nozzle Delta P	= 1.1019 In/H2o
ID Unit Airflow	= 359.53 CFM
*ID Unit Standard Airflow	= 343.72 SCFM
*ID Total Cooling Capacity	= 12691. BTU/h
ID Sensible Cooling Capacity	= 8822.0 BTU/h
ID Sensible/Total Ratio	= 69.515
OD Inlet DB	= 94.847 DegF
OD Inlet WB	= 72.939 DegF
OD Outlet DB	= 109.04 DegF
OD Outlet WB	= 77.583 DegF
OD Static	= .0026551 In/H2o
OD B4 Nozzle	= -.072600 In/H2o
OD Nozzle Delta P	= .98510 In/H2o
OD Unit Airflow	= 1129.7 CFM
*OD Unit Standard Airflow	= 982.05 SCFM
*OD Sensible Cooling Capacity	= 12294. BTU/h
ID Voltage	= .0 Volts
ID Amperage	= .0 Amps
*ID Wattage	= .0 Watts
OD Voltage	= 230.03 Volts
OD Amperage	= 4.3282 Amps
*OD Total Wattage	= 952.51 Watts
*Total Test Unit Power	= 952.52 Watts
*EER	= 13.323 BTU/W
Heat Balance	= 3.1142 %
Percent of Rated Airflow	= 81.838 %
Percent of Rated EER	= 103.33 %
Percent of Rated Capacity	= 105.75 %
Discharge at Compressor	= 158.10 DegF
Vapor at Outdoor Coil	= 98.855 DegF
Liquid at Outdoor Coil	= 65.348 DegF
Liquid at Indoor Coil	= 56.779 DegF
Vapor at Indoor Coil	= 49.028 DegF
Suction at Service Port	= 48.459 DegF
Suction at Compressor	= 51.328 DegF
Highside Refrigerant Pressure	= 2.0596 psig
Lowside Refrigerant Pressure	= 135.84 psig

COPY

TEST_NAME = Heat_HiTemp_CD	Test Date = 10/16/2010	Trade Name = MR.SLI
Manf. = MITSUBISHI ELECTRIC AN	Operator = JEREMY VANCISE	ID Serial = 0000629
ARI No. = USHP-10621-1	Order No. = G100016615/290	OD Serial = 0003746
ID Model = MSZ-FE12NA	OD Model = MUZ-FE12NA	Type = HRCU-A-CB-0
Blower Model = "H1-2" TEST	Test Voltage = 230	Recvd = 09/16/10
Rated Capacity = 13600 JV	Rated COP = 4.2 JV	Filter = YES
Rated Airflow = 420 JV	Rated SEER =	Blower Speed = HIGH

ID Inlet DB	=	69.875 DegF
ID Inlet WB	=	54.580 DegF
ID Outlet DB	=	102.91 DegF
ID Outlet WB	=	67.084 DegF
*ID Static	=	.0009099 In/H2o
ID B4 Nozzle	=	-.031275 In/H2o
ID Nozzle Delta P	=	1.4153 In/H2o
ID Unit Airflow	=	420.66 CFM
*ID Unit Standard Airflow	=	378.72 SCFM
*ID Total Sensible Heating Capacity	=	13731. BTU/h
ID Sensible/Total Ratio	=	100.00
OD Inlet DB	=	47.172 DegF
OD Inlet WB	=	42.981 DegF
OD Outlet DB	=	39.611 DegF
OD Outlet WB	=	38.595 DegF
OD Static	=	.0035124 In/H2o
OD B4 Nozzle	=	-.10850 In/H2o
OD Nozzle Delta P	=	1.1161 In/H2o
OD Unit Airflow	=	1131.2 CFM
*OD Unit Standard Airflow	=	1139.2 SCFM
*OD Total Heating Capacity	=	14032. BTU/h
ID Voltage	=	.0 Volts
ID Amperage	=	.0 Amps
*ID Wattage	=	.0 Watts
OD Voltage	=	230.04 Volts
OD Amperage	=	4.3251 Amps
*OD Total Wattage	=	950.44 Watts
*Total Test Unit Watts	=	950.49 Watts
*Total Test Unit w/h	=	950.49 W/h
Delta T Factor	=	1.0280 1 ±.06
Delta T Integrator	=	33.051 DegF
Delta T RTD	=	33.043 DegF
*COP	=	4.2364 W/W
Heat Balance	=	-2.1870 %
Percent of Rated Airflow	=	90.169 %
Percent of Rated COP	=	100.86 %
Percent of Rated Capacity	=	100.96 %
Discharge at Compressor	=	156.23 DegF
Vapor at Outdoor Coil	=	57.177 DegF
Liquid at Outdoor Coil	=	78.496 DegF
Liquid at Indoor Coil	=	78.391 DegF
Vapor at Indoor Coil	=	143.26 DegF
Suction at Service Port	=	148.58 DegF
Suction at Compressor	=	36.071 DegF
Highside Refrigerant Pressure	=	.72724 psig
Lowside Refrigerant Pressure	=	1.1626 psig

Appendix E

AHRI Testing Procedure

ANSI/AHRI Standard 1230

2010 Standard for

Performance Rating of Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment

Approved by ANSI on 2 August 2010

Air-Conditioning, Heating, and Refrigeration Institute

2111 Wilson Boulevard, Suite 500
Arlington, VA 22201, USA
www.ahrinet.org

PH 703.524.8800
FX 703.562.1942

IMPORTANT

SAFETY DISCLAIMER

AHRI does not set safety standards and does not certify or guarantee the safety of any products, components or systems designed, tested, rated, installed or operated in accordance with this standard/guideline. It is strongly recommended that products be designed, constructed, assembled, installed and operated in accordance with nationally recognized safety standards and code requirements appropriate for products covered by this standard/guideline.

AHRI uses its best efforts to develop standards/guidelines employing state-of-the-art and accepted industry practices. AHRI does not certify or guarantee that any tests conducted under its standards/guidelines will be non-hazardous or free from risk.

Important Note:

Until AHRI Standard 1230 is approved by DOE,VRF multi-split air-cooled air conditioners and heat pumps, below 65,000 Btu/h [19,000 W] shall be rated in accordance with ARI Standard 210/240-2008.

Applicability

Integrated Energy Efficiency Ratio (IEER) is effective beginning January 1, 2010. Integrated Part-Load Value is in effect until January 1, 2010. On January 1, 2010, IEER will supersede IPLV.

AHRI CERTIFICATION PROGRAM PROVISIONS

Scope of the Certification Program

The Certification Program includes all Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment rated at AHRI Standard Rating Conditions (Cooling).

Certified Ratings

The following Certification Program ratings are verified by test:

Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment

a. For VRF Multi-Split Air-Conditioners < 65,000 Btu/h [19,000 W]

1. Standard Rating Cooling Capacity, Btu/h [W]
2. Seasonal Energy Efficiency Ratio, SEER, Btu/(W·h)

b. For VRF Multi-Split Air-Conditioners ≥ 65,000 Btu/h [19,000 W]

1. Standard Rating Cooling Capacity, Btu/h [W]
2. Energy Efficiency Ratio, EER, Btu/(W·h)
3. Integrated Energy Efficiency Ratio, IEER (Integrated Part-Load Value, IPLV is Superseded by IEER January 1, 2010)

c. For VRF Multi-Split Heat Pumps < 65,000 Btu/h [19,000 W]

1. Standard Rating Cooling Capacity, Btu/h [W]
2. Seasonal Energy Efficiency Ratio, SEER, Btu/(W·h)
3. High Temperature Heating Standard Rating Capacity, Btu/h [W]
4. Region IV Heating Seasonal Performance Factor, HSPF, Minimum Design Heating Requirement, Btu/(W·h)

d. For VRF Multi-Split Heat Pumps ≥ 65,000 Btu/h [19,000 W]

1. Standard Rating Cooling Capacity, Btu/h [W]
2. Energy Efficiency Ratio, EER, Btu/(W·h)
3. Integrated Energy Efficiency Ratio, IEER (Integrated Part-Load Value, IPLV is Superseded by IEER January 1, 2010)
4. High Temperature Heating Standard Rating Capacity, Btu/h [W]
5. High Temperature Coefficient of Performance, COP
6. Low Temperature Heating Standard Rating Capacity, Btu/h [W]
7. Low Temperature Coefficient of Performance, COP

e. For VRF Multi-Split Heat Recovery Heat Pumps

1. Ratings Appropriate in (c) and (d) above
2. Simultaneous Cooling and Heating Efficiency (SCHE) (50% heating/50% cooling).

f. For VRF Multi-Split Heat Pump Systems that Use a Water Source for Heat Rejection

1. Standard Rating Cooling Capacity, Btu/h [W]
2. Energy Efficiency Ratio, EER, Btu/(W·h)
3. Integrated Energy Efficiency Ratio, IEER (Integrated Part-Load Value, IPLV is Superseded by IEER January 1, 2010)
4. Heating Standard Rating Capacity, Btu/h [W]
5. Heating Coefficient of Performance, COP
6. Simultaneous Cooling and Heating Efficiency (SCHE) (50% heating/50% cooling)(Heat Recovery models only)

Conformance to the requirements of the Maximum Operating Conditions Test, Voltage Tolerance Test, Low-Temperature Operation Test (Cooling), Insulation Effectiveness Test (Cooling), and Condensate Disposal Test (Cooling), as outlined in Section 8, are also verified by test.

NOTE:
THIS IS A NEW STANDARD.

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Performance Rating of Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment

Section 1. Purpose

1.1 *Purpose.* The purpose of this standard is to establish for Variable Refrigerant Flow (VRF) Multi-Split Air Conditioners and Heat Pumps: definitions; classifications; test requirements; rating requirements; minimum data requirements for Published Ratings; operating requirements; marking and nameplate data; and conformance conditions.

1.1.1 *Intent.* This standard is intended for the guidance of the industry, including manufacturers, engineers, installers, contractors and users.

1.1.2 *Review and Amendment.* This standard is subject to review and amendment as technology advances.

Section 2. Scope

2.1 This standard covers matched variable refrigerant flow Multi-Split Air Conditioners and Multi-Split Heat Pumps using distributed refrigerant technology with cooling and heating capacities for outdoor units from 12,000 Btu/h [3508 W] to 300,000 Btu/h [90,000 W] and indoor units from 5,000 Btu/h [1,000W] to 60,000 Btu/h [20,000 W]. Each indoor unit is designed to condition a single zone.

2.2 This standard applies to variable refrigerant flow multi-split systems consisting of the following matched components: a) an outdoor unit with single or multiple compressors or variable capacity compressor or with a variable speed drive; b) indoor unit(s) that have a coil, air movement device intended for single zone air distribution, and a temperature sensing control; and c) a zone temperature control device.

2.3 The multi-split systems covered in this standard are Variable Refrigerant Flow (VRF) Multi-Split Systems and Heat Recovery (VRF) Multi-Split Systems. Included are multi-split, matched system air conditioners and heat pumps irrespective of their type of electric power source, type of refrigeration cycle, or secondary fluid (e.g. air-to-air or water-to-air)

2.4 This standard does not apply to the testing and rating of individual assemblies for separate use. It also does not cover ductless mini-splits (one-to-one split systems) which are covered by AHRI Standard 210/240.

2.5 Energy Source. This standard applies only to electrically operated, vapor compression refrigeration systems.

NOTE: For the purpose of the remaining clauses, the terms equipment and systems will be used to mean multi-split air-conditioners and/or multi-split heat pumps that are described in 2.1 to 2.5.

Section 3. Definitions

All terms in this document shall follow the standard industry definitions established in the current edition of *ASHRAE Terminology of Heating, Ventilation, Air Conditioning and Refrigeration*, unless otherwise defined in this section.

For the purposes of this Standard, the following definitions apply

3.1 *Standard Air*. Air weighing 0.075 lb/ft^3 [1.2 kg/m^3] which approximates dry air at 70°F [21°C] and at a barometric pressure of 29.92 in Hg [101.3 kPa].

3.2 *Multi-Split Air-Conditioner*. An encased, factory-made assembly or assemblies designed to be used as permanently installed equipment to provide conditioned air to an enclosed space(s). It includes a prime source of refrigeration for cooling and dehumidification and may optionally include other means for heating, humidifying, circulating and cleaning the air. It normally includes multiple evaporator(s), compressor(s), and condenser(s). Such equipment may be provided in more than one assembly, the separated assemblies of which are intended to be used together.

3.3 *Capacity,*

3.3.1 *Full Capacity*. The capacity of the system when all indoor units and outdoor units are operated in the same mode, at their rated capacity in Btu/h [W].

3.3.2 *Heating Capacity*. The amount of heat the equipment can add to the conditioned space in a defined interval of time in Btu/h [W].

