BETTERBRICKS
Powerful Energy Ideas. Delivered by NEEA.

VERY HIGH EFFICIENCY DEDICATED OUTSIDE AIR SYSTEM PILOT PROJECT REPORT

MARCH 2020

TABLE OF CONTENTS

EXECUTIVE SUMMARY

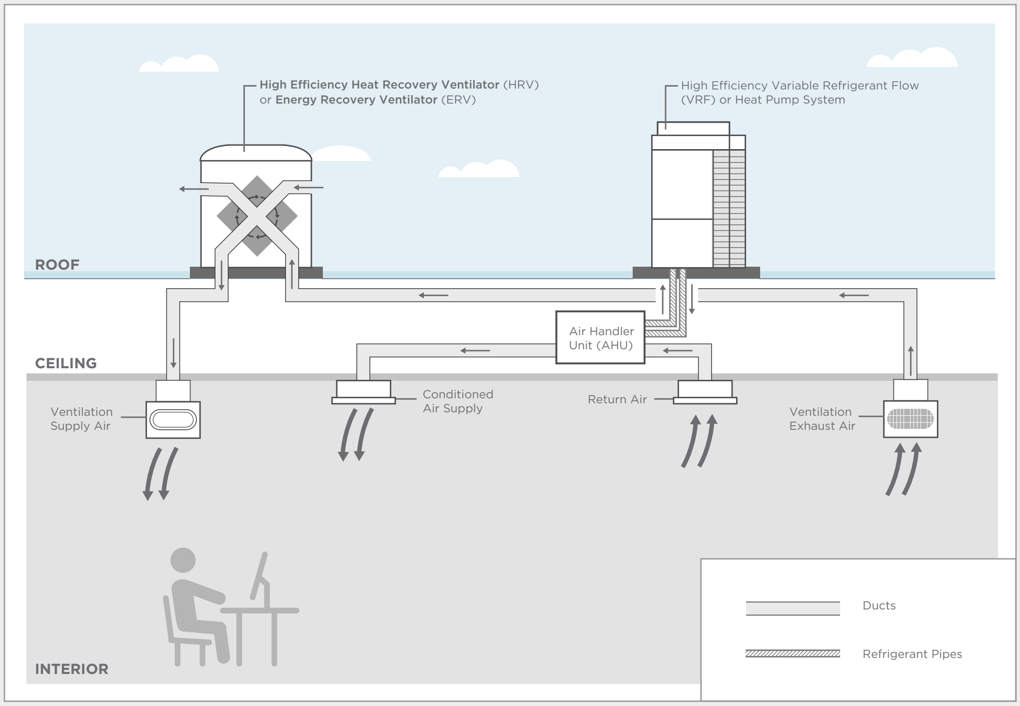
PILOT PROJECT OVERVIEW

As a result of the scanning of emerging technologies in the commercial building sector, a dedicated outside air system (DOAS) was identified as an effective strategy for significant energy savings potential in the Northwest. The following report details the findings from eight pilot projects conducted by BetterBricks, a commercial resource of the Northwest Energy Efficiency Alliance (NEEA), in partnership with local utilities and energy efficiency program partners. These pilots tested a DOAS approach that included dedicated ventilation air (decoupled from primary heating and cooling air) with high-efficiency heat recovery and a high-efficiency heating and cooling system. This equipment was coupled with key design principles to maximize system performance. The pilot projects, conducted on mostly small-to-medium-sized buildings, helped validate energy savings assumptions and gain a better understanding of the design and installation process.

Technology Overview

DOAS separates the functions of building ventilation and building heating and cooling so that each of these critical building functions can be optimally controlled. Typically, heating and cooling is controlled on a setpoint with little variance from hour-to-hour and day-to-day based on outdoor conditions. These building heating and cooling setpoints have to be met regardless of whether there is anyone in the building. Ventilation, on the other hand, is primarily for building occupants, requires much lower air flows than heating and cooling, and can be significantly reduced or turned off when the building is not occupied. This presents significant opportunity for reductions in fan energy and reconditioning of ventilation air, and can create noticeable energy savings by separating these two functions. Further, even more substantial energy savings can be realized when applying the DOAS approach while using very high efficiency heat recovery and a right-sized, very high efficiency heating and cooling system.

Figure 1: Very High Efficiency DOAS Diagram



PERFORMANCE

				
Location	**Portland, OR**	**Corvallis, OR**	**Seattle, WA**	**Corvallis, OR**
Building type	Second-story law office space	Single-story government office building	Third-floor office space	Restaurant
Conditioned area (sq. ft.)	11,615	3,770 of 13,200	5,911	1,360
Total project cost (per sq. ft.)	$15.61	$11.47	$16.83	$27.50
Reduction in building energy use	63%	39%*	42%	8%**
Reduction in HVAC energy use	72%	70%*	69%	43%**
Existing HVAC system	9 RTUs (35 tons in total)	2 4-ton RTUs (in the 3,770 sq. ft. retrofitted zones)	14-ton electric resistance RTU	7.5-ton RTU
New HVAC system	16-ton Mitsubishi VRF 4 Ventacity VS1000RT HRVs	4-ton Mitsubishi multi-zone mini-split heat pump 1 Ventacity VS1000RT HRV	14-ton Mitsubishi VRF 1 Ventacity VS1000RT HRV	2-ton Fujitsu single-zone ductless heat pump 3 3-ton Daikin single-zone ductless heat pumps 1 Ventacity VS1000RT HRV

**The HVAC upgrade was limited to two (28% of floor area) of the five zones, and saved 11% of whole-building energy use and 19% in HVAC energy use. For a more accurate illustration of total HVAC and whole-building energy use, the results in the table above are extrapolated and represent whole-building impacts if all systems and zones were converted in the same manner.*

***This was atypical for restaurants due to low occupancy and high take out in a small floor area arrangement; some building uses such as restaurants, hospitals and warehouses are less ideal for this HVAC system and require careful analysis and design to be effective.*

				
Location	**Seattle, WA**	**Darby, MT**	**Libby, MT**	**Portland, OR**
Building type	Airport terminal building	Dormitory	Office building	Restaurant
Conditioned area (sq. ft.)	25,200	11,000 (per dorm)	5,735	1,147
Total project cost (per sq. ft.)	$36.85	$9.64	$21.90	$30.99
Reduction in building energy use	61%	24%	29%	20%
Reduction in HVAC energy use	85%	52%	45%	73%
Existing HVAC system	3 RTUs (95 tons in total)	5 electric forced-air furnaces 1 exhaust fan	1 electric boiler 2 swamp coolers 1 6-ton heat pump RTU 1 server room heat pump	1 3-ton RTU
New HVAC system	4 Mitsubishi VRF systems (32 tons in total) 3 Ventacity VS1000RT HRVs	4 3-ton York split system heat pumps 1 4-ton York split system heat pump 1 Ventacity VS1000RT HRV	2 4.5-ton Mitsubishi heat pump units 1 Ventacity VS1000RT HRV	1 3-ton Daikin multi-zone ductless heat pump 1 Ventacity VS1000RT HRV

SYSTEM BENEFITS

The pilots demonstrated that HVAC conversions to very high efficiency DOAS resulted in the following benefits and outcomes:

- **Lower Energy Bills**—Every project in the pilot resulted in year-round HVAC and total building energy savings.
- **Improved Indoor-Air Quality**—The improvement most reported by building occupants was vastly enhanced indoor-air quality. This is a result of the HRV or ERV bringing in 100-percent fresh and filtered outside air with no recirculation of outgoing air.
- **Increased Comfort**—Building occupant comfort was considerably improved through a combination of improved air quality, fresh air delivered close to room temperature resulting in more consistent temperatures, and enhanced control of heating and cooling by zone and occupancy rates.
- **Reduced Equipment and Ductwork**—Very high efficiency DOAS saves roof space by reducing system size and ductwork needs.
- **Enabled Demand Control Ventilation (DCV)**—The opportunity to automatically ensure that ventilation rates reflect occupancy pattern (and rely less on pre-programmed schedules) is a potentially significant benefit of the new system. However, as documented in the pilots, it's a challenge to educate building occupants on proper control usage.

LESSONS

The following are common takeaways and lessons learned regarding very high efficiency DOAS installations:

- **Training is Important**—Because the very high efficiency DOAS design and installation approach are new to North America, many designers and installers aren't familiar with design and installation best practices.
- **Right-Sizing Equipment**—The pilot projects have demonstrated that many standard heating and cooling systems are significantly oversized.
- **Tightly Sealed, Insulated Ductwork**—Properly sealed ductwork is critical to ensure very high efficiency DOAS operates efficiently. In multiple pilot projects, compromised system performance stemmed from air leaks in the ventilation ducting and ductwork in the unconditioned attic spaces.
- **Current Software Isn't Sufficient for System Modeling**—VRFs and HRVs are difficult to individually model, and even more difficult to model when combined. Built-in modeling assumptions in most current software tend to result in significantly overestimating loads and baseline system performance, which necessitates supplemental calculations for a very high efficiency HRV. For instance, for one of the pilots, it's likely that neither the VRF manufacturer's software nor the software used by the engineer (eQuest) took adequate account of the reduction in both heating and cooling loads made possible by use of the project's very

high efficiency HRV. While eQuest can be useful for many building modeling tasks, nearly all of the modelers consulted in the course of the pilot project agree that eQuest is not capable of accurately modeling the kind of conversions that the project accomplishes. Additionally, eQuest cannot accommodate the common work-arounds used to improve accuracy in Energy Plus modeling. The remaining pilot projects used Energy Plus for modeling, which has resulted in recommendations for at least some downsizing of system capacities, even if not to an optimal degree.

- **Cross-Team Communication is Critical**—This type of conversion, using brand new HRV technology, is unfamiliar to most project teams. In cases where team members weren't aligned in regard to design and installation, there were numerous problems, increased cost, and installation mistakes or failures.
- **Existing Buildings Tend to Be Unique**—No design guidelines work consistently, and many buildings come with unique physical or code obstacles. For example, restaurants are particularly difficult, as the airflow is dominated by vent-hood operation, and energy use is dominated by cooking, hot water and refrigeration loads. Savings percentages will be smaller, but large enough to be worthwhile when the dining room area is a large fraction of total conditioned space.