3.3. 3 *Latent Cooling Capacity*. Capacity associated with a change in humidity ratio.

3.3.4 *Sensible Cooling Capacity*. The amount of sensible heat the equipment can remove from the conditioned space in a defined interval of time in Btu/h [W].

3.3.5 *Total Cooling Capacity*. The amount of sensible and latent heat the equipment can remove from the conditioned space in a defined interval of time in Btu/h [W].

3.4 *Coefficient of Performance (COP)*. A ratio of the heating capacity in watts [W] to the power input values in watts [W] at any given set of rating conditions expressed in watts/watts [W/W]. For heating *COP*, supplementary resistance heat shall be excluded.

3.5 *Degradation Coefficient (C_D).* The measure of the efficiency loss due to the on/off cycling of the complete system as determined in Appendices C, D and G.

3.6 *Effective Power Input (P_E)*. Average electrical power input to the equipment expressed in watts [W] and obtained from:

a) Power input for operation of the compressor
b) Power input to electric heating devices used only for defrosting
c) Power input to all control and safety devices of the equipment
d) Power input to factory installed condensate pumps and
e) Power input for operation of all fans and, if applicable, any water-cooled condenser pump(s).

3.7 *Energy Efficiency Ratio (EER)*. A ratio of the Total Cooling Capacity in Btu/h to the power input values in watts [W] at any given set of rating conditions expressed in Btu/W·h.

3.8 *Ground-Water Heat Pump.* Water-to-air heat pump using water pumped from a well, lake, or stream functioning as a heat source/heat sink. The temperature of the water is related to the climatic conditions and may vary from 41° to 77°F [5° to 25°C] for deep wells.

3.9 *Ground-Loop Heat Pump.* Brine-to-air heat pump using a brine solution circulating through a subsurface piping loop functioning as a heat source/heat sink. The heat exchange loop may be placed in horizontal trenches, vertical bores, or be submerged in a body of surface water. (ANSI/ARI/ASHRAE ISO Standard 13256-1:1998) The temperature of the brine is related to the climatic conditions and may vary from 23° to 104°F [–5° to 40°C].

3.10 *Multi-Split Heat Pump*. One or more factory-made assemblies designed to be used as permanently installed equipment to take heat from a heat source and deliver it to the conditioned space when heating is desired. It may be constructed to remove heat from the conditioned space and discharge it to a heat sink if cooling and dehumidification are desired from the same equipment. It normally includes multiple indoor conditioning coils, compressor(s), and outdoor coil(s). Such equipment may be provided in more than one assembly, the separated assemblies of which are intended to be used together. The equipment may also provide the functions of cleaning, circulating and humidifying the air.

3.11 *Heating Seasonal Performance Factor (HSPF).* The total heating output of a heat pump, including supplementary electric heat, necessary to achieve building heating requirements during its normal annual usage period for heating divided by the total electric power during the same period, as determined in Appendix C expressed in Btu/[W·h].

3.12 *Heating Unit.* A component of a VRF Multi-Split System air conditioner or heat pump that is designed to transfer heat between the refrigerant and the indoor air, and which consists of an indoor coil, a cooling mode expansion device, an air moving device, and a temperature sensing device.

3.13 *Integrated Energy Efficiency Ratio (IEER).* A single number that is a cooling part-load efficiency figure of merit calculated per the method described in paragraph 6.5.

3.14 *Integrated Part-Load Value (IPLV).* A single number that is a cooling part-load efficiency figure of merit calculated per the method described in Appendix H.

3.15 *Mini-Split Air-Conditioners and Heat Pumps*. Systems that have a single outdoor section and one or more indoor sections. The indoor sections cycle on and off in unison in response to a single indoor thermostat (As defined by DOE, See Appendix C, Paragraph 1.29).

3.16 *Multiple-Split Air-Conditioners and Heat Pumps [a.k.a .Multi-Split Air Conditioners and Heat Pumps]*. Systems that have two or more indoor sections. The indoor sections operate independently and can be used to condition multiple zones in response to multiple indoor thermostats (As defined by DOE, See Appendix C, Paragraph 1.30).

3.17 *Non-Ducted System*. An air conditioner or heat pump that is designed to be permanently installed equipment and directly heats or cools air within the conditioned space using one or more indoor coils that are mounted on room walls and/or ceilings. The unit may be of a modular design that allows for combining multiple outdoor coils and compressors to create one overall system. Non-ducted systems covered by this test procedure are all split systems.

3.18 *Oil Recovery Mode*. An automatic system operation that returns oil to the compressor crank case when the control system determines oil recovery is needed.

3.19 *Outdoor Unit*. A component of a split-system central air conditioner or heat pump that is designed to transfer heat between refrigerant and air, or refrigerant and water, and which consists of an outdoor coil, compressor(s), an air moving device, and in addition for heat pumps, a heating mode expansion device, reversing valve, and defrost controls.

3.20 *Published Rating.* A statement of the assigned values of those performance characteristics, under stated rating conditions, by which a unit may be chosen to fit its application. These values apply to all systems of like nominal size and type produced by the same manufacturer. As used herein, the term Published Rating includes the rating of all performance characteristics shown on the unit or published in specifications, advertising or other literature controlled by the manufacturer, at stated Rating Conditions.

3.20.1 *Application Rating.* A rating based on tests performed at application Rating Conditions (other than Standard Rating Conditions).

3.20.2 *Standard Rating.* A rating based on tests performed at Standard Rating Conditions.

3.21 *Seasonal Energy Efficiency Ratio (SEER).* The total cooling of a system covered by this standard with a capacity <65,000 Btu/h [19,000 W] during its normal usage period for cooling (not to exceed 12 months) divided by the total electric energy input during the same period as determined in Appendix C, expressed in Btu/[W·h].

3.22 *"Shall" or "Should".* "Shall" or "should" shall be interpreted as follows:

3.22.1 *Shall.* Where "shall" or "shall not" is used for a provision specified, that provision is mandatory if compliance with the standard is claimed.

3.22.2 *Should.* "Should" is used to indicate provisions which are not mandatory but which are desirable as good practice.

3.23 *Simultaneous Cooling and Heating Efficiency (SCHE).* The ratio of the total capacity of the system (heating and cooling capacity) to the effective power when operating in the heat recovery mode. (Where SCHE is stated without an indication of units, it shall be understood that it is expressed in Btu/[W·h].)

3.24 *System Controls.* The following items characterize system controls:

a. An integral network operations and communications system with sensors to monitor and forecast the status of items such as temperature, pressure, oil, refrigerant levels and fan speed.

b. A micro-processor, algorithm-based control scheme to: (1) communicate with an optimally managed variable capacity compressor, fan speed of indoor units, fan speed of the outdoor unit, solenoids, various accessories; (2) manage metering devices; and (3) concurrently operate various parts of the system.

c. These controls optimize system efficiency and refrigerant flow through an engineered distributed refrigerant system to conduct zoning operations, matching capacity to the load in each of the zones.

3.25 *Tested Combination.* A sample basic model comprised of units that are production units, or are representative of production units, of the basic model being tested. The tested combination shall have the following features:

a. The basic model of a variable refrigerant flow system ("VRF system") used as a tested combination shall consist of an outdoor unit (an outdoor unit can include multiple outdoor units that have been manifolded into a single refrigeration system, with a specific model number) that is matched with between 2 and 5 indoor units (for systems with nominal cooling capacities greater than 150,000 Btu/h [43,846 W], the number of indoor units may be as high as 8 to be able to test non-ducted indoor unit combinations)

b. The indoor units shall:

b.1 Represent the highest sales model family as determined by type of indoor unit e.g. ceiling cassette, wall-mounted, ceiling concealed. etc. If 5 are insufficient to reach capacity another model family can be used for testing.

b.2 Together, have a nominal cooling capacity between 95% and 105% of the nominal cooling capacity of the outdoor unit.

b.3 Not, individually, have a nominal cooling capacity greater than 50% of the nominal cooling capacity of the outdoor unit, unless the nominal cooling capacity of the outdoor unit is 24,000 Btu/h [7016 W] or less.

b.4 Have a fan speed that is consistent with the manufacturer's specifications.

b.5 All be subject to the same minimum external static pressure requirement while being configurable to produce the same static pressure at the exit of each outlet plenum when manifolded as per section 2.4.1 of 10 CFR Part 430, Subpart B, Appendix M.

3.26 *Variable Refrigerant Flow (VRF) System.* An engineered direct exchange (DX) multi-split system incorporating at least one variable capacity compressor distributing refrigerant through a piping network to multiple indoor fan coil units each capable of individual zone temperature control, through proprietary zone temperature control devices and common communications network. Variable refrigerant flow implies three or more steps of control on common, inter-connecting piping.

3.27 *VRF Multi-Split System*. A split system air-conditioner or heat pump incorporating a single refrigerant circuit, with one or more outdoor units, at least one variable speed compressor or an alternative compressor combination for varying the capacity of the system by three or more steps, multiple indoor fan coil units, each of which is individually metered and individually controlled by a proprietary control device and common communications network. The system shall be capable of operating either as an air conditioner or a heat pump. Variable refrigerant flow implies three or more steps of control on common, inter-connecting piping.

3.28 *VRF Heat Recovery Multi-Split System*. A split system air-conditioner or heat pump incorporating a single refrigerant circuit, with one or more outdoor units at least one variable-speed compressor or an alternate compressor combination for varying the capacity of the system by three or more steps, multiple indoor fan coil units, each of which is individually metered and individually controlled by a proprietary control device and common communications network. This system is capable of operating as an air-conditioner or as a heat pump. The system is also capable of providing simultaneous heating and cooling operation, where recovered energy from the indoor units operating in one mode can be transferred to one or more other indoor units operating in the other mode. Variable refrigerant flow implies 3 or more steps of control on common, inter-connecting piping.

NOTE: This may be achieved by a gas/liquid separator or a third line in the refrigeration circuit.

3.29 *Water-To-Air Heat Pump and/or Brine-to-Air Heat Pump.* A heat pump which consists of one or more factory-made assemblies which normally include an indoor conditioning coil with air-moving means, compressor(s), and refrigerant-to-water or refrigerant-to-brine heat exchanger(s), including means to provide both cooling and heating, cooling-only, or heating-only functions. When such equipment is provided in more than one assembly, the separated assemblies should be designed to be used together. Such equipment may also provide functions of sanitary water heating, air cleaning, dehumidifying, and humidifying.

3.30 *Water Loop Heat Pump.* Water-to-air heat pump using liquid circulating in a common piping loop functioning as a heat source/heat sink. The temperature of the liquid loop is usually mechanically controlled within a temperature range of 59°F [15°C] to 104°F [40.0°C].

Section 4. Classifications

Equipment covered within the scope of this standard shall be classified as shown in Table 1.

Table 1. Classification of VRF Multi-Split Systems

Attribute \ System Identification		VRF Multi-Split Air Conditioner or Heat Pump	VRF Heat Recovery Multi-Split
Refrigerant Circuits		One shared with all indoor units	One shared with all indoor units
Compressors		One or more variable speed or alternative method resulting in three or more steps of capacity.	One or more variable speed or alternative method resulting in three or more steps of capacity.
Indoor Units	Qty.	Greater than one indoor unit	
	Operation	Individual Zones/Temp	Individual Zones/Temp
Outdoor Unit(s)	Qty.	One or multiple-manifolded outdoor units with a specific model number.	One or multiple-manifolded outdoor units with a specific model number.
	Steps of Control	Three or More	Three or More
	Mode of Operation	A/C, H/P	A/C, H/P, H/R
	Heat Exchanger	One or more circuits of shared refrigerant flow	One or more circuits of shared refrigerant flow
Classification	Air-Conditioner (air-to-air)	MSV-A-CB	
	Air-Conditioner (water-to-air)	MSV-W-CB	
	Heat Pump (air-to-air)	HMSV-A-CB	HMSR-A-CB
	Heat Pump (water-to-air)	HMSV-W-CB	HMSR-W-CB

NOTES:
[1] A suffix of "-O" following any of the above classifications indicates equipment not intended for use with field-installed duct systems (6.1.5.1.2).
[2] A suffix of "-A" indicates air-cooled condenser and "-W" indicates water-cooled condenser.
[3] For the purposes of the tested combination definition, when two or more outdoor units are connected, they will be considered as one outdoor unit.

Section 5. Test Requirements

5.1 All Standard Ratings shall be verified by tests conducted in accordance with the test methods and procedures as described in this standard and its appendices.

5.1.1 Air-cooled, water-cooled and evaporative-cooled units shall be tested in accordance with ANSI/ASHRAE Standard 37 and with Appendices C and D.