CONCLUSION

The pilot projects achieved the overall goal of demonstrating the potential for cost-effective and substantial energy and demand savings in existing commercial buildings, while maintaining or improving indoor air quality and occupant comfort. This was particularly the case when project specifications and guidelines were followed.

On average, participants saw a 70% reduction of their actual HVAC energy use and a 42% reduction in actual whole-building energy use. Even if these pilot buildings had started with standard code-minimum equipment prior to the conversion, modeling still shows significant average energy savings of 65% for HVAC and 36% for the entire building.

PROJECT REPORT

BACKGROUND

In 2010, in addition to commercial building benchmarking work in Portland and Seattle, NEEA spoke with Pacific Northwest energy efficiency stakeholders about the state of energy performance among small-to-medium existing commercial buildings. There were a few key takeaways:

- Aside from lighting and custom programs, there were few utility programs focused on significantly reducing the energy use in this sector, especially in smaller buildings that primarily use packaged HVAC systems.
- The Energy Use Index (EUI in Btu/sq. ft./year) values for these buildings vary significantly, with the worst often using 5 times more energy per square foot than the best.
- HVAC best practices common in Europe involving heating and cooling systems that are completely separate from very high efficiency ventilation systems showed that savings potential was high for all but the most efficient existing buildings.
- The amount of building square footage in the Northwest is very large, with NEEA's data suggesting that more than half the existing commercial building square footage in the region could be targeted for HVAC system conversion. This would cost-effectively deliver very large regional savings and radically lower building EUIs.

Additionally, related marketplace trends were increasing the visibility of commercial building energy use. For instance, climate change response policies in certain states and jurisdictions generated building EUI benchmarking initiatives in an attempt to quantify carbon footprint contributions from building energy use. Early data showed even some of the best-rated new buildings were using a lot more energy than they were modeled to use.

The best buildings typically use a combination of a very high efficiency ventilation system, separate from a very high efficiency heating and cooling system, with the overall system designed to maximize the time that HVAC systems are off,[1] using greatly simplified controls and control strategies. The worst performing buildings tend to use complicated HVAC systems that combine space conditioning and ventilation airflows, exhibiting large numbers of hours in a simultaneous heating and cooling mode, often by system design.

Many of the technologies needed to significantly reduce building energy consumption have been in widespread use in Europe for many years. The key technology required for enabling significant HVAC savings—a very high performance HRV or ERV[2] and associated distribution systems and controls—is available from multiple manufacturers in a wide range of capacities in the European market, but much less so in North America.

[1]This means heating and cooling systems have no load to meet to maintain their setpoint, and ventilation is off when spaces are not occupied (barring other reasons for running the HRV/ERV, such as for economizing).

[2]An HRV transfers sensible energy only, while an ERV transfers both sensible and latent energy (humidity). The Pacific Northwest climate calls for HRV use rather than ERV, but there were hardly any HRVs available in the market. This is most likely because the American HVAC industry is focused primarily on the cooling function, and has shown little interest in heat pump technology.

A high-level NEEA analysis in 2011 suggested that the installed cost of a less efficient built-up ERV system would be as high or higher than the cost of more expensive packaged European technology— but the savings would be half or less than those possible with the European equipment. This was partially due to lower HRV efficiencies requiring the heating and cooling of ventilation air before delivering it, and partially due to poor efficiencies of standard fans in the North American models. This meant conversions using the available technologies would not be cost-effective by the standards of NEEA and utility programs.

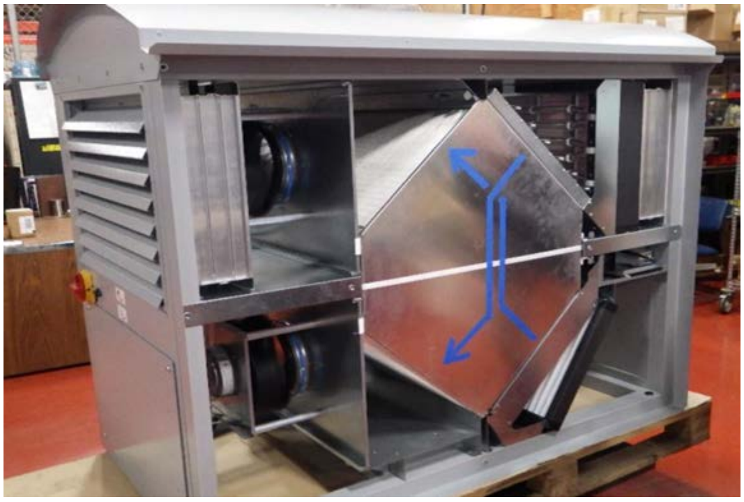
Because of this, the pilot project for very high efficiency DOAS was sidelined until an adequate HRV or ERV technology arrived in the market.

In late 2014, an opportunity arrived in the form of a local Portland entrepreneur with an interest in solving the problem. After a period of due diligence and technology licensing activity, NEEA formed a partnership with Cinagro Ventures (later renamed Ventacity Systems) to begin a regional market transformation program, and national product line distribution. Early work involved redesigning a selected HRV product line from a well-respected European manufacturer, adapting it to the needs of the North American market, and lab-testing to validate performance. The product redesign primarily focused on developing a product variant that could be installed on existing rooftop curbs, with downward supply airflow and upward exhaust airflow (which is not customary in Europe).

PILOT PROJECT OVERVIEW

By mid-2015, Ventacity Systems had their first product—the VS1000RT—undergoing the UL-listing process, which was completed by mid-2016. Passive House Certification was awarded a year later. Unofficial lab testing illustrated the efficiency range of Ventacity units by fall of 2015, concurrent with NEEA's recruiting for pilot project buildings. By late 2015, planning and construction of the first pilot project was underway: A gut-remodel of one floor of a two-story historic building in downtown Portland, Ore., that would be a law office when finished. That pilot was followed quickly by two projects in Corvallis, Ore., (two zones of a five-zone state government district office and a pizza restaurant), and two projects in Seattle (another gut remodel of one floor of a historic mixed-use building and a major HVAC system overhaul for a 1930 airport terminal building).

The design details for each project were unique. The design and specification work was performed for the various projects by a combination of VRF system distributor personnel, HVAC contractors and mechanical engineers. Five of the eight pilot projects are office occupancies, two are restaurants, and one is a set of four dormitories at a Montana federal government training center. Systems were tested in two climate zones (4 and 5), with the coldest being in Darby, Mont., and the more moderate being in Portland, Ore., and the Puget Sound area.



All but one of the projects used either a VRF/VRV system[3] or a multi-zone DHP system for heating and cooling. Because of federal requirements to purchase American equipment, the Darby, Mont., project used multiple conventional split-system heat pumps for space conditioning. The other Montana project left in place an electric boiler to provide back-up for the heat pump system during the coldest hours. The multi-zone DHP systems, which are very appropriate for smaller projects, can significantly lower costs relative to a VRF installation. The smaller systems are also inherently more efficient.

The eight projects used the separate control systems that come with the Ventacity HRVs (including those enabling scheduling, economizing, and demand control ventilation), combined with the controls that came with the respective VRF/VRV or DHP systems. In most projects, this HRV/VRF (or DHP) combination replaces one or more packaged or split systems of 10 tons or less in capacity—systems that inefficiently combine space conditioning and ventilation air. Ceiling radiant hydronic heating and cooling systems, now standard practice in Europe, would also be an ideal heating and cooling system type for such conversions.

Blower-door tests were incorporated into the pilots projects, as it was unknown how leaky the building envelopes would be. The tests revealed that building leakiness was not a substantial factor.

Pilot participants were recruited on an ad hoc basis. Some sites were found by utility partners and some were brought by contractors or clients that wanted to do a project. Not all of the potential sites assessed ended up in the pilot, for various reasons. Table 1 lists the projects that were completed.

Table 1: Pilot Project Characteristics Summary

Building Type	Location	Project Floor Area (sq. ft.)	Existing System Type	Conversion System Type	Starting/Ending Whole-Building EUIs[4]
Law Office	Portland, OR	11,615	Gas Heat/ Elec Cooling RTUs	VRF	**56.3 / 19.1**
Pizza Restaurant	Corvallis, OR	1,360	Gas Heat/ Elec Cooling RTUs	Multi-zone DHPs	**1,515 / 1,352**
Government District Office[5]	Corvallis, OR	3,770 of 13,200	Gas Heat/ Elec Cooling RTUs	Multi-zone DHPs	**49.2 / 43.4[6]**
Utility District Office	Libby, MT	5,735	Elec Boilers + HP RTU	Multi-zone DHPs	**102.2 / 70.0**
Airport Terminal	Seattle, WA	25,200	Dual-Duct Gas Heat/Elec Cooling	VRF	**152.4 / 48.1**
Government Dormitories (4)	Darby, MT	~11,000, each	Elec Res. Forced Air	Split System HP	**102.8 / 51.5**
Engineering Office	Seattle, WA	5,911	Elect Res. RTU w/ Elec Res. Re-heat	VRF	**51.5 / 29.7**
Restaurant	Portland, OR	1,147	Gas/Elec RTU	Multi-zone DHP	**924.2 / 701.0**

[3]Daikin calls their system a VRV, or Variable Refrigerant Volume system, but it's the same as a VRF system.

[4]Starting Actual EUIs are modeled numbers from RDH's analysis based on actual original equipment in a Typical Meteorological Year (TMY) and calibrated with pre-conversion utility data.

[5]This project converted two of five zones (3,770 sq. ft. of 13,200 sq. ft.). The other zones were converted in a later project, based on the excellent results of the pilot.

[6]The ending EUI can be extrapolated to 28.1 kBtu/sq. ft./year if the whole-building HVAC system were converted assuming similar energy reduction.

Key Pilot Project Activities

- Accomplish system conversion in several buildings of a few occupancy types (office, retail, school, restaurant) to validate the concept of very high efficiency DOAS. It turned out that offices were the majority of the projects (five of eight) and there were no retail or school projects.
- Determine building HVAC loads, conduct billing analyses, and sub-meter HVAC end-uses to quantify HVAC and non-HVAC energy use and demand.
- Monitor indoor temperature and CO_2 levels to document system comfort and indoor-air quality performance.
- Model each project and calibrate the models with field metering and energy bill data.
- Model a consistent base case to compare against conversion system energy use and demand. The base case is a replacement of existing systems with the latest simple code-minimum version of the same.
- Blower-door test each project space to determine air leakage rates. Use results to eliminate an assumption in the models and begin to collect data on smaller commercial building air leakage rates.
- Collect as much pre-conversion HVAC system energy use at the component level, and as much indoor temperature and CO_2 data as possible before the conversions take place. Six months of pre-conversion data was preferred, backed up by at least a year of pre-conversion energy bill data.
- Follow the supply chain analysis, design, proposal and decision-making processes to understand how such conversions might take place in the absence of program specifications and guidance.
- Document lessons learned as project team members use the project specifications and guidelines to accomplish system conversions.
- Document permitting, installation, and start-up issues.
- Document system installed costs and gather information on alternative system costs.
- Commission new systems and verify performance.
- Collect at least 13 months of post-conversion HVAC and whole-building energy use data, and indoor-air quality and temperature data.
- Model the pre- and post-conversion systems and estimate energy and demand savings.
- Gather feedback from building occupants and project owners regarding their satisfaction with the performance of the new systems.

PERFORMANCE

The following tables and figures summarize the performance of the eight pilot projects. There are a few preliminary notes to keep in mind as you review the performance data:

- There results vary significantly by occupancy type.
- There is a wide range of baseloads (i.e., non-HVAC-related energy use). In the office occupancies, these range from 6.8 Btu/sq. ft./year in a very lightly loaded law firm, to 34.1 Btu/sq. ft./year for the airport terminal building (due to the airport runway lighting controls in the basement.) In several cases, the baseload EUI is far larger than the final HVAC EUI (see Figure 2 below). This is the reason both whole-building and HVAC EUI results are presented. While there is significant variance in baseloads and starting actual EUIs, the final HVAC EUIs don't have much variance for the office buildings.
- Savings is largely determined by how inefficient the building was to begin with. In the office buildings, start with the whole-building EUI, subtract the baseload, and subtract the estimated final HVAC EUI (about 11 kBtu/sq. ft., plus or minus 3). The result will be a reasonable savings estimate. If the final HVAC EUI is off by 25 percent (14 instead of 11), the savings estimate will be off by a much smaller percentage (e.g., 6-7 percent), and therefore negligible. Table 2 has the final HVAC EUI (combined gas and electric) results for the office buildings highlighted in red. They range in Climate Zone 4 from a low of 7.6 to a high of 13.3. The utility office in Montana's Climate Zone 6, with the existing large electric resistance boiler left in place for back-up heating, was substantially higher at 37.0.
- There are three cases analyzed and documented for each building in the project: starting actual (as found), code minimum (the alternative replacement equipment option of the latest version of what's already there), and post-conversion (as installed). All were modeled to Typical Meteorological Year (TMY) conditions in EnergyPlus, and calibrated with detailed field monitoring data.
- Electric demand impacts were recorded, modeled and analyzed, but only a sample is provided here. Each project produced its own set of demand impacts, largely based on the electric demand behavior of the system in place, as found. The conversion systems all produced a much less variable seasonal demand pattern, and a generally lower level of demand. In the cases where winter electric demand increased, the increase was modest, and was the result of relatively lower pre-conversion fan power and the addition of heat pump heating.

Energy Performance

The modeling results for the energy performance of the three cases for each of the eight completed projects are shown in Table 2. All of the models were calibrated with at least a year of detailed post-conversion field monitoring of the installed systems and at least a few months of pre-conversion monitoring, along with at least a year's worth of pre- and post-conversion billing data in all but two projects. A few projects did not have a year of pre-conversion billing data: the law office, where the space was vacant for more than 2 years prior to its conversion; the government dormitory, where the energy use for individual buildings could not be separated from the energy bills for the whole facility; and the Seattle office, where multiple spaces in the building were on the same meter, and the conversion space had been vacant for some time prior to the major tenant improvement project beginning.

Table 2: Energy Performance Results (EUIs)

Project	Climate Zone	Project Floor Area (sq. ft.)	Baseload EUI (Btu/sq. Ft.)	Pre-Conversion EUI (Btu/sq. ft.)		Code Minimum Equivalent EUI (Btu/sq. ft.)		Post-Conversion EUI (Btu/sq. ft.)	
				Bldg.	HVAC	Bldg.	HVAC	Bldg.	HVAC
Law Office	4	11,615	6.8	56.3	49.5	51.4	44.6	**19.1**	**12.3**
Pizza Restaurant	4	1,360	1,193	1,515	322	1,470	277	**1,352**	**159**
Government District Office[7]	4	13,200	20.7	49.2	28.5	45.9	25.2	**28.1**	**7.4**
Utility District Office	6	5,735	35.7	102.2	66.5	98.0	62.3	**70.0**	**34.3**
Airport Terminal Building	4	25,200	34.7	152.4	117.7	122.0	87.3	**48.1**	**13.3**
Government Dormitories (4)	6	~11,000, each building	36.2	102.8	66.6	67.9	31.6	**51.5**	**15.3**
Engineering Office	4	5,911	20.1	51.5	31.4	51.3	31.2	**29.7**	**9.6**
Restaurant	4	1,147	635.7	924.2	288.5	874.5	238.8	**701**	**65.3**

There were some consistent outcomes when isolating the HVAC energy use—the portion addressed by the HVAC system conversion. The end-point HVAC EUI results were quite consistent, with the HVAC energy percentage saved often surpassing two-thirds (66%).

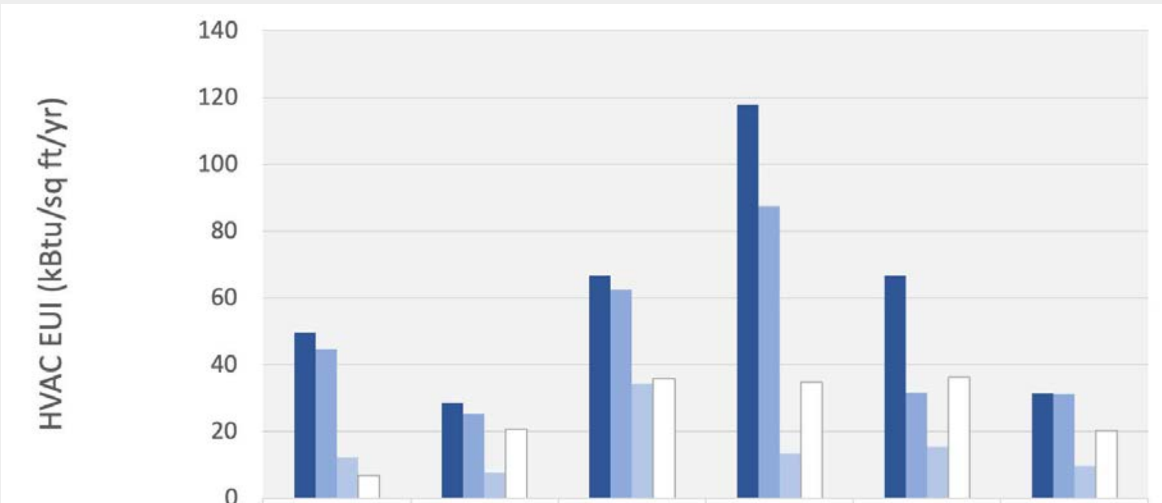
Table 3: Percent HVAC Savings for Each of the Three Modeled Cases

Project	Pre-Conversion HVAC EUI (Btu/sq. ft.)	Code Minimum HVAC EUI (Btu/sq. ft.)	Post-Conversion HVAC EUI (Btu/sq. ft.)	Pre-Conversion to Code-Min. HVAC Savings	Code to Post-Conversion HVAC Savings	Pre-Conversion to Post-Conversion HVAC Savings
Law Office	49.5	44.6	**12.3**	10%	72%	**75%**
Pizza Restaurant	322	277	**159**	14%	43%	**51%**
Government District Office[8]	28.5	25.2	**7.4**	12%	71%	**74%**
Utility District Office	66.5	62.3	**34.3**	6%	45%	**48%**
Airport Terminal Building	117.7	87.3	**13.3**	26%	85%	**89%**
Government Dormitories (4)	66.6	31.6	**15.3**	53%	52%	**77%**
Engineering Office	31.4	31.2	**9.6**	1%	69%	**69%**
Restaurant	289	239	**65**	17%	73%	**77%**

[7,8] This project converted two of five zones (3,770 sq. ft. of 13,200 sq. ft.). The numbers here are for the whole-building model (all five zones converted) in order to make this project comparable to the other office projects. The other three zones were converted in a later project, due to the excellent results of the pilot and the analysis results presented here.

Figure 2 shows the HVAC EUI results for the three modeled cases (starting actual, code minimum, and post-conversion). Baseloads are shown for comparison. Note that in four of the six cases, HVAC EUIs end up well below baseloads after conversion. Typical offices (like the government and Seattle projects) have baseloads of about 20 kBtu/sq. ft., and will have post-conversion HVAC EUIs of 10-12 kBtu/sq. ft. in Climate Zone 4. Added together, an average whole-building EUI of around 30 kBtu/sq. ft. will result from the kind of system conversions conducted in the office pilots, if best practices are followed.