5.1.2 To set up equipment for test which incorporates inverter-controlled compressors, skilled personnel with knowledge of the control software will be required

5.1.3 If the equipment cannot be maintained at steady state conditions by its normal controls, then the manufacturer shall modify or over-ride such controls so that steady state conditions are achieved.

5.1.4 If a manufacturer indicates that its system is designed to recover oil more frequently than every two hours of continuous operation, the oil recovery mode shall be activated during testing. In all other cases, this mode should be disabled during testing.

5.2 *Number of Tests to be Conducted*.

5.2.1 Multi-split manufacturers must test two or more combinations of indoor units with each outdoor unit:

5.2.1.1 The first system combination shall be tested using only non-ducted indoor units that meet the definition of a Tested Combination. The rating given to any untested multi-split system combination having the same outdoor unit and all non-ducted indoor units shall be set equal to the rating of the tested system having all non-ducted indoor units.

5.2.1.2 The second system combination shall be tested using only ducted indoor units that meet the definition of a Tested Combination. The rating given to any untested multi-split system combination having the same outdoor unit and all ducted indoor units shall be set equal to the rating of the tested system having all ducted indoor units. In order to be considered a ducted unit, the indoor unit must be intended to be connected with ductwork and have a rated external static pressure capability greater than zero (0).

5.2.2 The rating given to any untested multi-split system combination having the same outdoor unit and a mix of non-ducted and ducted indoor units shall be set equal to the average of the ratings for the two required tested combinations.

Section 6. Rating Requirements

6.1 *Standard Ratings.* Standard Ratings shall be established at the Standard Rating Conditions specified in 6.1.3.

Air-cooled Multi-Split Air Conditioner and Heat Pumps <65,000 Btu/h [19,000W] shall be rated at conditions specified in section 6.2, in Tables 4, 5, and 6.

Air-cooled Multi-Split Air Conditioners and Heat Pumps and evaporatively and water-cooled air-conditioning-only systems ≥65,000 Btu/h shall be rated at conditions specified in 6.3 and Table 8.

Multi-Split Heat Pump that use a water-source for heat rejection shall be rated at conditions specified in Section 6.4 and Tables 9 and 10.

If a non-ducted or ducted indoor unit contains an integral condensate pump, the power to operate the pump shall be included in the system total power calculation.

Standard Ratings relating to cooling or heating capacities shall be net values, including the effects of circulating-fan heat, but not including supplementary heat. Power input shall be the sum of power input to the compressor(s) and fan(s), plus controls and other items required as part of the system for normal operation.

Standard Ratings of water-cooled units from 65,000 to below 300,000 Btu/h [19,000 to 88,000 W] shall include a total allowance for cooling tower fan motor and circulating water pump motor power inputs to be added in the amount of 10.0 W per 1000 Btu/h [34.1 W per 1000 W] cooling capacity.

6.1.1 *Values of Standard Capacity Ratings.* These ratings shall be expressed only in terms of Btu/h [W] as shown:

Table 2. Values of Standard Capacity Ratings

Capacity Ratings, Btu/h [W]	Multiples, Btu/h [W]
< 20,000 [5,900]	100 [30]
≥ 20,000 and < 38,000 [5,900 up to 11,000]	200 [60]
≥ 38,000 and < 65,000 [11,000 up to 19,000]	500 [150]
≥65,000 and < 135,000 [19,000 up to 39,600]	1000 [300]
≥ 136,000 and < 300,000 [39,800 up to 88,000]	2000 [600]

6.1.2 *Values of Energy Efficiency*

6.1.2.1 *For Systems < 65,000 Btu/h [19,000W]; Values of Measures of Energy Efficiency.* Standard measures of energy EFFICIEN1.1CY, whenever published, shall be expressed in multiples of the nearest 0.05 Btu/(W·h) for EER, SEER and HSPF.

6.1.2.2 *For Systems ≥ 65,000 Btu/h [19,000W]; Values of Measures of Energy Efficiency.* Energy Efficiency Ratios (EER), and Integrated Energy Efficiency Ratios (IEER) [Integrated Part-Load Values (IPLV)] for cooling, whenever published shall be expressed in multiples of the nearest 0.1 Btu/W·h [0.03 W/W]. Coefficients of Performance (COP) shall be expressed in multiples of the nearest 0.01.

6.1.3 *Standard Rating Tests.* Tables 4 - 10 indicate the test and test conditions which are required to determine values of standard capacity ratings and measures of energy efficiency.

6.1.3.1 *For Systems < 65,000 Btu/h [19,000W]; Assigned Degradation Factor.* In lieu of conducting the heating or cooling cycling test, an assigned value of 0.25 may be used for either the cooling or heating Degradation Coefficient, C_D, or both.

6.1.3.2 *Electrical Conditions.* Standard rating tests shall be performed at the nameplate rated frequency. For equipment which is rated with 208/230 V dual nameplate voltages, standard rating tests shall be performed at 230 V. For all other dual nameplate voltage equipment covered by this standard, the standard rating tests shall be performed at both voltages or at the lower of the two voltages if only a single Standard Rating is to be published.

6.1.4 *Control of System and Indoor Units.* The manufacturer must provide a schematic and sequence of operation for providing control of the system during testing.

6.1.5 Airflow *Requirements for Systems with Capacities <65,000 Btu/h [19,000 W].* Air volume rate is equivalent to air flow rates, volumetric air flow rate and may be used interchangeably.

6.1.5.1 *Cooling Full-Load Air Volume Rate.*

6.1.5.1.1. *Cooling Full-Load Air Volume Rate for Ducted Units.* The manufacturer must specify the cooling air volume rate. Use this value as long as the following two requirements are satisfied. First, when conducting the A_2 test (exclusively), the measured air volume rate, when divided by the measured indoor air-side total cooling capacity, must not exceed 37.5 scfm per 1,000 Btu/h [0.06 m^3/s per 1,000 W]. If this ratio is exceeded, reduce the air volume rate until this ratio is equaled. Use this reduced air volume rate for all tests that call for using the Cooling Full-load Air Volume Rate. The second requirement is as follows:

a. For all ducted units tested with an indoor fan installed, except those having a variable-speed, constant-air-volume-rate indoor fan. The second requirement applies exclusively to the A_2 test and is met as follows.

1. Achieve the cooling full-load air volume rate, determined in accordance with the previous paragraph;

2. Measure the external static pressure;

3. If this pressure is equal to or greater than the applicable minimum external static pressure cited in Table 7, this second requirement is satisfied. Use the current air volume rate for all tests that require the Cooling Full-load Air Volume Rate.

4. If the Table 7 minimum is not equaled or exceeded,

   4a. reduce the air volume rate until the applicable Table 7 minimum is equaled, or

   4b. until the measured air volume rate equals 95 percent of the air volume rate from step 1, whichever occurs first.

5. If the conditions of step 4a occur first, this second requirement is satisfied. Use the step 4a reduced air volume rate for all tests that require the cooling full-load air volume rate.

6. If the conditions of step 4b occur first, make an incremental change to the set-up of the indoor fan (e.g., next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning at above step 1. If the indoor fan set-up cannot be further changed, reduce the air volume rate until the applicable Table 7 minimum is equaled. Use this reduced air volume rate for all tests that require the cooling full-load air volume rate.

b. For ducted units that are tested with a variable-speed, constant-air-volume-rate indoor fan installed. For all tests that specify the cooling full-load air volume rate, obtain an external static pressure as close to (but not less than) the applicable Table 7 value that does not cause instability or an automatic shutdown of the indoor blower.

6.1.5.1.2 *Cooling Full-load Air Volume Rate for Non-ducted Units*. For non-ducted units, the Cooling Full-load Air Volume Rate is the air volume rate that results during each test when the unit is operated at an external static pressure of zero in H_2O [zero Pa].

6.1.5.2 *Cooling Minimum Air Volume Rate.*

a. For ducted units that regulate the speed (as opposed to the cfm) of the indoor fan,

$$\text{Cooling Minimum Air Vol. Rate} = \text{Cooling Full-load Air Vol. Rate} \times \frac{\text{Cooling Minimum Fan Speed}}{\text{A}_2 \text{ Test Fan Speed}} \tag{l}$$

where "cooling minimum fan speed" corresponds to the fan speed used when operating at the minimum compressor speed. For such systems, obtain the Cooling Minimum Air Volume Rate regardless of the external static pressure.

b. For ducted units that regulate the air volume rate provided by the indoor fan, the manufacturer must specify the cooling minimum air volume rate. For such systems, conduct all tests that specify the cooling minimum air volume rate — (i.e., the B_1, F_1, and G_1 tests) — at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$B_1,\ F_1,\text{ and }C_1\text{ Test }\Delta P_{st} = \Delta P_{st,A_2} \times \left[\frac{\text{Cooling Minimum Air Volume Rate}}{\text{Cooling Full-load Air Volume Rate}}\right]^2 \qquad (2)$$

where $\Delta P_{st,A_2}$ is the applicable Table 7 minimum external static pressure that was targeted during the A_2 (and B_2) test.

c. For non-ducted units, the Cooling Minimum Air Volume Rate is the air volume rate that results during each test when the unit operates at an external static pressure of zero in H_2O [zero Pa] and at the indoor fan setting used at minimum compressor speed.

6.1.5.3 *Cooling Intermediate Air Volume Rate.*

a. For ducted units that regulate the speed of the indoor fan,

$$\text{Cooling Intermediate Air Volume Rate} = \text{Cooling Full-load Air Volume Rate} \times \frac{E_V\text{ Test Fan Speed}}{A_2\text{ Test Fan Speed}} \qquad (3)$$

For such units, obtain the Cooling Intermediate Air Volume Rate regardless of the external static pressure.

b. For ducted units that regulate the air volume rate provided by the indoor fan, the manufacturer must specify the cooling intermediate air volume rate. For such systems, conduct the E_V test at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$E_V\text{ Test }\Delta P_{st} = \Delta P_{st,A_2} \times \left[\frac{\text{Cooling Intermediate Air Volume Rate}}{\text{Cooling Full-load Air Volume Rate}}\right]^2 \qquad (4)$$

where $\Delta P_{st,A_2}$ is the applicable Table 7 minimum external static pressure that was targeted during the A_2 (and B_2) test.

c. For non-ducted units, the Cooling Intermediate Air Volume Rate is the air volume rate that results when the unit operates at an external static pressure of zero in H_2O [zero Pa] and at the fan speed selected by the controls of the unit for the E_V test conditions.

6.1.5.4 *Heating Full-load Air Volume Rate.*

6.1.5.4.1 *Ducted Heat Pumps where the Heating and Cooling Full-load Air Volume Rates are the Same.*

a. Use the Cooling Full-load Air Volume Rate as the Heating Full-load Air Volume Rate for:

1. Ducted heat pumps that operate at the same indoor fan speed during both the A_2 and the $H1_2$ tests;

2. Ducted heat pumps that regulate fan speed to deliver the same constant air volume rate during both the A_2 and the $H1_2$ tests; and

3. The airflow of all of the individual ducted indoor units must be added together to arrive at the full-load air volume rate

b. For heat pumps that meet the above criteria "1" and "3," no minimum requirements apply to the measured external static pressure. For heat pumps that meet the above criterion "2," test at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than, the same Table 7 minimum external static pressure as was specified for the A_2 cooling mode test.

6.1.5.4.2 *Ducted Heat Pumps where the Heating and Cooling Full-load Air Volume Rates are Different due to Indoor Fan Operation.*

a. For ducted heat pumps that regulate the speed (as opposed to the cfm) of the indoor fan,

$$\text{Heating Full-load Air Volume Rate} = \text{Cooling Full-load Air Volume Rate} \times \frac{\text{H1 or H1}_2\text{ Test Fan Speed}}{\text{A or A}_2\text{ Test Fan Speed}} \quad (5)$$

For such heat pumps, obtain the Heating Full-load Air Volume Rate without regard to the external static pressure.

b. For ducted heat pumps that regulate the air volume rate delivered by the indoor fan, the manufacturer must specify the Heating Full-load Air Volume Rate. For such heat pumps, conduct all tests that specify the Heating Full-load Air Volume Rate at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$\text{Heating Full-load } \Delta P_{st} = \text{Cooling Full Load } \Delta P_{st} \times \left[\frac{\text{Heating Air Volume Rate}}{\text{Cooling Air Volume Rate}}\right]^2 \quad (6)$$

where the cooling ΔP_{st}, $H1_2$ is the applicable Table 7 minimum external static pressure that was specified for the A_2 test.

6.1.5.4.3 *Non-ducted Heat Pumps, Including Non-ducted Heating-only Heat Pumps*. For non-ducted heat pumps, the Heating Full-load Air Volume Rate is the air volume rate that results during each test when the unit operates at an external static pressure of zero in H_2O [zero Pa].