Figure 2: EUIs for Baseload and Three Modeled Cases in the Office and Dormitory Occupancies



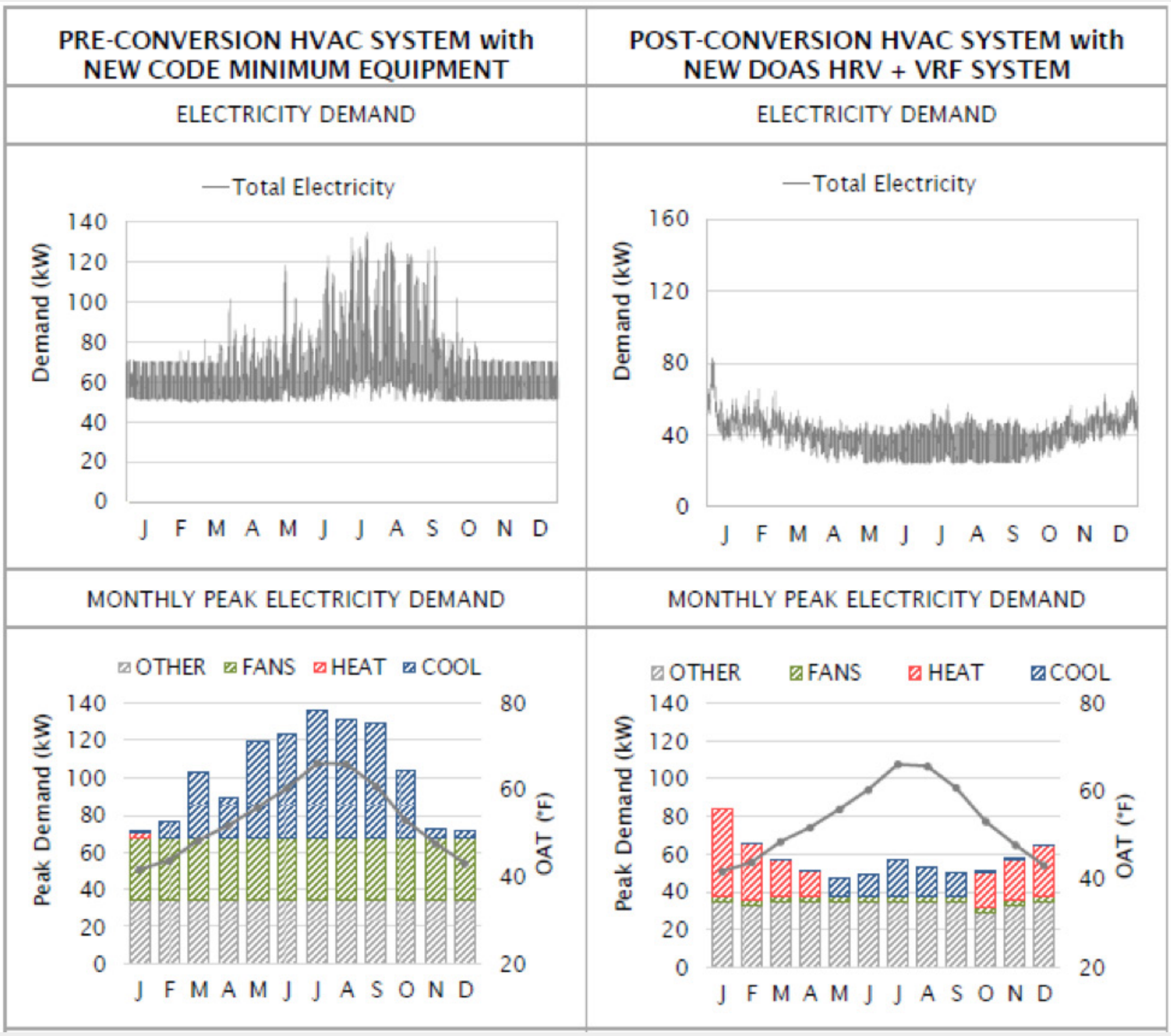
	Law Office	Government District Office	Utility District Office	Airport Terminal Building	Government Dormitories	Engineering Office
Pre-Conversion	49.5	28.5	66.5	117.7	66.6	31.4
Code-Minimum	44.6	25.2	62.3	87.3	31.6	31.2
Post-Conversion	12.3	7.6	34.3	13.3	15.3	9.6
Non-HVAC (Baseload)	6.8	20.7	35.7	34.7	36.2	20.1

The large code minimum savings in the airport terminal building are due to the elimination of the very inefficient existing dual-deck RTU systems that resulted in a significant amount of simultaneous heating and cooling during many hours of the year. In the government dormitory project, the large code-minimum savings come from the elimination of the existing electric-resistance heating in the starting actual case, substituting minimum efficiency heat pump systems for the code-minimum case.

Significant savings came from the near-elimination of the conditioning loads for ventilation air. Additional savings came from the improved efficiency of the heating and cooling systems, and the downsizing of system capacities, as smaller systems tend to be inherently more efficient when properly sized.

During the course of the project there were a number of conversions of the heating function from a natural gas-fired system to an electric heat pump system without significant increase in winter electric demand. This is especially true for the least efficient as-found systems, where inefficient and oversized fans and/or large amounts of simultaneous heating and cooling cause significant wintertime electricity demand and energy use. A typical example is shown in Figure 3, where the demand reductions are clear. Note that the scales are slightly different in the upper graphs.

Figure 3: Demand Reductions in Airport Terminal Building Project



In the office occupancies (five of the eight projects), cooling demand during the peak summer cooling months (July and August) can most often be reduced by a third or more. Actual outcomes ranged from 13 to 59 percent, but three of the five fell into the 25 to 40 percent range. Demand reduction outcomes are detailed in the individual case studies for each project below.

LESSONS

While not all lessons applied to all projects, many projects shared key findings. Table 4 provides the background project information that underlies some of the lessons discussed here.

Table 4: Key Project Metrics

Project	Floor Area (sq. ft.)	Installed System Cooling Capacity (tons)	Conditioned Floor Area / Ton Cooling (sq. ft. / ton)	Number of System Zones	Conditioned Floor Area per Zone (sq. ft. / zone)	Project Cost (per sq. ft.)
Law Office	11,615	16	**726**	8	**1,452**	$15.61
Pizza Restaurant	1,360	11	**157**	4	**433**	$27.50
Government District Office[9]	3,770	4	**943**	2	**1,885**	$11.47
Utility District Office	5,735	9	**710**	8	**637**	$21.90
Airport Terminal Building	25,200	32	**788**	37	**681**	$36.85
Government Dormitories (4)	~11,000, (each building)	16	**688**	5	**2,200**	$9.64
Engineering Office	5,911	14	**422**	12	**493**	$16.83
Restaurant	1,147	3	**382**	3	**382**	$30.99

The lessons learned include the following:

- The HVAC market relies on selling equipment, which leads some contractors and distributors to have a tendency to oversize, or unnecessarily complicate, HVAC systems. But since engineering fees are typically based on the HVAC budget, this system oversizing and/or added system complexity can lead to higher project costs than that of an optimally designed and specified system. Large numbers of indoor units (zones) also results in much higher maintenance costs on the owners or occupants of the building. For instance, every VRF/VRV indoor unit has a condensate pump (which often fails), and a relatively expensive filter that has to be cleaned or changed at least twice a year.

- One project metric that is an indicator of oversized equipment and too many heating/ cooling zones is the "conditioned area per zone" metric in Table 4. A typical 4-ton, single-zone RTU, using the longtime industry standard of 400 sq. ft. per ton of cooling capacity, will serve about 1,600 sq. ft. of office floor area. For project purposes, we regarded proposal values lower than 1,000 sq. ft. per zone as an indicator that the system likely has too many indoor units, and more first costs and maintenance costs than necessary. This

[9]This project converted two of five zones (3,770 sq. ft. of 13,200 sq. ft.). The numbers here are for the whole-building model (all five zones converted) in order to make this project comparable to the other office projects. The other three zones were converted in a later project, due to the excellent results of the pilot.

doesn't apply to the restaurants where loads per square foot is often double or more that for office or retail occupancies. In restaurant systems, roughly double the capacity needed per square foot yields half the per-zone floor area that one would expect for an office.

- While system design is crucial for optimal performance, contractors and engineers have little experience designing with this equipment, especially the very high efficiency HRV.

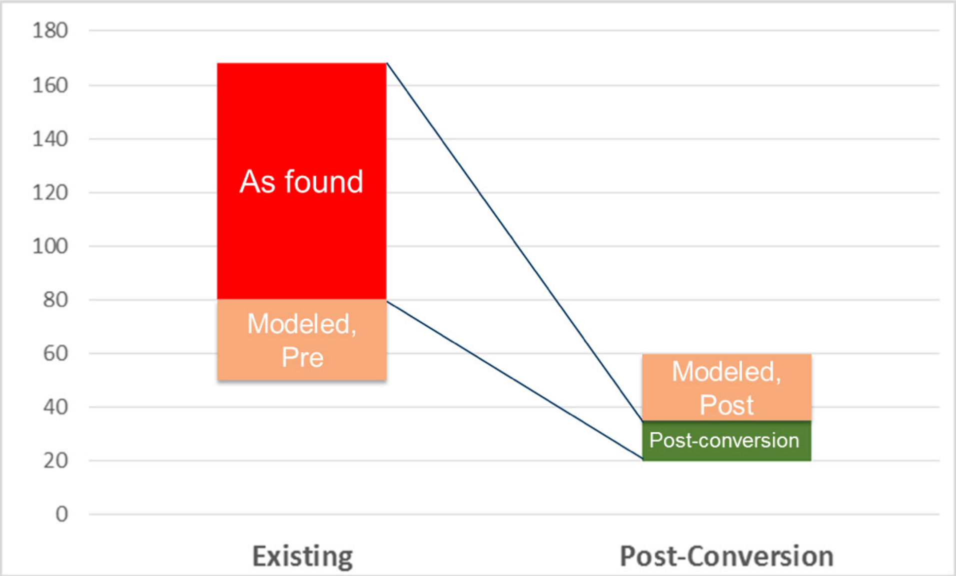
- Many typical guidelines do not apply with this equipment. For example, in older buildings in Climate Zone 4, proper conversion system sizing is greater than 600 sq. ft. of conditioned area per ton of cooling system capacity—not the typical 300-400 sq. ft. per ton. Table 4 shows the floor area served by each ton of cooling system capacity in the conversion system for each project. Only two of the pilot projects (the government office and Seattle office) had system sizing fall outside the project guidelines. These were designed with the industry standard 400 sq. ft./ton of capacity. As both are in Climate Zone 4, they have a considerable amount of excess capacity (and some additional cost) built into the systems.

  The other major design challenge for contractors and engineers is the proper design of ventilation systems using highly capable and efficient HRV or ERV technology. Most engineers and contractors are unfamiliar with packaged HRV or ERV systems that perform the cooling season economizing function without any connection to the heating and cooling system or other building controls. The unit must be sized properly if economizing is to be carried out effectively. This generally means sizing the ventilation capacity for any given zone or zones being served by a unit so that the ASHRAE 62.1 airflows are between 40 and 60 percent of the full-rated flow of the unit (i.e., a 1,000 cfm unit would require 400-600 cfm for maximum ventilation airflow requirements). If this is done properly, the performance of the HRV or ERV will increase the number of cooling season hours per day that the cooling system is off—a large source of cooling energy savings. The excess HRV or ERV flow capacity also allows for boosting airflows when a zone experiences higher occupancy levels than expected.

  In addition, ventilation air must be delivered to one side of the space being ventilated and exhausted from the other side, so the occupants in the middle receive the benefits of the fresh air, and no mixing of the air is required. In all but the most extreme climates, the ventilation air does not need to be conditioned, even on the coldest winter days.

- Energy models don't yet accurately model the impacts of such system conversions, either in the base case or the conversion case. Sometimes this is because the individual system performance data in the models, for both base case and high performance systems, is based on test and rating methods that do not accurately reflect field performance. In most cases, the performance of the systems (especially the HRV system) can't be accurately represented in even the best of the models due to the form of inputs required or the lack of inputs altogether. The ranges of typical modeling outcomes are shown in Figure 4 below.

Figure 4: Typical Modeling Outcomes



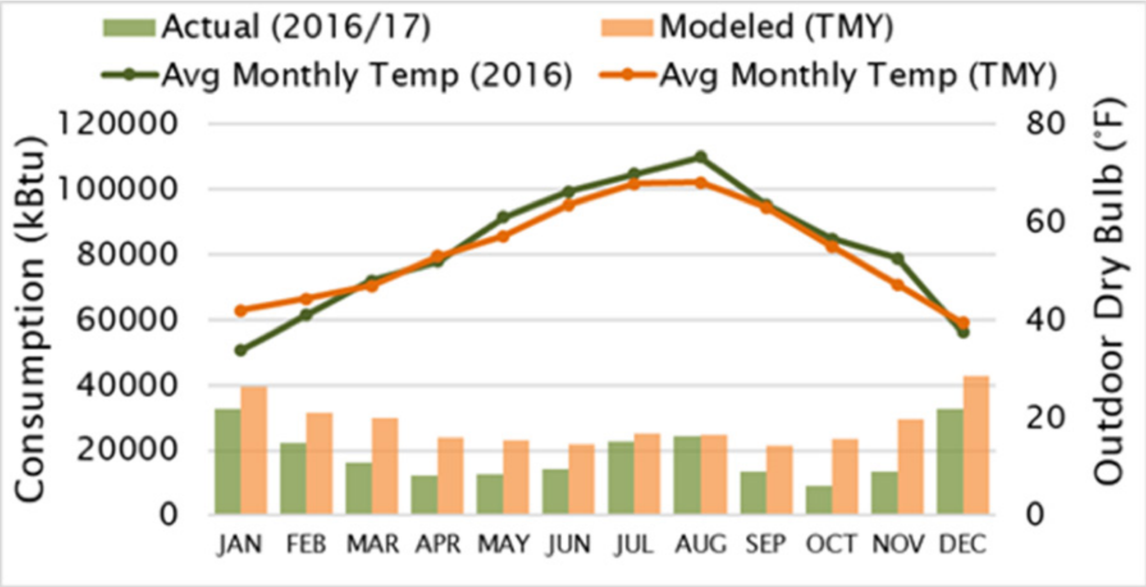
Packaged HRVs and ERVs have their own characteristic (and non-linear) performance curves based on flow rates and external static pressures. Flow rates in sophisticated ventilation systems, especially those using demand-based controls programming, are highly variable, on an hourly basis, for each building. The power used by the HRV model (1,000 cfm nominal maximum flow rate) is, on average, over a day of operation, lower than expected (about 0.1 W/cfm, or 10 cfm/W), which is needed for the simplistic inputs that most energy models allow. This radical simplification compared to the actual operating characteristics of the system may or may not deliver a reasonable estimate of HRV and ERV performance.

According to many modelers, VRF modeling in Energy Plus has improved of late, but there are still deficiencies. Some of these deficiencies stem from uncertain real-performance curves and refrigerant piping loss assumptions that are not based on any field validation. The more zones, the larger the losses and the larger the error in loss estimates. There is also the matter of sizing, which most models assign based on incorrect assumptions. Absent strong guidance for designers and specifiers, most systems will be significantly oversized. For instance, VRF distributors, who design most systems, will usually have many more zones than necessary. Whole-system behavior under low-load conditions is not reflected in the performance curves used in the models.

Temperature setback, assumed by the models to save energy, may actually not save any energy at all. This is due to the way variable capacity heat pump systems handle large discrepancies between space temperature and setpoint. Because of the non-linear efficiency curve of the systems (fan laws are exponential), it is most often more efficient to leave setpoints alone than to prompt operation in high power modes by varying the setpoint by more than a degree or two.

Figure 5 shows how wrong a good energy model can be when attempting to model the conversion system in the law office project. The models consistently over-predict actual energy use, sometimes by a large percentage.

Figure 5: Early Energy Modeling Results for the Law Office Project



While some important conversion system characteristics cannot be well represented in the models, they still impact system energy use. With good field data for calibration, and with the right adjustments, models can become more representative. In the end, modeling improved as knowledge was gained over the course of the projects. This was important in creating valid comparisons among the three cases modeled (starting actual, code minimum, post-conversion).

- In most buildings, the heat recovery option (simultaneous heating and cooling) for VRF/VRV systems will seldom save enough energy to justify its substantial cost. If the option is included, the zoning must be specified in a way that allows excess energy in one zone to be utilized in another. While some building layouts and orientations may lend themselves to this type of strategy, we did not see this analysis as a part of the decision-making process in the several pilot project buildings that have the heat recovery option.

- Many existing buildings that use packaged HVAC systems (such as RTUs) aren't ventilated as designed. Heating and cooling ventilation air can be expensive, so building occupants or owners often take steps to minimize outside airflow into the system. In many cases this means locking or taping the outside air dampers shut. In these instances, savings would come from a very high efficiency conversion system eliminating almost all of the conditioning of the ventilation air that will not materialize.

- Project costs in the pilot were typically under $25/sq. ft. for office projects. However, this isn't how most projects were initially proposed. Several were proposed at $25–35/sq. ft. In one case, where a cost breakdown was requested, the proposal contained $5/sq. ft. for unspecified controls, as requested by the controls company or distributor. The proposed controls costs encountered during the course of the pilot consistently ran between $4.50 and $7/sq. ft. The control systems described in the section immediately above virtually eliminate this cost.

All four of the projects in Table 4 with costs above $20/sq. ft. had circumstances that explained the high costs, except for the airport terminal building project. Despite the ventilation system being under-capacity, this system had the highest cost per square foot, by far. Six engineering studies for other building projects were reviewed during the course of the project and shortly after. All six, by five different engineering firms, priced the standard three types of HVAC systems—an upgraded RTU-based system, VRF, and VAV—as follows: $25/sq. ft. for the RTU-based system, $35/sq. ft. for VRF, and $40/sq. ft. for a VAV system. The buildings were of varying ages, of varying sizes (9,000 sq. ft. to 80,000 sq. ft.), but all were offices.

The pilot project costs demonstrate a potential for building owners to save a lot of money if the right specifications and design guidelines are followed, and the right controls are used.

- HRV and ERV system efficiency is very sensitive to the characteristics of the ventilation duct system. Elevated external static pressures, on either side of the unit, will notably increase fan power, especially at higher airflow rates. Duct leakage, especially if on the exhaust intake side of the unit from unconditioned spaces, can seriously impact the temperature of the exhaust air entering the unit. In a very efficient HRV or ERV, this impacts the temperature of the fresh air stream on the other side of the heat exchanger. During the heating season, in addition to increasing the heating load, fresh air temperatures may be reduced enough to affect thermal comfort conditions. During the cooling season, both the sensible and latent cooling load will increase and reduce comfort conditions in the space.

  This is especially a problem for ducting above the roof, where both duct leakage and insufficient insulation can combine to seriously lower system efficiency and thermal comfort while increasing space conditioning loads. In conventional HVAC systems, where ventilation air and space conditioning air are combined, the heating and cooling components of the systems are sized with enough capacity to overcome the thermal losses of the ducting and maintain delivered air temperatures within the comfort range. While the ventilation system can be designed to maintain comfort, such system requirements can add significant cost and impose significant system efficiency penalties. In general, outdoor ducting should be avoided whenever possible. In most projects this will not be an issue, but where outdoor ducting must be used, attention must be paid to the airtightness of the ducting, and added insulation for ventilation ducting is strongly advised.

- Restaurants introduce unique challenges. Most restaurants have one or more powerful range hood systems serving the kitchen. These systems and their make-up air units dominate the airflow and building pressure for this type of occupancy. This requires special attention to the interaction of the balanced ventilation system that serves the dining area and the exhaust vent system that serves the kitchen. In order to maximize the effectiveness of the ventilation system, the balance between the vent hood and make-up air fans must be carefully calibrated, leaving the kitchen slightly depressurized relative to the dining area. As difficult as this can be to accomplish, the Portland restaurant pilot project did this very well and the results were exemplary.

  The guidelines for space conditioning system design and sizing are also necessarily different for restaurants than for other occupancies, as the peak loads per square foot are inherently larger.

CONCLUSION

The pilots prove that, if project specifications and guidelines are followed, it is neither difficult nor expensive to dramatically reduce energy use and moderately reduce electric demand in commercial buildings—while also improving indoor conditions for the building occupants.

On average, participants saw a 70% reduction of actual HVAC energy use and a 42% reduction in actual whole-building energy use. Even if these pilot buildings had started with standard code-minimum equipment prior to the conversion, modeling still shows significant average energy savings of 65% for HVAC and 36% for the entire building.

All of the work done in the pilot buildings will be much easier and less expensive to accomplish in new buildings, of any size, and in larger existing commercial buildings.

System simplification turned out to be the primary tool for producing the best results. Today's conventional HVAC systems tend to be inefficient, overly complex, relatively expensive, not very reliable, and poorly controlled. Standard direct digital controls are typically too complex for these systems, and, in general, unaffordable for smaller buildings. However, recent developments have substantially raised the bar for control system performance and affordability.

APPENDIX: INDIVIDUAL PILOT PROJECT REPORTS

BETTERBRICKS
Powerful Energy Ideas. Delivered by NEEA.

PILOT REPORT

UTILITY DISTRICT OFFICE

LIBBY, MT

Background

This single-story 1960s-vintage building features offices at the front and a combination of storage space and four garage bays for utility-line maintenance trucks at the back. The Flathead Electric's building facility team partnered with Bonneville Power Administration (BPA) and Northwest Energy Efficiency Alliance (NEEA) to determine and implement a cost-effective and energy-efficient HVAC conversion to increase comfort and reduce energy use.