6.1.5.4.4 *Heating Minimum Air Volume Rate.*

a. For ducted heat pumps that regulate the speed (as opposed to the cfm) of the indoor fan,

$$\text{Heating Minimum Air Volume Rate} = \text{Heating Full-load Air Volume Rate} \times \left[\frac{\text{Heating Minimum Fan Speed}}{\text{H1}_2\text{ Test Fan Speed}}\right] \quad (7)$$

where "heating minimum fan speed" corresponds to the lowest fan speed used at any time when operating at the minimum compressor speed (variable-speed system). For such heat pumps, obtain the Heating Minimum Air Volume Rate without regard to the external static pressure.

b. For ducted heat pumps that regulate the air volume rate delivered by the indoor fan, the manufacturer must specify the Heating Minimum Air Volume Rate. For such heat pumps, conduct all tests that specify the heating minimum air volume rate (i.e., the $H0_1$, $H0C_1$, and $H1_1$ tests) — at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$H0_1, H1_1, H2_1, \; H0C_1 \; \Delta P_{st,H1_2} \times \left[\frac{\text{Heating Minimum Air Volume Rate}}{\text{Heating Full} - \text{load Air Volume Rate}}\right]^2 \tag{8}$$

where $\Delta P_{st,H1_2}$ is the minimum external static pressure that was targeted during the $H1_2$ test.

c. For non-ducted heat pumps, the Heating Minimum Air Volume Rate is the air volume rate that results during each test when the unit operates at an external static pressure of zero in H_2O [zero Pa] and at the indoor fan setting used at minimum compressor speed.

6.1.5.4.5 *Heating Intermediate Air Volume Rate.*

a. For ducted heat pumps that regulate the speed of the indoor fan,

$$\text{Heating Intermediate Air Volume Rate} = \text{Heating } \mathit{Full-load} \text{ Air Volume Rate} \times \frac{H2_V \text{ Test Fan Speed}}{H1_2 \text{ Test Fan Speed}} \tag{9}$$

For such heat pumps, obtain the Heating Intermediate Air Volume Rate without regard to the external static pressure.

b. For ducted heat pumps that regulate the air volume rate delivered by the indoor fan, the manufacturer must specify the Heating Intermediate Air Volume Rate. For such heat pumps, conduct the $H2_V$ test at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$H2_V \text{ Test } \Delta P_{st,\,H1_2} = \left[\frac{\text{Heating Intermediate Air Volume Rate}}{\text{Heating } \mathit{Full-load} \text{ Air Volume Rate}}\right]^2 \tag{10}$$

where $\Delta P_{st,\,H1_2}$ is the minimum external static pressure that was specified for the $H1_2$ test.

c. For non-ducted heat pumps, the Heating Intermediate Air Volume Rate is the air volume rate that results when the heat pump operates at an external static pressure of zero in H_2O [zero Pa] and at the fan speed selected by the controls of the unit for the $H2_V$ test conditions.

6.1.5.4.6 *Heating Nominal Air Volume Rate*. Except for the noted changes, determine the Heating Nominal Air Volume Rate using the approach described in section 6.1.5.4.5. Required changes include substituting "$H1_N$ test" for "$H2_V$ test" within the first section 6.1.5.4.5 equation, substituting "$H1_N$ test ΔP_{st}" for "$H2_V$ test ΔP_{st}" in the second section 6.1.5.4.5 equation, substituting "$H1_N$ test" for each "$H2_V$ test", and substituting "Heating Nominal Air Volume Rate" for each "Heating Intermediate Air Volume Rate."

$$\text{Heating Intermediate Air Volume Rate} = \text{Heating } \mathit{Full-load} \text{ Air Volume Rate} \times \frac{H2_V \text{ Test Fan Speed}}{H1_2 \text{ Test Fan Speed}} \tag{11}$$

$$\text{H1}_{\text{N}}\ \text{Test}\ \Delta P_{st} = \Delta P_{st,\text{H1}_2} \left[\frac{\text{Heating Nominal Air Volume Rate}}{\text{Heating } \textit{Full}-\textit{load} \text{ Air Volume Rate}} \right]^2 \tag{12}$$

6.1.6 *Outdoor-Coil Airflow Rate (Applies to all Air-to-Air Systems).* All Standard Ratings shall be determined at the outdoor-coil airflow rate specified by the manufacturer where the fan drive is adjustable. Where the fan drive is non-adjustable, ratings shall be determined at the outdoor-coil airflow rate inherent in the equipment when operated with all of the resistance elements associated with inlets, louvers, and any ductwork and attachments considered by the manufacturer as normal installation practice. Once established, the outdoor coil air circuit of the equipment shall remain unchanged throughout all tests prescribed herein.

6.1.7 *Requirements for Separated Assemblies (Applies to all Systems).* All standard ratings for equipment in which the condenser and the evaporator are two separate assemblies, as in Types: MSV-A-CB, MSV-W-CB, HMSV-A-CB, HMSV-W-CB, HMSR-A-CB, (See Table 1 Notes) and HMSR-W-CB, shall be obtained with a minimum 25 ft. [7.6 m] of interconnecting tubing length (for one indoor unit with additional length requirements for each additional unit). Refer to Table 3 for minimum total refrigerant tube lengths. The complete length of tubing furnished as an integral part of the unit (and not recommended for cutting to length) shall be used in the test procedure, or with 25 ft [7.6 m] of refrigerant path, whichever is greater. At least 10 ft [3.0 m] of the system interconnection tubing shall be exposed to the outside conditions. The line diameters, insulation, installation details, evacuation and charging shall follow the manufacturer's published recommendations. The manufacturer will provide a schematic of the tested combination installation (See Figure 1).

Table 3. Piping Requirements for Tested Combinations

(Piping length from outdoor unit to each indoor unit)

System Capacity	Systems with Non-ducted Indoor Units	Systems with Ducted Indoor Units
0 to <65,000 Btu (0 to <10,950 W)	25' (7.6 m)	25' (7.6 m)
≥65,000 Btu to <105,000 Btu (≥10,950 W to <30,800 W)	50' (15.5 m)	25' (7.6 m)
≥106,000 Btu to <134,000 Btu (≥31,100 W to <39,300 W)	75' (23 m)	25' (7.6 m)
≥135,000 Btu to <350,000 Btu (≥40,000 W to <102,550 W)	100' (30.5 m)	50' (15.5 m)
>350,000 Btu (>102,550 W)	150' (75.2 m)	75' (23 m)

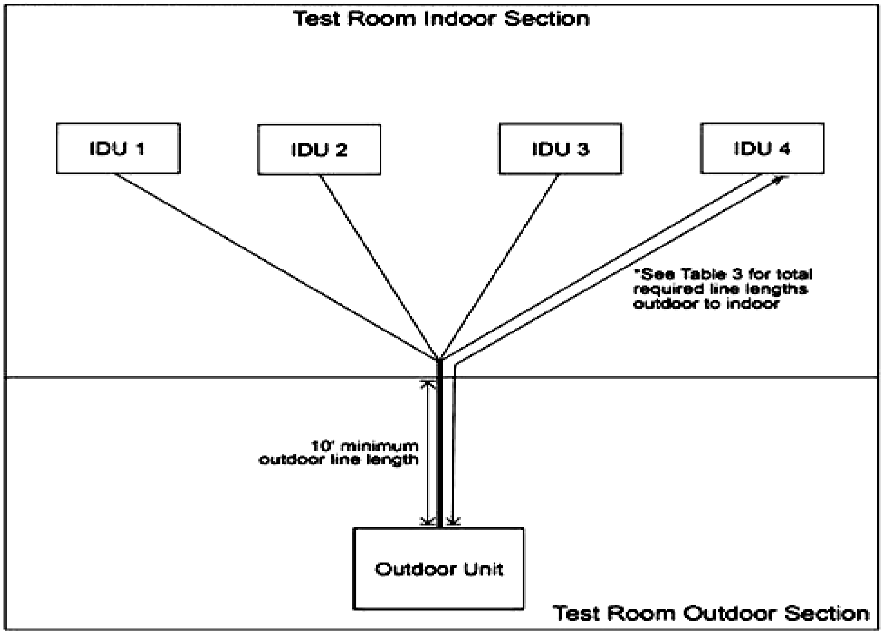

Figure 1. Test Room Layout

6.2 *Conditions for Standard Rating Test for Air-cooled Systems < 65,000 Btu/h [19,000W]*

6.2.1 *Instructions for Multiple Indoor Unit Testing*

a. At least one indoor unit must be turned off for tests conducted at minimum compressor speed. In addition, the manufacturer may elect to have one or more indoor units turned off for tests conducted at the intermediate compressor speed. In all cases, the manufacturer specifies the particular indoor unit(s) that is turned off.

6.2.2 *Compressor Speed.* The speed at which the compressor runs to deliver the capacity of the tested combination.

6.2.2.1 *Maximum Compressor Speed.* Manufacturers shall designate the maximum compressor speed. The maximum compressor speed for cooling mode tests is a fixed value. The maximum compressor speed for heating mode tests is also a fixed value that may be the same or different from the cooling mode value.

6.2.2.2 *Intermediate Compressor Speed.* For each test manufactures will designate the intermediate compressor speed that falls within ¼ and ¾ of the difference between the minimum and maximum speeds for both cooling and heating.

6.2.2.3 *Minimum Compressor Speed.* Manufacturers shall designate the minimum compressor speed at a steady-state level below which the system would rarely operate. The minimum compressor speed for cooling mode tests is a fixed value. The minimum compressor speed for heating mode tests is also a fixed value that may be the same or different from the cooling mode value.

6.2.3 *Cooling Tests for a Unit Having a Variable-speed Compressor.*

a. Conduct five steady-state wet coil tests: the A_2, E_V, B_2, B_1, and F_1 tests. Use the two optional dry-coil tests, the steady-state G_1 test and the cyclic I_1 test, to determine the cooling mode cyclic degradation coefficient, C_D^c. If the two optional tests are not conducted, assign C_D^c the default value of 0.25. Table 4 specifies test conditions for these seven tests.

Table 4. Cooling Mode Test Conditions for Units < 65,000 Btu/h [19,000 W]

Test Description	Air Entering Indoor Unit Temperature		Air Entering Outdoor Unit Temperature		Compressor Speed	Cooling Air Volume Rate
	Dry-Bulb °F [°C]	Wet-Bulb °F [°C]	Dry-Bulb °F [°C]	Wet-Bulb °F [°C]		
A_2 Test - required (steady, wet coil)	80.0 [26.7]	67.0 [19.4]	95.0 [35.0]	75.0(1) [23.9(1)]	Maximum7	Cooling Full-load Air Volume Rate(2)
B_2 Test - required (steady, wet coil)	80.0 [26.7]	67.0 [19.4]	82.0 [27.8]	65.0(1) [18.3(1)]	Maximum7	Cooling Full-load Air Volume Rate(2)
E_V Test - required (steady, wet coil)	80.0 [26.7]	67.0 [19.4]	87.0 [30.6]	69.0(1) [20.6(1)]	Intermediate8	Cooling Intermediate(3)
B_1 Test - required (steady, wet coil)	80.0 [26.7]	67.0 [19.4]	82.0 [27.8]	65.0(1) [18.3(1)]	Minimum9	Cooling Minimum(4)
F_1 Test - required (steady, wet coil)	80.0 [26.7]	67.0 [19.4]	67.0 [19.4]	53.5(1) [11.9(1)]	Minimum9	Cooling Minimum(4)
G_1 Test(5) - optional (steady, dry coil)	80.0 [26.7]	(6)	67.0 [19.4]	NA	Minimum9	Cooling Minimum(4)
I_1 Test(5) - optional (cyclic, dry coil)	80.0 [26.7]	(6)	67.0 [19.4]	NA	Minimum9	(6)

NOTES:
(1) The specified test condition only applies if the unit rejects condensate to the outdoor coil.
(2) Defined in section 6.1.5.1
(3) Defined in section 6.1.5.3
(4) Defined in section 6.1.5.2
(5) The entering air must have a low enough moisture content so no condensate forms on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57.0 °F [13.9 °C] or less be used.)
(6) Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the G_1 Test.
(7) Maximum compressor speed is defined in paragraph 6.2.2.1
(8) Intermediate compressor speed is defined in paragraph 6.2.2.2
(9) Minimum compressor speed is defined in paragraph 6.2.2.3

6.2.4 *Heating Mode Tests for a Heat Pump Having a Variable-speed Compressor.*

a. Conduct one maximum temperature test ($H0_1$), two high temperature tests ($H1_2$ and $H1_1$), one frost accumulation test ($H2_V$), and one low temperature test ($H3_2$). Conducting one or both of the following tests is optional: an additional high temperature test ($H1_N$) and an additional frost accumulation test ($H2_2$). Conduct the optional maximum temperature cyclic ($H0C_1$) test to determine the heating mode cyclic degradation coefficient, C_D^h. If this optional test is not conducted, assign C_D^h the default value of 0.25. Table 5 specifies test conditions for these eight tests.

Table 5. Heating Mode Test Conditions for Units < 65,000 Btu/h [19,000 W]

Test Description	Air Entering Indoor Unit Temperature		Air Entering Outdoor Unit Temperature		Compressor Speed	Heating Air Volume Rate
	Dry-Bulb °F [°C]	Wet-Bulb (max) °F [°C]	Dry-Bulb °F [°C]	Wet-Bulb °F [°C]		
$H0_1$ Test (required, steady)	70.0 [21.1]	60.0 [15.6]	62.0 [16.7]	56.5 [13.6]	Minimum[6]	Heating Minimum[(1)]
$H0C_1$ Test (optional, cyclic)	70.0 [21.1]	60.0 [15.6]	62.0 [16.7]	56.5 [13.6]	Minimum[6]	[(2)]
$H1_2$ Test (required, steady)	70.0 [21.1]	60.0 [15.6]	47.0 [8.3]	43.0 [6.1]	Maximum[8]	Heating Full-load Air Volume Rate[(3)]
$H1_1$ Test (required, steady)	70.0 [21.1]	60.0 [15.6]	47.0 [8.3]	43.0 [6.1]	Minimum[6]	Heating Minimum[(1)]
$H1_N$ Test (optional, steady)	70.0 [21.1]	60.0 [15.6]	47.0 [8.3]	43.0 [6.1]	Cooling Mode Maximum[7]	Heating Nominal[(4)]
$H2_2$ Test (optional)	70.0 [21.1]	60.0 [15.6]	35.0 [1.7]	33.0 [0.6]	Maximum[8]	Heating Full-load Air Volume Rate[(3)]
$H2_V$ Test (required)	70.0 [21.1]	60.0 [15.6]	35.0 [1.7]	33.0 [0.6]	Intermediate[7]	Heating Intermediate[(5)]
$H3_2$ Test (required, steady)	70.0 [21.1]	60.0 [15.6]	17.0 [-8.3]	15.0 [-9.4]	Maximum[8]	Heating Full-load Air Volume Rate[(3)]

NOTES:

(1) Defined in section 6.1.5.4.4

(2) Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the $H0_1$ Test.

(3) Defined in section 6.1.5.4.

(4) Defined in section 6.1.5.4.6.

(5) Defined in section 6.1.5.4.5.

(6) Minimum compressor speed is defined in paragraph 6.2.2.1

(7) Intermediate compressor speed is defined in paragraph 6.2.2.3

(8) Maximum compressor speed is defined in paragraph 6.2.2.3

Table 6. Conditions for Operating Requirement Tests for Air-Cooled Equipment < 65,000 Btu/h [19,000 W]					
TEST		INDOOR UNIT		OUTDOOR UNIT	
		Air Entering Temperature		Air Entering Temperature	
		Dry-Bulb °F [°C]	Wet-Bulb °F [°C]	Dry-Bulb °F [°C]	Wet-Bulb °F [°C]
COOLING	Voltage Tolerance	80.0 [26.7]	67.0 [19.4]	95.0 [35.0]	75.0 (1) [23.9]
	Low Temperature Operation Cooling	67.0 [19.4]	57.0 [13.9]	67.0 [19.4]	57.0 (1) [13.9]
	Insulation Efficiency	80.0 [26.7]	75.0 [23.9]	80.0 [26.7]	75.0 (1) [23.9]
	Condensate Disposal	80.0 [26.7]	75.0 [23.9]	80.0 [26.7]	75.0 (1) [23.9]
	Maximum Operating Conditions	80.0 [26.7]	67.0 [19.4]	115.0 [46.1]	75.0 (1) [23.9]
HEATING	Voltage Tolerance (Heating-only units)	70.0 [21.1]	60.0 [15.6] (max)	47.0 [8.3]	43.0 [6.1]
	Maximum Operating Conditions	80.0 [26.7]	NA NA	75.0 [23.9]	65.0 [18.3]

NOTE:

(1) The wet-bulb temperature condition is not required when testing air-cooled condensers which do not evaporate condensate.

Table 7. Minimum External Static Pressure for Ducted Systems Tested with External Static Pressure > 0 [in H2O]			
Rated Cooling(1) or Heating(2) Capacity		Minimum External Resistance(3)	
Btu/h	kW	in H_2O	Pa
Up through 28,800(4)	6.40 to 8.44	0.10	25
29,000 to 42,500	8.5 to 12.4	0.15	37
43,000 thru 60,000	12.6 thru 19.0	0.20	50

NOTES:

(1) For air conditioners and heat pumps, the value cited by the manufacturer in published literature for the unit's capacity when operated at the A_2 Test conditions.

(2) For heating-only heat pumps, the value the manufacturer cites in published literature for the unit's capacity when operated at the $H1_2$ Test conditions.

(3) For ducted units tested without an air filter installed, increase the applicable tabular value by 0.08 in H_2O [20 Pa].

(4) If the manufacturer's rated external static pressure is less than 0.10 in H_2O (25 Pa), then the indoor unit should be tested at that rated external static pressure. (See 5.2.1.2)

6.3 *Conditions for Standard Rating Test for Air-cooled Air Conditioner and Heat Pump Systems and Water-cooled Air Conditioning Systems ≥ 65,000 Btu/h [19,000W]*

6.3.1 *Indoor-Coil Airflow Rate.* All Standard Ratings shall be determined at an indoor-coil airflow rate as outlined below. All airflow rates shall be expressed in terms of Standard Air.

a. Equipment with indoor fans intended for use with field installed duct systems shall be rated at the manufacturer specified airflow rate (not to exceed 37.5 SCFM per 1000 Btu/h [0.06 m^3/s per 1000 W] of rated capacity) while meeting or exceeding the minimum external resistance specified in Table 5.

b. Equipment with indoor fans not intended for use with field installed duct systems (free discharge) shall be rated at the indoor-side air quantity delivered when operating at zero in H_2O [zero Pa] external pressure.

c. 100% recirculated air shall be used.

d. Equipment which does not incorporate an indoor fan is not covered in this standard.

e. Indoor-coil airflow rates and pressures as referred to herein apply to the airflow rate experienced when the unit is cooling and dehumidifying under the conditions specified in this section. This airflow rate, except as noted in 6.3.1b and 8.8 shall be employed in all other tests prescribed herein without regard to resultant external static pressure.

6.3.2 *External Resistances.* Commercial and Industrial Unitary Air-Conditioners and Heat Pumps shall be tested at the minimum external resistances in Table 7 when delivering the rated capacity and airflow rate specified in 6.3.1.

Indoor air-moving equipment not intended for use with field installed duct systems (free discharge) shall be tested at zero in H_20 [zero Pa] external pressure.

6.3.3 *Rating Conditions for Air Conditioning Equipment with Optional Outdoor Air Cooling Coil.* Commercial and Industrial Unitary Air Conditioners which incorporate an outdoor air cooling coil shall use the Standard Rating Conditions (Table 8) for rating except for the following changes:

a. Unit shall be adjusted to take in 20% outdoor air at conditions specified in Table 8.

b. Return air temperature conditions shall be 80.0ºF [27.0ºC] dry-bulb, 67.0ºF [19.0ºC] wet-bulb.

6.3.4 *Outdoor-Coil Airflow Rate (Applies to All Air-to-air Systems).* All Standard Ratings shall be determined at the outdoor-coil airflow rate specified by the manufacturer where the fan drive is adjustable. Where the fan drive is non-adjustable, they shall be determined at the outdoor-coil airflow rate inherent in the equipment when operated with all of the resistance elements associated with inlets, louvers, and any ductwork and attachments considered by the manufacturer as normal installation practice. Once established, the outdoor-side air circuit of the equipment shall remain unchanged throughout all tests prescribed herein unless automatic adjustment of outdoor airflow rates by system function is made.

Table 8. Operating Conditions for Standard Rating and Performance Operating Tests for Systems ≥ *65,000 Btu/h* [19,000 W]

	TEST	INDOOR SECTION		OUTDOOR SECTION					
		Air Entering		Air Entering				Water[5]	
				Air Cooled		Evaporative			
		Dry-Bulb °F [°C]	Wet-Bulb °F [°C]	Dry-Bulb °F [°C]	Wet-Bulb °F [°C]	Dry-Bulb °F [°C]	Wet-Bulb °F [°C]	IN °F [°C]	OUT °F [°C]
COOLING	Standard Rating Conditions Cooling[3]	80.0 [26.7]	67.0 [19.4]	95.0 [35.0]	75.0[1] [23.9]	95.0 [35.0]	75.0 [23.9]	85.0 [29.4]	95.0 [35.0]
COOLING	Low Temperature Operating Cooling[3]	67.0 [19.4]	57.0 [13.9]	67.0 [19.4]	57.0[1] [13.9]	67.0 [19.4]	57.0 [13.9]	NA	70.0[2] [21.1]
COOLING	Maximum Operating Conditions[3]	80.0 [26.7]	67.0 [19.4]	115 [46.1]	75.0[1] [23.9]	100 [37.8]	80.0[4] [26.7]	90.0[2] [32.2]	NA
COOLING	Part-Load Conditions (IEER)[3]	80.0 [26.7]	67.0 [19.4]	Varies with load per Table 11	[1]Varies with load per Table 11	Varies with load per Table 11	Varies with load per Table 11	[2]Varies with load per Table 11	Varies with load per Table 11
COOLING	Part-Load Conditions (IPLV)[3]	80.0 [26.7]	67.0 [19.4]	80.0 [26.7]	67.0[1] [19.4]	80.0 [26.7]	67.0 [26.7]	75.0[2] [23.9]	NA
COOLING	Insulation Efficiency[3]	80.0 [26.7]	75.0 [23.9]	80.0 [26.7]	75.0[1] [23.9]	80.0 [26.7]	75.0 [23.9]	NA	80.0 [26.7]
COOLING	Condensate Disposal[3]	80.0 [26.7]	75.0 [23.9]	80.0 [26.7]	75.0[1] [23.9]	80.0 [26.7]	75.0 [23.9]	NA	80.0 [26.7]
HEATING	Standard Rating Conditions (High Temperature Steady State Heating)	70.0 [21.1]	60.0 [15.6] (max)	47.0 [8.3]	43.0 [6.1]	NA	NA	NA	NA
HEATING	Standard Rating Conditions (Low Temperature Steady State Heating)	70.0 [21.1]	60.0 [15.6] (max)	17.0 [-8.3]	15.0 [-9.4]	NA	NA	NA	NA
HEATING	Maximum Operating Conditions	80.0 [26.7]	NA	75.0 [23.9]	65.0 [18.3]	NA	NA	NA	NA

NOTES:

1. The wet-bulb temperature condition is not required when testing air cooled condensers which do not evaporate condensate except for units with optional outdoor cooling coil.
2. Water flow rate as determined from Standard Rating Conditions Test.
3. Cooling rating and operating tests are not required for heating only heat pumps.
4. Make-up water temperature shall be 90.0ºF [32.0ºC].
5. The ratings for water-cooled outdoor sections in this table apply only to air conditioning-only systems.

6.4 *Conditions for Standard Rating Tests for Heat Pump Systems that use Water-source for Heat Rejection*

6.4.1 *Standard Ratings.* Standard ratings shall be established at the standard rating conditions specified in 6.4.8 and Tables 9 and 10. Standard ratings relating to cooling and heating capacities shall be net values, including the effects of circulating- fan heat, but not including supplementary heat. Standard efficiency ratings shall be based on the effective power input as defined in 3.6.

6.4.2 *Power Input of Liquid Pumps.*

6.4.2.1 If no liquid pump is provided with the heat pump, a pump power adjustment is to be included in the effective power consumed by the heat pump, using the following formula:

$$\varphi_{pa} = q \times \Delta p / \eta \tag{13}$$

where:

φ_{pa} = Pump power adjustment, in watts;
η = 1.59 (gpm)(ft H_2O)(1/W) [0.3 × 10^3 Liter/s*Pa*(1/W)] by convention;
Δp = Measured internal static pressure difference, (feet H_2O)[pascals];
q Nominal fluid flow rate, in gallons per minute [liters per second].