This project was one in a series of pilots across the Northwest to help the region better understand the design, installation and expected energy savings of an advanced HVAC solution for small-to-medium commercial buildings. The system approach includes dedicated ventilation air (decoupled from primary heating and cooling air) with high-efficiency heat recovery, a high-efficiency heating and cooling system, and key design principles.

PRE-CONVERSION DETAILS

Wall and Window Assembly

The walls are constructed of a combination of 8- and 12-inch CMU block with an insulated nominal 2-inch strapped interior wall assembly for electrical and plumbing. The wall between the storage/garage area and offices is fully insulated and framed with 2x4s. The windows have a combination of single- and double-pane glazing units, none of which have a low-e surface.

Conversion Challenges

This conversion project featured a variety of challenges including the differing uses of the building spaces, the cold climate, the sometimes irregular operating hours, and the uniquely assembled existing-system.

OVERVIEW

Gross Floor Area
5,735 sq. ft.

Starting / Ending EUI
98 / 70 (Building)
62.3 / 34.3 (HVAC)

Typical Operating Hours
M-F, 7 a.m. to 5 p.m.; with some after hours

Setpoint/Setback in Heating Season
72° F / 65° F

Setpoint/Setback in Cooling Season
73° F / 78° F

Envelope Thermal Characteristics
Windows: **U-0.5 office, U-0.9 storage/garage**
Ext. Walls: **R-10 office, R-7 storage/garage**
Roof: **R-8 office, R-35 storage/garage**

Energy Utility
Flathead Electric in partnership with the Bonneville Power Administration

This conversion project featured a variety of challenges including the differing uses of the building spaces, the cold climate, the sometimes irregular operating hours, and the uniquely assembled existing-system.

The existing HVAC system, focused mostly on heating, consisted of a 160-kW, two-circuit electric boiler serving hydronic unit heaters in the storage/garage space, and wall-mounted perimeter radiator units in the office spaces. To the project team's knowledge, there is no zone isolation in the boiler hydronic heating loops, so heating in one space impacts the temperatures in other spaces.

Additionally, there was a 6-ton heat pump rooftop unit (RTU) that provided cooling to the office spaces in the summer season. This unit provided the only ventilation air to the office spaces. Two swamp coolers were used for cooling in the storage/garage space, and served as those spaces' only source of ventilated air. There was also a .75-ton ductless heat pump unit that provided conditioning to a small computer server room.

There is a single thermostat in the office area that controls the perimeter hydronic baseboard units, and a single thermostat outside the storage/garage space that controls the unit heaters in that space. Both are operated manually by the building occupants. Prior to conversion,

Pre-Conversion Air Leakage

The building was blower door-tested for air leakage prior to system conversion using the U.S. Army Corps of Engineers test protocols. The most significant air leakage pathways were through exhaust fan and swamp cooler duct openings, the overhead doors in the truck bays, and a location where a power pole penetrated the roof assembly.

Air Leakage Test Results – Pre-Conversion

Test Condition	Depressurize	Pressurize	Average
Enclosure Airtightness *[cfm/ft^2@75 Pa]*	0.769	0.723	0.746
Equivalent Leakage Area *[ft^2@75Pa]*	11.15	10.48	10.82
Air Changes *[per hour@50 Pa]*	6.31	6.07	6.19
Air Leakage Text Coefficient (C) *[cfm/Pa^n]*	1080.0	1312.0	N/A
Flow Exponent (n) *[dimensionless]*	0.609	0.549	N/A
Squared Correlation Coefficient (r^2) *[dimensionless]*	0.999	0.999	N/A

SYSTEM CONVERSION DETAILS

Multi-Zone Heat-Pump Systems

The conversion project team removed the heat pump RTU and the swamp coolers in the storage/garage space. They then installed two new multi-zone heat-pump systems, and added a very high efficiency heat recovery ventilator system, while placing the electric boiler-fed hydronic system in a back-up mode to supplement heat-pump capacity under the most extreme ambient conditions. While the hydronic system controls were upgraded, none of the distribution system components were replaced or upgraded.

To replace the existing heat pump RTU in the office space, and to displace a significant portion of the electric boiler load, two Mitsubishi 4.5-ton multi-zone variable capacity ductless heat-pump systems were installed—one to serve the office space and one to serve the storage/garage space. The office space has seven individual indoor units, each with its own wall controller. The open area of the storage/garage space is served by two 2-ton air handlers, both managed by a single wall controller.

Very High Efficiency HRV System

A single Ventacity VS 1000 RT HRV unit was installed to provide year-round ventilation. Ducting in the heat pump RTU office space was modified for ventilation-only use, and new ventilation ducting was added to serve the storage/garage space.

The conversion project added ventilation-zone control dampers to ensure that the ventilation air in the office space was not affected by the ventilation and exhaust movement in the storage/garage space. As exterior ducting heat loss significantly affects the performance of an HRV system, this system characteristic has an adverse impact on the energy performance of the ventilation system overall.

Boiler Controls

The electric boiler and its distribution system were placed in a backup role, with new controls, for periods when ambient conditions approached winter conditions. The new boiler controls include an outdoor reset function, with supply water at 180 F when outdoor temperatures are at -20F, and at 100 F when outdoor temperatures are at 60 F. The controls also include a 15-minute allowance for reaching heating setpoint before activating the backup system.

Ventilation Controls

Ventilation controls were set up to maximize HRV airflow for up to an hour after a garage bay door was opened or closed. As target ventilation rates during business hours are 320 cfm for office spaces and 200 cfm in the storage/garage area, the increased airflow (400 cfm) goes to the storage/garage bay space while the office space airflow remains at 320 cfm.

	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Existing System Type				
Electric Boiler	1) Weil-McLain CEW-80	N/A	546 kBtu/hr	2
Swamp Coolers	(2) Champion 7500 SD	Unknown	N/A	1
Heat Pump RTU	(1) Lennox CHP16-953-3Y	6 tons	6 tons	1
Server Room Heat Pump	(1) Fujitsu AOU9RLFW	9 kBtu/hr	12 kBtu/hr	1

	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Conversion System				
VC Heat Pump Units	(2) Mitsubishi MXZ48-8C	4.5 tons each	4.75 tons each	8
Packaged HRV	(1) Ventacity VS1000RT	1,025 cfm		2

The total installed system cost was $21.90/sq. ft. This includes the $1,200 cost for the ventilation controls. Aside from updated boiler system controls, these were the only controls added to those that came with the heat-pump and HRV systems.

POST-CONVERSION SYSTEM PERFORMANCE

Post-Conversion Air Leakage

The post-conversion results show a notable reduction in building air leakage. Most of this improvement likely came from the removal of the swamp coolers and the integration of the building exhaust requirements into the ventilation system. As expected, air leakage in the storage/garage area remained relatively high, given the four overhead doors that comprise a significant portion of the envelope area for these spaces.

Air Leakage Test Results – Post-Conversion

Test Condition	Depressurize	Pressurize	Average
Enclosure Airtightness *[cfm/ft²@75 Pa]*	0.415	0.314	0.365
Equivalent Leakage Area *[ft²@75Pa]*	6.02	4.65	5.29
Air Changes *[per hour@50 Pa]*	3.28	2.68	2.98
Air Leakage Text Coefficient (C) *[cfm/Paⁿ]*	392.0	674.0	N/A
Flow Exponent (n) *[dimensionless]*	0.700	0.511	N/A
Squared Correlation Coefficient (r^2) *[dimensionless]*	1.000	1.000	N/A

Post-Conversion Energy Performance

Performance data was collected from January 1 through December 31, 2017. As building operation during the analysis year was considered very unusual, the monitoring period was extended into late 2018 and utility data was collected into early 2019.

Pre-Conversion Minimum (As Modeled)	
EUI:	**98** kBtu/sq. ft./yr.
Fans:	**12.8** kBtu/sq. ft./yr.
Heating:	**46.2** kBtu/sq. ft./yr.
Cooling:	**3.3** kBtu/sq. ft./yr.
HVAC:	**62.3** kBtu/sq. ft./yr.

Post-Conversion (As Modeled)		
EUI:	**70.0** kBtu/sq. ft./yr.	29% Reduction
Fans:	**3.2** kBtu/sq. ft./yr.	75% Reduction
Heating (Boiler):	**9.6** kBtu/sq. ft./yr.	38% Reduction
Heating (HP):	**19.2** kBtu/sq. ft./yr.	
Cooling:	**2.3** kBtu/sq. ft./yr.	30% Reduction
HVAC:	**34.3** kBtu/sq. ft./yr.	45% Reduction

Pre-Conversion (Actual Use)	
EUI:	**102.2** kBtu/sq. ft./yr.
HVAC:	**66.5** kBtu/sq. ft./yr.

Post-Conversion (Actual Use)		
EUI:	**70.0** kBtu/sq. ft./yr.	32% Reduction
HVAC:	**34.3** kBtu/sq. ft./yr.	48% Reduction

Pre- and post-conversion modeled energy consumption are based on a typical meteorological year (TMY).
Models updated based on a full year of sub-metered energy end-use data.

Energy-Use Variables and Considerations

Base load in the analysis period was estimated at 31.3 kBtu/sq. ft./year. However, based on utility bill analysis and weather normalization of the HVAC use, base load in 2014 was only about 24 kBtu/sq. ft./year. It rose fairly steadily to an estimate of 34 kBtu/sq. ft./year in 2018. This does not affect the total savings since the models hold baseline constant pre- and post-conversion.

However, if the operation of the ventilation system is based on normal occupancy or demand control, its load can look a lot like the lighting load, with a slight increase in the winter months but otherwise in the same daily pattern as the lighting. The project team could not determine the cause of the apparent base load increase over the 5 years of data examined, but it is significant enough to make the HVAC contribution to whole-building EUI reduction seem smaller than it actually was.

Another large energy use variable is the role of the backup system, the electric boiler, and the various terminal units providing space-heating energy before and after the conversion. To help mitigate this energy use, outdoor reset controls were installed to reduce boiler supply-water temperatures during milder periods.