6.4.2.2 If a liquid pump is an integral part of the heat pump, only the portion of the pump power required to overcome the internal resistance shall be included in the effective power input to the heat pump. The fraction which is to be excluded from the total power consumed by the pump shall be calculated using the following formula:

$$\varphi_{pa} = q \times \Delta p / \eta \tag{14}$$

where:

φ_{pa} Pump power adjustment, in watts;
η = 1.59 (gpm)(ft H_2O)(1/W) [0.3 × 10^3] Liter/s*Pa*(1/W)] by convention; See note below.
Δp = The measured external static pressure difference, (feet H_2O)[pascals];
q = Nominal fluid flow rate, in gallons per minute [liters per second].

Note: 0.3 ×10^3 (L/s)(Pa)(1/W)
= 0.3 ×10^3 (L/s)(Pa)(1/W)(15.850323 gpm/ (L/s)) (.000334552 ft H_2O/ Pa)
= 1.59 (gpm)(ft H_2O)(1/W)

6.4.3 *Liquid Flow Rates*

6.4.3.1 All standard ratings shall be determined at a liquid flow rate described below, expressed as gallons per minute (liters per second).

6.4.4 Heat pumps with integral liquid pumps shall be tested at the liquid flow rates specified by the manufacturer or those obtained at zero external static pressure difference, whichever provides the lower liquid flow rate.

6.4.5 Heat pumps without integral liquid pumps shall be tested at the flow rates specified by the manufacturer.

6.4.6 The manufacturer shall specify a single liquid flow rate for all of the tests required in 6.4 unless automatic adjustment of the liquid flow rate is provided by the equipment. A separate control signal output for each step of liquid flow rate will be considered as an automatic adjustment.

6.4.7 *Test Liquids*

6.4.7.1 The test liquid for water-loop heat pumps and ground-water heat pumps shall be water.

6.4.7.2 The test liquid for ground-loop heat pumps shall be a 15% solution by mass of sodium chloride in water.

6.4.7.3 The test liquid shall be sufficiently free of gas to ensure that the measured result is not influenced by the presence of gas.

6.4.8 *Standard Rating and Part-load Rating Test Conditions*

6.4.8.1 The test conditions for the determination of standard and part-load cooling ratings are specified in Table 9.

6.4.8.2 The test conditions for determination of standard and part-load heating ratings are specified in Table 10.

6.4.8.3 Heat pumps intended for a specific application shall be rated at the conditions specified for that application, for example, water-loop, ground-water, or ground-loop, and shall be identified as such (i.e., water-loop heat pump, groundwater heat pump, or ground-loop heat pump). Heat pumps intended for two or three applications shall be rated at the conditions specified for each of these applications and shall be so identified (see 7.3 of ANSI/ARI/ASHRAE ISO Standard 13256-1:1998)

6.4.8.4 For each test, the equipment shall be operated continuously until equilibrium conditions are attained, but for not less than one hour before capacity test data are recorded. The data shall then be recorded for 30 minutes at 5-minutes intervals until seven consecutive sets of readings have been attained within the tolerances specified in 8.13.5. The averages of these data shall be used for the calculation of the test results.

Table 9. Test Conditions for The Determination of Cooling Capacity for Systems that use a Water Source for Heat Rejection

	Water-loop heat pumps	**Ground-water heat pumps**	**Ground-loop heat pumps**
Air entering indoor side — dry bulb, °F [°C] — wet bulb, °F [°C]	 80.6 [27.0] 66.2 [19.0]	 80.6 [27.0] 66.2 [19.0]	 80.6 [27.0] 66.2 [19.0]
Air surrounding unit — dry bulb, °F [°C]	 80.6 [27.0]	 80.6 [27.0]	 80.6 [27.0]
<u>Standard Rating Test</u> Liquid entering heat exchanger, °F [°C]	 86.0 [30.0]	 59.0 [15.0]	 77.0 [25.0]
<u>Part Load Rating Test</u> Liquid entering heat exchanger, °F [°C]	 86.0 [30.0]	 59.0 [15.0]	 68.0 [20.0]
Frequency*	Rated	Rated	Rated
Voltage**	Rated	Rated	Rated
*Equipment with dual-rated frequencies shall be tested at each frequency. **Equipment with dual-rated voltages shall be tested at both voltages, or at the lower if the two voltages if only a single rating is published.			

Table 10. Test Conditions for the Determination of Heating Capacity for Systems that use a Water Source for Heat Rejection

	Water-loop heat pumps	**Ground-water heat pumps**	**Ground-loop heat pumps**
Air entering indoor side* — dry bulb, °F [°C] — maximum wet bulb, °F [°C]	 68.0 [20.0] 59.0 [15.0]	 68.0 [20.0] 59.0 [15.0]	 68.0 [20.0] 59.0 [15.0]
Air surrounding unit — dry bulb, °F [°C]	 68.0 [20.0]	 68.0 [20.0]	 68.0 [20.0]
Standard Rating Test Liquid entering heat exchanger, °F [°C]	 68.0 [20.0]	 50.0 [10.0]	 32.0 [0]
Part Load Rating Test Liquid entering heat exchanger, °F [°C]	 68.0 [20.0]	 50.0 [10.0]	 41.0 [5.0]
Frequency*	Rated	Rated	Rated
Voltage**	Rated	Rated	Rated
*Equipment with dual-rated frequencies shall be tested at each frequency. **Equipment with dual-rated voltages shall be tested at both voltages, or at the lower if the two voltages if only a single rating is published.			

6.5 *Part-Load Rating.* Integrated Part-Load Value (IPLV) is in effect until January 1, 2010. See Appendix H for the method and calculation of IPLV. Effective January 1, 2010, all units ≥ 65000 Btu/h [19,000W] rated in accordance with this standard shall include an Integrated Energy Efficiency Ratio (IEER).

6.5.1 *Part-load Rating Conditions.* Test conditions for part-load ratings shall be per Table 8. Any water flow required for system function shall be at water flow rates established at (full load) Standard Rating Conditions. Capacity reduction means may be adjusted to obtain the specified step of unloading. No manual adjustment of indoor and outdoor airflow rates from those of the Standard Rating Conditions shall be made. However, automatic adjustment of airflow rates by system function is permissible.

6.5.2 *General.* The IEER is intended to be a measure of merit for the part-load performance of the unit. Each building may have different part-load performance due to local occupancy schedules, building construction, building location and ventilation requirements. For specific building energy analysis an hour-by-hour analysis program should be used.

6.5.3 *Integrated Energy Efficiency Ratio (IEER).* For equipment covered by this standard, the IEER shall be calculated using test derived data and the following formula.

$$IEER = (0.020 \cdot A) + (0.617 \cdot B) + (0.238 \cdot C) + (0.125 \cdot D)$$

Where:

A = EER at 100% net capacity at AHRI standard rating conditions
B = EER at 75% net capacity and reduced ambient (see Table 11)
C = EER at 50% net capacity and reduced ambient (see Table 11)
D = EER at 25% net capacity and reduced ambient (see Table 11)

The IEER rating requires that the unit efficiency be determined at 100%, 75%, 50% and 25% load (net capacity) at the conditions specified in Table 11. If the unit, due to its capacity control logic cannot be operated at the 75%, 50%, or 25% load points, then the 75%, 50%, or 25% EER is determined by plotting the tested EER vs. the percent load and using straight line segments to connect the actual performance points. Linear interpolation is used to determine the EER at 75%, 50% and 25% net capacity. For the interpolation, an actual capacity point equal to or less than the required rating point must be used to plot the curves. Extrapolation of the data is not allowed.

If the unit has a variable indoor airflow rate, the external static pressure shall remain constant at the full load rating point as defined in Table 11, but the airflow rate should be adjusted to maintain the unit leaving dry bulb air temperature measured at the full load rating point.

If the unit cannot be unloaded to the 75%, 50%, or 25% load then the unit should be run at the minimum step of unloading at the condenser conditions defined for each of the rating load points and then the efficiency should be adjusted for cyclic performance using the following equation.

$$EER = \frac{LF \cdot \text{Net Capacity}}{LF \cdot [C_D \cdot (P_C + P_{CF})] + P_{IF} + P_{CT}} \tag{15}$$

Where:

Net Capacity	=	Measured net capacity at the lowest machine unloading point operating at the desired part load rating condition, indoor measured capacity minus fan heat, Btu/h
P_C	=	Compressor power at the lowest machine unloading point operating at the desired part load rating condition, watts
P_{CF}	=	Condenser fan power, if applicable at the minimum step of unloading at the desired part load rating condition, watts
P_{IF}	=	Indoor fan motor power at the fan speed for the minimum step of capacity, watts
P_{CT}	=	Control circuit power and any auxiliary loads, watts
C_D	=	Degradation coefficient to account for cycling of the compressor for capacity less than the minimum step of capacity. C_D should be determined using the following equation.

$$C_D = (-0.13 \cdot LF) + 1.13 \tag{16}$$

Where:

LF = Fractional "on" time for last stage at the desired load point.

$$LF = \frac{\left(\frac{\%Load}{100}\right) \cdot (\text{Full Load Unit Net Capacity})}{\text{Part Load Unit Net Capacity}} \tag{17}$$

%Load = The standard rating point i.e. 75%, 50%, 25%.

Table 11. IEER Part-Load Rating Conditions

CONDITIONS	°F	°C
Indoor Air		
Return Air Dry-Bulb Temperature	80.0	26.7
Return Air Wet-Bulb Temperature	67.0	19.4
Indoor Airflow Rate	Note 1	Note 1
Condenser (Air Cooled)		
Entering Dry-Bulb Temperature Outside Air Temperature (OAT)	For % Load > 44.4%, OAT = 0.54 · % Load + 41 For % Load ≤ 44.4%, OAT = 65.0	For % Load > 44.4%, OAT = 0.30 · % Load + 5.0 For % Load ≤ 44.4%, OAT = 18.3
Condenser Airflow Rate (cfm)	Note 2	Note 2
Condenser (Water Cooled)		
Condenser Entering Water Temperature (EWT)	For % Load > 34.8%, EWT = 0.460 · % LOAD + 39 For % Load ≤ 34.8%, EWT = 55.0	For % Load > 34.8% , EWT = 0.256 · % LOAD + 3.8 For % Load ≤ 34.8%, EWT = 12.8
Condenser Water Flow Rate (gpm)	full load flow	full load flow
Condenser (Evaporatively Cooled)		
Entering Wet-Bulb Temperature (EWB)	For % Load > 36.6%, EWB = 0.35 · % Load + 40 For % Load ≤ 36.6%, EWB = 52.8	For % Load > 36.6%, EWB = 0.19 · % Load + 4.4 For % Load ≤36.6%, EWB = 11.6

Table 11. IEER Part-Load Rating Conditions
NOTES: 1 For fixed speed indoor fans the airflow rate should be held constant at the full load airflow rate. For units using discrete step fan control, the fan speed should be adjusted as specified by the controls. 2 Condenser airflow should be adjusted as required by the unit controls for head pressure control.

6.5.4 *Example Calculations.*

Example 1 - Unit with proportional capacity control and can be run at the 75%, 50%, and 25% rating points and has a fixed speed indoor fan.

Assume that the unit has the following measured capacity:

Stage	Ambient	Actual % Load	Net Cap	Cmpr (P_C)	Cond (P_{CF})	Indoor (P_{IF})	Control (P_{CT})	EER
	(F)	(Net Cap)	Btu/h	W	W	W	W	Btu/W
4	95.0	100	114,730	8,707	650	1,050	100	10.92
3	81.5	75	86,047	5,928	650	1,050	100	11.13
2	68.0	50	57,365	3,740	650	1,050	100	10.35
1	65.0	25	28,682	2,080	650	1,050	100	7.39

Using the measured performance you can then calculate the IEER as follows:

$$IEER = (0.020 \cdot 10.92) + (0.617 \cdot 11.13) + (0.238 \cdot 10.35) + (0.125 \cdot 7.39) = 10.48$$

Example 2 – Unit has a single compressor with a fixed speed indoor fan.

Assume the unit has the following measured capacity:

Stage	Ambient	Actual % Load	Net Cap	Cmpr (P_C)	Cond (P_{CF})	Indoor (P_{IF})	Control (P_{CT})	EER
	(°F)	(Net Cap)	Btu/h	W	W	W	W	Btu/W
1	95.0	100	114,730	8,707	650	1,050	100	10.92
1	81.5	104.8	120,264	7,623	650	1,050	100	12.76
1	68.0	108.6	124,614	6,653	650	1,050	100	14.74
1	65.0	109.1	125,214	6,450	650	1,050	100	15.18

The unit cannot unload to the 75%, 50% or 25% points so tests were run with the compressor on at the ambient temperatures specified for 75%, 50%, and 25%

Stage	Ambient	Actual % Load	Net Cap	Cmpr (P_C)	Cond (P_{CF})	Indoor (P_{IF})	Control (P_{CT})	EER	C_D	LF
	(F)	(Net Cap)	Btu/h	W	W	W	W	Btu/W		
1	95.0	100.0	114,730	8,707	650	1,050	100	10.92		
1	81.5	104.8	120,264	7,623	650	1,050	100	12.76		
		75.0			Adjusted for Cyclic Performance			11.81	1.037	0.715
1	68.0	108.6	124,614	6,653	650	1,050	100	14.74		
		50.0			Adjusted for Cyclic Performance			12.08	1.070	0.460
1	65.0	109.1	125,214	6,450	650	1,050	100	15.18		
		25.0						9.76	1.100	0.229

Calculate the Load Factor (LF) and the C_D factors and then calculate the adjusted performance for the 75%, 50%, and 25% points and then calculate the IEER.

The following is an example of the C_D calculation for the 50% point:

$$LF = \frac{\left(\frac{50}{100}\right)\cdot 114,730}{124,614} = .460$$

$$C_D = (-0.13 \cdot .460) + 1.13 = 1.070$$

$$EER_{50\%} = \frac{.460 \times 124,614}{.460 \cdot (1.070 \cdot (6,653 + 650)) + 1,050 + 100} = 12.08$$

$$IEER = (0.020 \cdot 10.92) + (0.617 \cdot 11.81) + (0.238 \cdot 12.08) + (0.125 \cdot 9.76) = 11.60$$

Example 3 – Unit has two refrigeration circuits with one compressor in each circuit and two stages of capacity with a fixed speed indoor fan.

Assume the unit has the following measured performance.

Stage	Ambient	Actual % Load	Net Cap	Cmpr (P_C)	Cond (P_{CF})	Indoor (P_{IF})	Control (P_{CT})	EER
	(F)	(Net Cap)	Btu/h	W	W	W	W	Btu/W
2	95.0	100	114,730	8,707	650	1,050	100	10.92
1	71.0	55.5	63,700	3,450	325	1,050	100	12.93
1	68.0	55.9	64,100	3,425	325	1,050	100	13.08
1	65.0	56.1	64,400	3,250	325	1,050	100	13.63

The unit can unload to get to the 75% point, but cannot unload to get to the 50% and 25% points so additional tests are run at the 50% and 25% load ambients with the stage 1 loading.

Calculate the 50% and 25% load factors and C_D factors as shown below.

Stage	Ambient	Actual % Load	Net Cap	Cmpr (P_C)	Cond (P_{CF})	Indoor (P_{IF})	Control (P_{CT})	EER	C_D	LF
	(F)	(Net Cap)	Btu/h	W	W	W	W	Btu/W		
2	95.0	100.0	114,730	8,707	650	1,050	100	10.92		
1	71.0	55.5	63,700	3,450	325	1,050	100	12.93		
		75.0					interpolation	12.05		
1	68.0	55.9	64,100	3,425	325	1,050	100	13.08		
		50.0			Adjusted for Cyclic Performance			12.60	1.014	0.895
1	65.0	56.1	64,400	3,250	325	1,050	100	13.63		
		25.0						10.04	1.072	0.445

Calculate the Load Factor (LF) and the C_D factors and then calculate the adjusted performance for the 75%, 50%, and 25% points and then calculate the IEER:

$$IEER = (0.020 \cdot 10.92) + (0.617 \cdot 12.05) + (0.238 \cdot 12.60) + (0.125 \cdot 10.04) = 11.91$$

Example 4 – Unit has three refrigeration circuits with one compressor in each circuit and three stages of capacity with a fixed speed indoor fan.

Assume the unit has the following measured performance.

Stage	Ambient	Actual % Load	Net Cap	Cmpr (P_C)	Cond (P_{CF})	Indoor (P_{IF})	Control (P_{CT})	EER
	(F)	(Net Cap)	Btu/h	W	W	W	W	Btu/W
3	95.0	100.0	114,730	8,707	650	1,050	100	10.92
2	79.5	71.3	81,841	5,125	433	1,050	100	12.20
1	65.0	38.3	43,980	2,250	217	1,050	100	12.16

The stage 1 operates at 38.3% capacity which is above the minimum 25% load point, but because the ambient condition was 65 °F, another test at the 25% load ambient condition is not required as it would be the same test point.

Calculate the IEER which requires interpolation for the 75% and 50% point and the use of the degradation factor for the 25% point.

Stage	Ambient	Actual % Load	Net Cap	Cmpr (P_C)	Cond (P_{CF})	Indoor (P_{IF})	Control (P_{CT})	EER	C_D	LF
	(F)	(Net Cap)	Btu/h	W	W	W	W	Btu/W	NA	NA
3	95.0	100.0	114,730	17,414	1,300	1,050	100	10.92	NA	NA
2	79.5	71.3	81,841	4,950	433	1,050	100	12.53	NA	NA
		75.0				interpolation		12.32	NA	NA
2	79.5	71.3	81,841	4,950	433	1,050	100	12.53	NA	NA
1	65.0	38.3	43,980	2,250	217	1,050	100	12.16	NA	NA
		50.0				interpolation		12.57	NA	NA
1	65.0	38.3	43,980	2,250	217	1,050	100	12.16	NA	NA
		25.0			Adjusted for Cyclic Performance			10.13	1.045	0.652

$$IEER = (0.02\cdot 10.92) + (0.617\cdot 12.32) + (0.238\cdot 12.57) + (0.125\cdot 10.13) = 12.08$$

Example 5 – Unit is a VAV unit and has 5 stages of capacity and a variable speed indoor.

Assume the unit has the following measured performance.

Stage	Ambient	Actual % Load	Net Cap	Cmpr (P_C)	Cond (P_{CF})	Indoor (P_{IF})	Control (P_{CT})	EER
	(F)	(Net Cap)	Btu/h	W	W	W	W	Btu/W
5	95.0	100.0	229,459	17,414	1,300	2,100	200	10.92
4	85.1	81.7	187,459	11,444	1,300	1,229	150	13.27
3	74.0	61.0	140,064	6,350	1,300	575	150	16.72
2	69.6	52.9	121,366	6,762	650	374	150	15.29
1	65.0	30.6	70,214	2,139	650	85	150	23.2

This unit can unload down to 30.6% so a degradation calculation will be required but because the stage 1 was already run at the lowest ambient and the ambient for the 25% load point no additional tests are required.

Using this data you can then calculate the standard load points.

Stage	Ambient	Actual % Load	Net Cap	Cmpr (P_C)	Cond (P_{CF})	Indoor (P_{IF})	Control (P_{CT})	EER	C_D	LF
	(F)	(Net Cap)	Btu/h	W	W	W	W	Btu/W		
5	95.0	100.0	229,459	17,414	1,300	2,100	200	10.92		
4	85.1	81.7	187,459	11,444	1,300	1,229	150	13.27		
3	74.0	61.0	140,064	6,350	1,300	575	150	16.72		
		75.0					interpolation	14.39		
2	69.6	52.9	121,366	6,762	650	374	150	15.29		
1	65.0	30.6	70,214	2,139	650	85	150	23.22		
		50.0					interpolation	16.32		
		25.0			Adjusted for Cyclic Performance			22.34	1.024	0.817

Note: Blank space equals NA.

With this you can then calculate the IEER:

$$IEER = (0.02 \cdot 10.92) + (0.617 \cdot 14.39) + (0.238 \cdot 16.32) + (0.125 \cdot 22.34) = 15.78$$

6.6 *Test Tolerances (Applies to all products covered by this standard).* To comply with this standard, measured test results shall not be less than 95% of Published Ratings for capacities, SEER, HSPF, EER values, and COP values and not less than 90%of Published Ratings for IEER and SCHE values.

Section 7. Minimum Data Requirements for Published Ratings

7.1 *Minimum Data Requirements for Published Ratings.* As a minimum, Published Ratings shall consist of the following information:

a. For VRF Multi-Split Air-Conditioners <65,000 Btu/h [19,000 W]
   1. Standard Rating Cooling Capacity Btu/h [W]
   2. Seasonal Energy Efficiency Ratio, SEER Btu/(W·h)

b. For VRF Multi-Split Air-Conditioners ≥ 65,000 Btu/h [19,000 W]
   1. Standard Rating Cooling Capacity Btu/h [W]
   2. Energy Efficiency Ratio, EER Btu/(W·h)
   3. Integrated Energy Efficiency Ratio, IEER (Integrated Part-Load Value, IPLV is Superseded by IEER January 1, 2010)

c. For all VRF Multi-Split Heat Pumps <65,000 Btu/h [19,000 W]
   1. Standard Rating Cooling Capacity Btu/h [W]
   2. Seasonal Energy Efficiency Ratio, SEER Btu/(W·h)
   3. High Temperature Heating Standard Rating Capacity Btu/(W·h) [W]
   4. Region IV Heating Seasonal Performance Factor, HSPF, minimum design heating requirement (W·h)

d. For VRF Multi-Split Heat Pumps ≥ 65,000 Btu/h [19,000 W]
   1. Standard Rating Cooling Capacity Btu/h [W]
   2. Energy Efficiency Ratio, EER Btu/(W·h)
   3. Integrated Energy Efficiency Ratio, IEER (Integrated Part-Load Value, IPLV is Superseded by IEER January 1, 2010)
   4. High Temperature Heating Standard Rating Capacity Btu/h [W]
   5. High Temperature Coefficient of Performance
   6. Low Temperature Heating Standard Rating Capacity Btu/h [W]
   7. Low Temperature Coefficient of Performance

e. For VRF Multi-Split Heat Recovery Heat Pumps
   1. Ratings Appropriate in 7 (c) (d) above
   2. Simultaneous Cooling and Heating Efficiency (SCHE) (50% heating/50% cooling)

f. For VRF Multi-Split Heat Pumps Systems that Use a Water Source for Heat Rejection
   1. Standard Rating Cooling Capacity Btu/h [W]
   2. Energy Efficiency Ratio, EER Btu/(W·h)
   3. Integrated Energy Efficiency Ratio, IEER (Integrated Part-Load Value, IPLV is Superseded by IEER January 1, 2010)
   4. Heating Standard Rating Capacity Btu/h [W]
   5. Heating Coefficient of Performance
   6. Simultaneous Cooling and Heating Efficiency (SCHE) (50% heating/50% cooling)/ (Heat Recovery models only)

7.2 *Latent Cooling Capacity Designation*. The moisture removal designation shall be published in the manufacturer's specifications and literature. The value shall be expressed consistently in either gross or net in one or more of the following forms:

a. Sensible cooling capacity/total cooling capacity ratio (sensible heat ratio) and total capacity, Btu/h [W]
b. Latent cooling capacity and total cooling capacity, Btu/h [W]
c. Sensible cooling capacity and total cooling capacity, Btu/h [W]

7.3 *Rating Claims*. All claims to ratings within the scope of this standard shall include the statement "Rated in accordance with AHRI Standard 1230". All claims to ratings outside the scope of this standard shall include the statement: "Outside the scope of AHRI Standard 1230". Wherever Application Ratings are published or printed, they shall include a statement of the conditions at which the ratings apply.

Section 8. Operating Requirements

8.1 *Operating Requirements.* Unitary equipment shall comply with the provisions of this section such that any production unit will meet the requirements detailed herein.

8.2 *Operating Requirements for Systems < 65,000 Btu/h [19,000 W]*

8.2.1 *Maximum Operating Conditions Test for Systems < 65,000 Btu/h [19,000 W]*. Unitary equipment shall pass the following maximum operating conditions test with an indoor-coil airflow rate as determined under 6.1.5.1.

8.2.1.1 *Temperature Conditions.* Temperature conditions shall be maintained as shown in Table 6.

8.2.2 *Voltages.* The test shall be run at the Range A minimum utilization voltage from AHRI Standard 110, Table 1, based upon the unit's nameplate rated voltage(s). This voltage shall be supplied at the unit's service connection and at rated frequency.

8.2.3 *Procedure.* The equipment shall be operated for one hour at the temperature conditions and voltage specified.

8.2.4 *Requirements.* The equipment shall operate continuously without interruption for any reason for one hour.

8.2.4.1 Units with water-cooled condensers shall be capable of operation under these maximum conditions at a water pressure drop not to exceed 413.5 in H_2O [103 kPa], measured across the unit.

8.3 *Voltage Tolerance Test for Systems < 65,000 Btu/h [19,000 W]*. Unitary equipment shall pass the following voltage tolerance test with a cooling coil airflow rate as determined under 6.1.5.1.

8.3.1 *Temperature Conditions.* Temperature conditions shall be maintained at the standard cooling (and/or standard heating, as required) steady state conditions as shown in Table 6.