The model calibrated with the monitoring data, showed that the heat pump systems used 58 percent of the heating energy overall. This was during a year when the boiler was being used for roof snow-melting, so the actual use during the heating season was notably larger than predicted. While additional savings might materialize with further fine-tuning, HVAC savings around 40 percent for this building, in this climate, with this combination of systems, and with this type of use, are substantial.

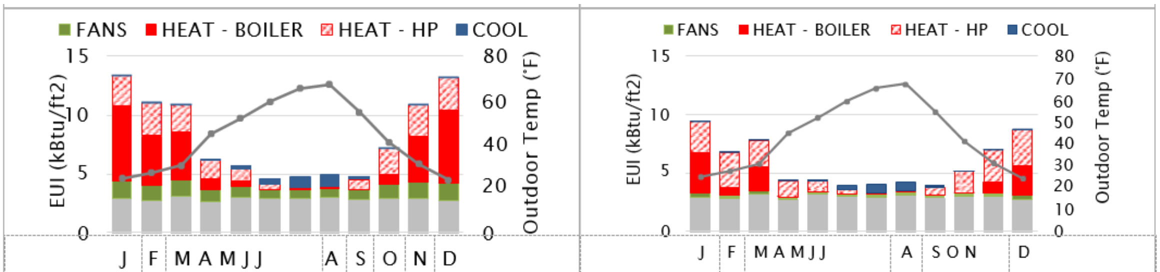
Post-Conversion Operating Practices

The unusual nature of this project makes it difficult to predict total-system performance. As the electric boiler remains installed as the back-up system, overall performance rests substantially on how the building is operated and the interaction between the heat pump and boiler controls. Without zoning of the boiler hydronic system, heat increases in one part of the building will have at least some impact on temperatures in other parts of the building. The thermostat for the storage/garage space operates the fans on the unit heaters, but that level of control doesn't exist for the convective hydronic radiator units in the office space.

These idiosyncrasies became especially apparent in the first few months of the monitoring period (January through April) when weekday temperatures in the office space routinely exceeded 75 F and storage/garage temperatures routinely rose above 70 F. Indoor temperatures in both zones were 3-5 F higher than pre-conversion temperatures.

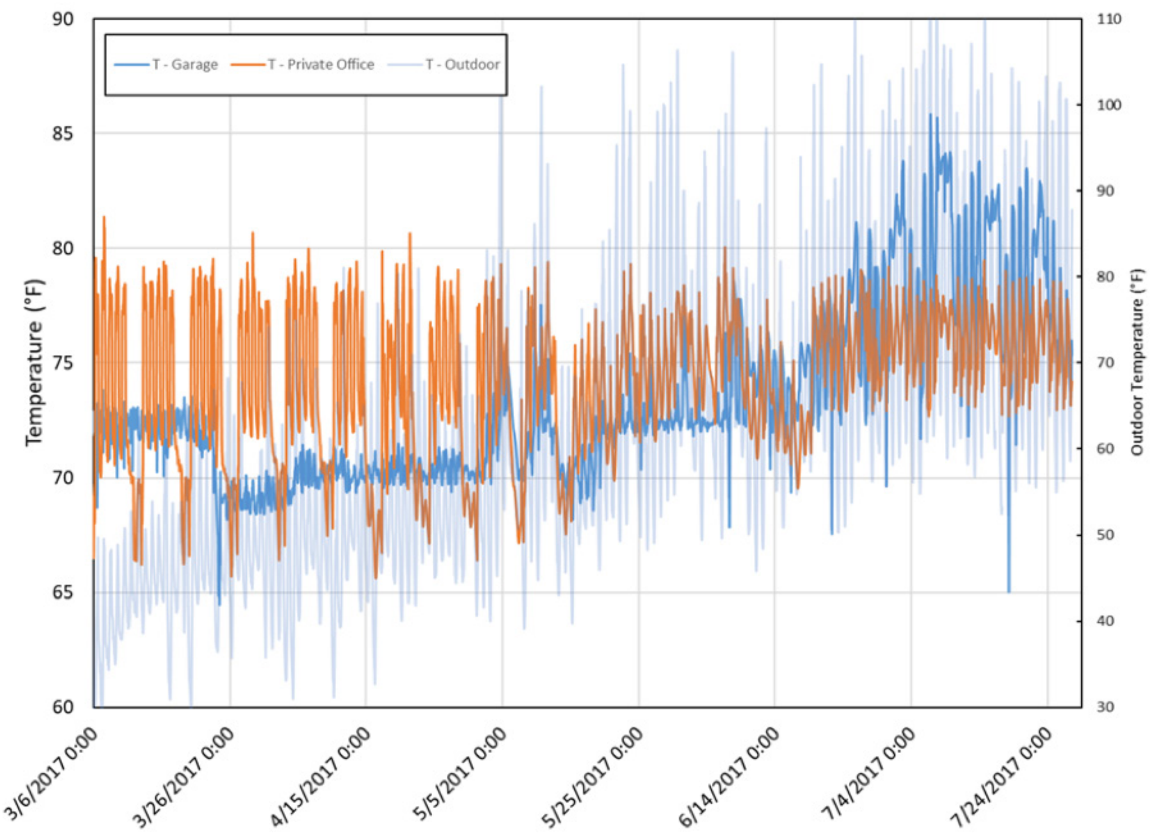
Annual Performance Data

Using the building models calibrated with post-conversion energy bills and system monitoring, the figures below show system annual and monthly energy use data.



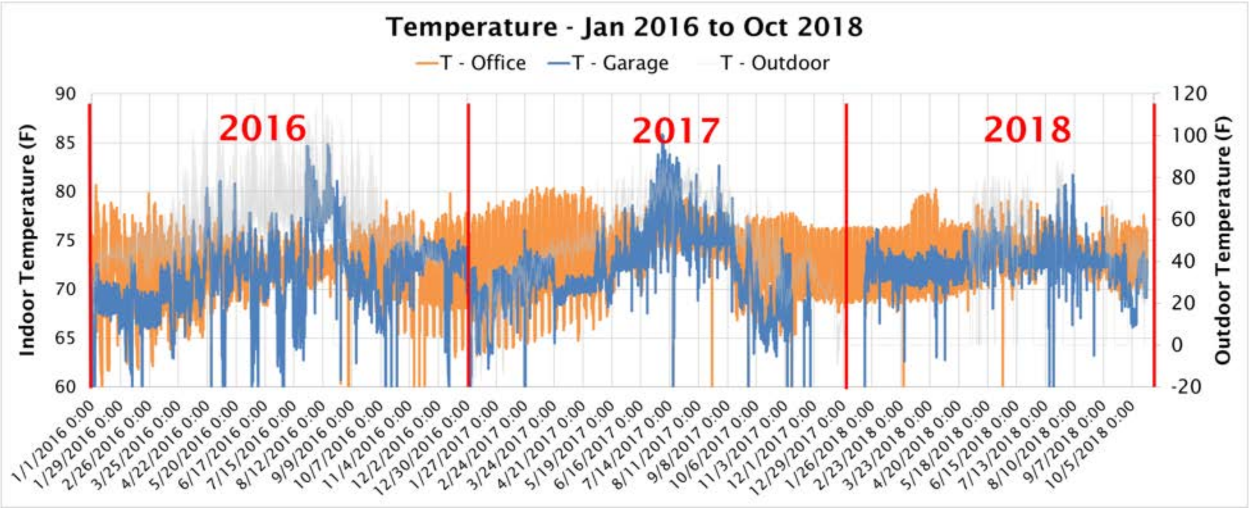
	ANNUAL EUI	---		ANNUAL EUI	ANNUAL SAVINGS
Total[2]**:**	98.0 kBtu/ft²	---	**Total:**	70.0 kBtu/ft²	28.0 kBtu/ft²
Fans:	12.8 kBtu/ft²	---	**Fans:**	3.2 kBtu/ft²	9.6 kBtu/ft²
Heating:	46.2 kBtu/ft²	---	**Heating:**	28.8 kBtu/ft²	17.4 kBtu/ft²
Cooling:	3.3 kBtu/ft²	---	**Cooling:**	2.3 kBtu/ft²	1.0 kBtu/ft²
HVAC:	62.3 kBtu/ft²	---	**HVAC:**	34.3 kBtu/ft²	28.0 kBtu/ft²
Electricity:	98.0 kBtu/ft²	---	**Electricity:**	70.0 kBtu/ft²	28.0 kBtu/ft²

Daily Indoor/Outdoor Temperature (Mar 2017 - July 2017)



When the project team inquired about the reason for the high indoor temperatures, staff expressed concern about the unusually heavy snow load on the roof of the building that year, and reported that they were using the combination of abnormally high indoor temperatures and the relatively poor roof insulation to melt some of the snow.

The graphic below shows the wide range of temperatures recorded and the trends over time, from January 2016 through October 2018. Note that during the analysis period, office space temperatures ranged between 72 and 78 F or more during weekdays and dropped below 72 F only on weekends—even during times when outdoor temperatures were at 40 F or higher. The office temperature range narrowed somewhat during the cooling season, while temperatures in the storage/garage space were maintained above 70 F on most winter days.



The range of temperatures narrowed somewhat in 2018, but office temperatures were still at 75 F or above during winter weekdays, and storage/garage temperatures were kept at 70–73 F through the winter.