8.3.2 *Voltages.*

8.3.2.1 Tests shall be run at the Range B minimum and maximum utilization voltages from ARI Standard 110, Table 1, based upon the unit's nameplate rated voltage(s). These voltages shall be supplied at the unit's service connection and at rated frequency. A lower minimum or a higher maximum voltage shall be used, if listed on the nameplate.

8.3.2.2 The power supplied to single phase equipment shall be adjusted just prior to the shut-down period (8.3.3.2) so that the resulting voltage at the unit's service connection is 86% of nameplate rated voltage when the compressor motor is on locked-rotor. (For 200V or 208V nameplate rated equipment the restart voltage shall be set at 180V when the compressor motor is on locked rotor). Open circuit voltage for three-phase equipment shall not be greater than 90% of nameplate rated voltage.

8.3.2.3 Within one minute after the equipment has resumed continuous operation (8.3.4.3), the voltage shall be restored to the values specified in 8.3.2.1.

8.3.3 *Procedure*.

8.3.3.1 The equipment shall be operated for one hour at the temperature conditions and voltage(s) specified.

8.3.3.2 All power to the equipment shall be interrupted for a period sufficient to cause the compressor to stop (not to exceed five seconds) and then restored.

8.3.4 *Requirements.*

8.3.4.1 During both tests, the equipment shall operate without failure of any of its parts.

8.3.4.2 The equipment shall operate continuously without interruption for any reason for the one hour period preceding the power interruption.

8.3.4.3 The unit shall resume continuous operation within two hours of restoration of power and shall then operate continuously for one-half hour. Operation and resetting of safety devices prior to establishment of continuous operation is permitted.

8.4 *Low-Temperature Operation Test for Systems < 65,000 Btu/h [19,000 W] (Cooling).* Unitary equipment shall pass the following low-temperature operation test when operating with initial airflow rates as determined in 6.1.5.1 and 6.1.6 and with controls and dampers set to produce the maximum tendency to frost or ice the evaporator, provided such settings are not contrary to the manufacturer's instructions to the user.

8.4.1 *Temperature Conditions.* Temperature Conditions shall be maintained as shown in Table 6.

8.4.2 *Procedure.* The test shall be continuous with the unit on the cooling cycle, for not less than four hours after establishment of the specified temperature conditions. The unit will be permitted to start and stop under control of an automatic limit device, if provided.

8.4.3 *Requirements.*

8.4.3.1 During the entire test, the equipment shall operate without damage or failure of any of its parts.

8.4.3.2 During the entire test, the air quantity shall not drop more than 25% from that determined under the Standard Rating test.

8.4.3.3 During the test and during the defrosting period after the completion of the test, all ice or meltage must be caught and removed by the drain provisions.

8.5 *Insulation Effectiveness Test (Cooling). Test for Systems < 65,000 Btu/h [19,000 W].* Unitary equipment shall pass the following insulation effectiveness (aka insulation efficiency test) when operating with airflow rates as determined in 6.1.5.1 and 6.1.6 with controls, fans, dampers, and grilles set to produce the maximum tendency to sweat, provided such settings are not contrary to the manufacturer's instructions to the user.

8.5.1 *Temperature Conditions.* Temperature conditions shall be maintained as shown in Table 6.

8.5.2 *Procedure.* After establishment of the specified temperature conditions, the unit shall be operated continuously for a period of four hours.

8.5.3 *Requirements.* During the test, no condensed water shall drop, run, or blow off from the unit casing.

8.6 *Condensate Disposal Test (Cooling). Test for Systems < 65,000 Btu/h [19,000 W].* Unitary equipment which rejects condensate to the condenser air shall pass the following condensate disposal test when operating with airflow rates as determined in 6.1.5.1 and 6.1.6 and with controls and dampers set to produce condensate at the maximum rate, provided such settings are not contrary to the manufacturer's instructions to the user. (This test may be run concurrently with the Insulation Effectiveness Test (8.5)).

8.6.1 *Temperature Conditions.* Temperature conditions shall be maintained as shown in Table 6.

8.6.2 *Procedure.* After establishment of the specified temperature conditions, the equipment shall be started with its condensate collection pan filled to the overflowing point and shall be operated continuously for four hours after the condensate level has reached equilibrium.

8.6.3 *Requirements.* During the test, there shall be no dripping, running-off, or blowing-off of moisture from the unit casing.

8.7 *Test Tolerance for Systems <65,000 Btu/h [19,000 W].* The conditions for the tests outlined in Section 8 are average values subject to tolerances of ± 1.0°F [± 0.6°C] for air wet-bulb and dry-bulb temperatures and ± 1.0% of the reading for voltages.

8.8 *Operating Requirements for Systems ≥ 65,000 Btu/h [19,000 W].*

8.8.1 *Maximum Operating Conditions Test (Cooling and Heating) Systems ≥ 65,000 Btu/h [19,000 W].* Multi-Split Air-Conditioners and Heat Pumps shall pass the following maximum cooling and heating operating conditions test with an indoor coil airflow rate as determined under 6.3.1 (refer to test for equipment with optional air cooling coils in 6.3.3).

8.8.2 *Temperature Conditions.* Temperature conditions shall be maintained as shown in Table 8.

8.8.3 *Voltages.* Tests shall be run at the minimum and maximum utilization voltages of Voltage Range B as shown in Table 1 of AHRI Standard 110, at the unit's service connection and at rated frequency.

8.8.4 *Procedure.*

8.8.4.1 Multi-split Air-Conditioners and Heat Pumps shall be operated continuously for one hour at the temperature conditions and voltage(s) specified.

8.8.4.2 All power to the unitary equipment shall be interrupted for a period sufficient to cause the compressor to stop (not to exceed five seconds) and then be restored.

8.8.5 *Requirements.*

8.8.5.1 During both tests, the unitary equipment shall operate without failure of any of its parts.

8.8.5.2 The unit shall resume continuous operation within one hour of restoration of power and shall then operate continuously for one hour. Operation and resetting of safety devices prior to establishment of continuous operation is permitted.

8.8.5.3 Units with water-cooled condensers shall be capable of operation under these maximum conditions at a water-pressure drop not to exceed 413.5 in H_2O [103 kPa] measured across the unit.

8.8.6 *Maximum Operating Conditions Test for Equipment with Optional Outdoor Cooling Coil.* Multi-split Air Conditioners and Heat Pumps which incorporate an outdoor air cooling coil shall use the conditions, voltages, and procedure (8.8.1 through 8.8.4) and meet the requirements of 8.8.5 except for the following changes.

a. Outdoor air set as in 6.3.1

b. Return air temperature conditions shall be 80.0ºF [26.7ºC] dry-bulb, 67.0ºF [19.4ºC] wet-bulb

c. Outdoor air entering outdoor air cooling coil shall be 115ºF [46.1ºC] dry-bulb and 75.0ºF [23.9ºC] wet-bulb

8.9 *Cooling Low Temperature Operation Test for Systems ≥ 65,000 Btu/h [19,000 W].* Multi-split Air-Conditioners and Heat Pumps shall pass the following low-temperature operation test when operating with initial airflow rates as determined in 6.3.1, 6.3.4, and with controls and dampers set to produce the maximum tendency to frost or ice the indoor coil, provided such settings are not contrary to the manufacturer's instructions to the user.

8.9.1 *Temperature Conditions.* Temperature conditions shall be maintained as shown in Table 8.

8.9.2 *Voltage and Frequency.* The test shall be performed at nameplate rated voltage and frequency.

For air-conditioners and heat pumps with dual nameplate voltage ratings, tests shall be performed at the lower of the two voltages.

8.9.3 *Procedure.* The test shall be continuous with the unit in the cooling cycle for not less than four hours after establishment of the specified temperature conditions. The unit will be permitted to start and stop under control of an automatic limit device, if provided.

8.9.4 *Requirements.*

8.9.4.1 During the entire test, the unitary equipment shall operate without damage to the equipment.

8.9.4.2 During the entire test, the indoor airflow rate shall not drop more than 25% from that specified for the Standard Rating test.

8.9.4.3 During all phases of the test and during the defrosting period after the completion of the test, all ice or meltage must be caught and removed by the drain provisions.

8.10 *Insulation Efficiency Test (Cooling) for Systems ≥ 65,000 Btu/h [19,000 W]*. Multi-Split Air-Conditioners and Heat Pumps shall pass the following Insulation Efficiency Test when operating with airflow rates as determined in 6.3.1, 6.3.4, and with controls, fans, dampers, and grilles set to produce the maximum tendency to sweat, provided such settings are not contrary to the manufacturer's instructions to the user.

8.10.1 *Temperature Conditions*. Temperature conditions shall be maintained as shown in Table 8.

8.10.2 *Procedure*. After establishment of the specified temperature conditions, the unit shall be operated continuously for a period of four hours.

8.10.3 *Requirements*. During the test, no condensed water shall drop, run, or blow off from the unit casing.

8.11 *Condensate Disposal Test (Cooling) for Systems ≥ 65,000 Btu/h [19,000 W]*. Multi-Split Air-Conditioners and Heat Pumps which reject condensate to the condenser air shall pass the following condensate disposal test when operating with airflow rates as determined in 6.3.1, 6.3.4, and with controls and dampers set to produce condensate at the maximum rate, provided such settings are not contrary to the manufacturer's instructions to the user (This test may be run concurrently with the insulation efficiency test (8.10)).

8.11.1 *Temperature Conditions.* Temperature conditions shall be maintained as shown in Table 8.

8.11.2 *Procedure.* After establishment of the specified temperature conditions, the equipment shall be started with its condensate collection pan filled to the overflowing point and shall be operated continuously for four hours after the condensate level has reached equilibrium.

8.11.3 *Requirements*. During the test, there shall be no dripping, running-off, or blowing-off of moisture from the unit casing.

8.12 *Tolerances for Systems ≥ 65,000 Btu/h [19,000 W]*. The conditions for the tests outlined in Section 8.2 and 8.3 are average values subject to tolerances of ±1.0ºF [±0.6ºC] for air wet-bulb and dry-bulb temperatures, ±0.5ºF [±0.3ºC] for water temperatures, and ±1.0% of the readings for specified voltage.

8.13 *Performance Requirements for Systems using a Water Source for Heat Rejection.*

8.13.1 *Capacity Requirements*

8.13.1.1 To be consistent with ISO 13256-1-2, water-to-air and brine-to-air heat pumps shall be designed and produced such that any production unit will meet the applicable requirements of this standard.

8.13.1.2 For heat pumps with capacity control, the performance requirements tests shall be conducted at maximum capacity.

8.13.2 *Maximum Operating Conditions Test*

8.13.2.1 *Test conditions.* The maximum operating conditions tests shall be conducted for cooling and heating at the test conditions established for the specific applications specified in Tables 12 and 13. Heat pumps intended for use in two or more applications shall be tested at the most stringent set of conditions specified in Tables 12 and 13.

8.13.2.2 *Test Procedures*

8.13.2.2.1 The equipment shall be operated continuously for one hour after the specified temperatures have been established at each specified voltage level.

8.13.2.2.2 The 110% voltage test shall be conducted prior to the 90% voltage test.

8.13.2.2.3 All power to the equipment shall be interrupted for three minutes at the conclusion of the one hour test at the 90% voltage level and then restored for one hour.

8.13.2.3 *Test Requirements.* Heat pumps shall meet the following requirements when operating at the conditions specified in Tables 12 and 13.

8.13.2.3.1 During the entire test, the equipment shall operate without any indication of damage.

8.13.2.3.2 During the test period specified in 8.13.2.2.1, the equipment shall operate continuously without tripping any motor overload or other protective devices.

8.13.2.3.3 During the test period specified in 8.13.2.2.3, the motor overload protective device may trip only during the first five minutes of operation after the shutdown period of three minutes. During the remainder of the test period, no motor overload protective device shall trip. For those models so designed that resumption of operation does not occur within the first five minutes after the initial trip, the equipment may remain out of operation for no longer than 30 minutes. It shall then operate continuously for the remainder of the test period.

8.13.3 *Minimum Operating Conditions Test.* Heat pumps shall be tested at the minimum operating test conditions for cooling and heating at the test conditions established for the specific applications specified in Tables 14 and 15. Heat pumps intended for use in two or more applications shall be tested at the most stringent set of conditions specified in Tables 14and 15.

8.13.3.1 *Test Procedures.* For the minimum operating cooling test, the heat pump shall be operated continuously for a period of no less than 30 minutes after the specified temperature conditions have been established. For the minimum operating heating test, the heat pump shall soak for 10 minutes with liquid at the specified temperature circulating through the coil. The equipment shall then be started and operated continuously for 30 minutes.

8.13.3.2 *Test Requirements.* No protective device shall trip during these tests and no damage shall occur to the equipment.