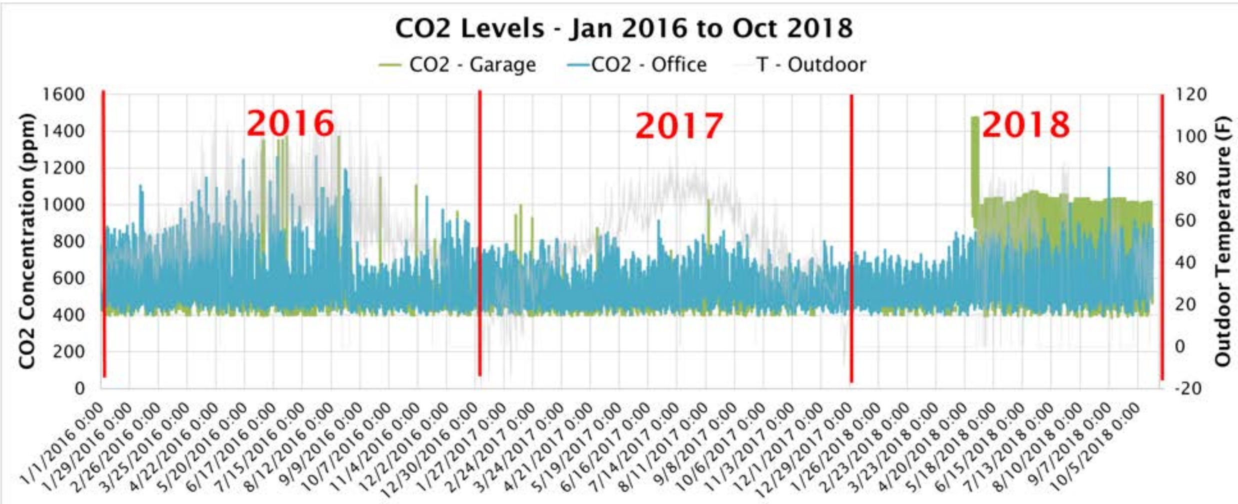
In the fall of 2018, setpoints were specified and the night and weekend temperature setback range was reduced. In general, most variable capacity heat-pump systems do not save energy when large setback or setup temperatures are used on a daily basis. While monitoring ended shortly thereafter, bill-data analysis from early 2019 shows additional savings after these changes.

Post-Conversion Ventilation and Increased CO_2 Levels

More reported operating practices emerged in conversations with utility staff. For instance, it was common during the cooling season, and perhaps also during the milder parts of the heating season, for one or more of the garage bay doors to remain open for extended periods—presumably while the heat pumps were attempting to condition the space and the HRV was running to ventilate the space.

Because of gaps in the data, it's unknown how many hours the system operated under these conditions. However, analysis of the system model and metering data shows the expanded full-flow HRV operation likely contributed to an increase of 30 percent to HRV energy use in 2018 compared to 2017. While this is not a lot of energy use compared to the heating and cooling functions, it is an opportunity for further fine-tuning and energy savings.

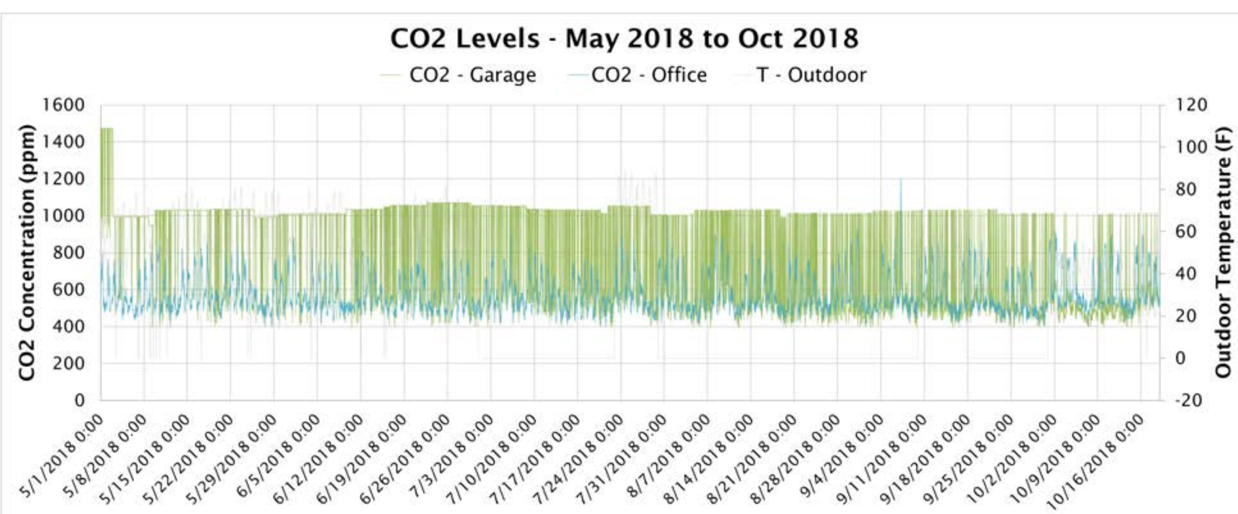
Increased CO_2 levels in the storage/garage area may be another reason for increased ventilation use in May 2018. These changes can be seen clearly in the below graphic, where office CO_2 levels are in blue and storage/garage levels are in green (the scale is in 200 ppm increments with May 2018 CO_2 levels capped at about 1,000 ppm, presumably by the HRV controls, after a brief period at up to 1,500 ppm).



Below is an enlarged view showing data from May–October 2018. These readings are at times much closer to ambient air CO_2 levels, especially at the beginning of October. High CO_2 levels are more prevalent during the summer months and it is possible that lower levels correlate with garage door openings. While the anomalous behavior roughly coincides with the changes from the heating season to the cooling season, and then back to the heating season, the project team has been unable to determine a cause for the sudden increase in the storage/garage area CO_2 levels.

Nevertheless, the significant increase in the energy use of the HRV system suggests that the system was responding to what may have been actual increases in CO_2 levels, or, increases in another in the family of compounds that mimic the effect of CO_2 in some sensors (such as VOCs).

One explanation explored was a correlation with 2018 Montana wildfires. Staff reported significantly poor air quality during this time due to fires in the area.

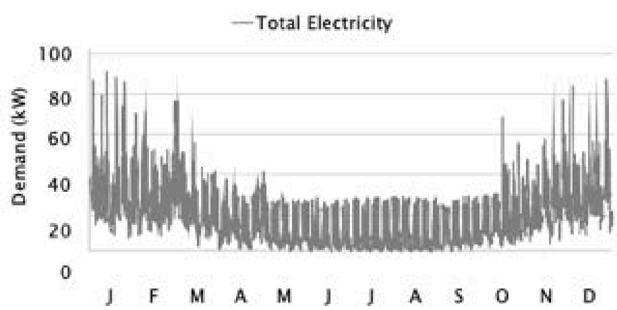


Post-Conversion Demand Impacts

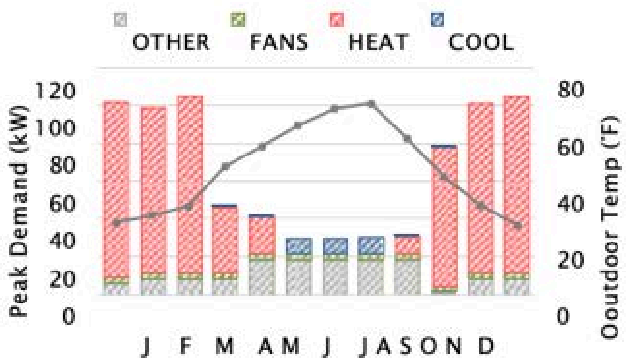
Given the nature of the pre-conversion systems and the high performance of the new systems, the team expected increased electricity demand savings. A primary goal of the project was curtailing the use of the large variable-capacity electric boiler. Heat pump systems that can maintain full-rated capacity down to 5 degrees F contribute to demand reduction, as do the very high ventilation heat-recovery efficiency coupled with very low fan power. The monthly hourly peak reductions look like this for the analysis year:

PRE-CONVERSION HVAC SYSTEM with NEW CODE MINIMUM EQUIPMENT

ELECTRICITY DEMAND



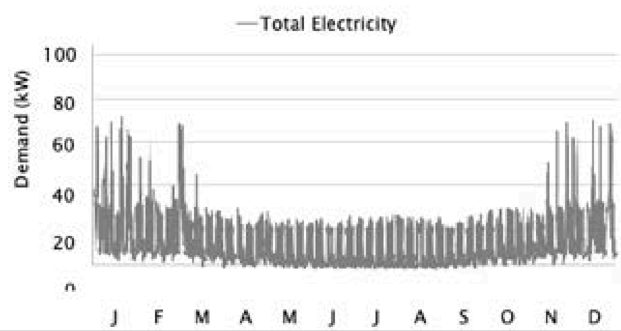
MONTHLY PEAK ELECTRICITY DEMAND



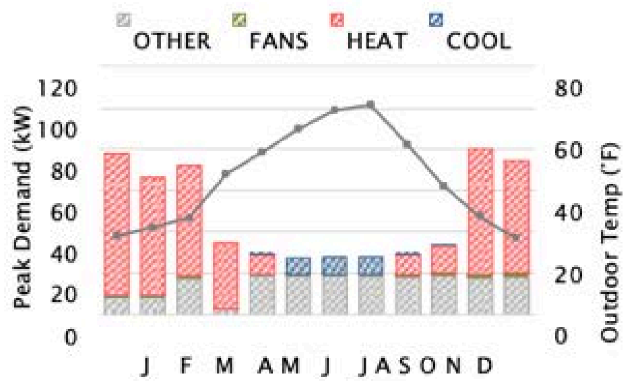
	PEAK DEMAND kW	PEAK OCCURRENCE
JAN	102	13-JAN-07:00
FEB	98	08-FEB-07:00
MAR	104	02-MAR-7:00
APR	46	11-APR-7:00
MAY	42	02-MAY-7:00
JUN	29	28-JUN-15:00
JUL	30	21-JUL-15:00
AUG	30	01-AUG-15:00
SEP	31	28-SEP-7:00
OCT	78	16-OCT-1:00
NOV	101	20-NOV-07:00
DEC	104	20-DEC-07:00

POST-CONVERSION HVAC SYSTEM with NEW DOAS HRV + VRF SYSTEM

ELECTRICITY DEMAND



MONTHLY PEAK ELECTRICITY DEMAND



	PEAK DEMAND kW	PEAK OCCURRENCE
JAN	78 (↓23%)	03-JAN-01:00
FEB	66 (↓32%)	08-FEB-07:00
MAR	72 (↓31%)	01-MAR-10:00
APR	35 (↓24%)	29-APR-6:00
MAY	29 (↓31%)	02-MAY-7:00
JUN	27 (↓9%)	28-JUN-15:00
JUL	27 (↓8%)	21-JUL-15:00
AUG	27 (↓8%)	01-AUG-15:00
SEP	29 (↓6%)	28-SEP-7:00
OCT	33 (↓57%)	16-OCT-1:00
NOV	80 (↓20%)	27-NOV-01:00
DEC	74 (↓29%)	27-DEC-01:00

→ Estimated decrease in monthly peak electricity demand is about 30% in the winter (heating season) and about 8% in the summer (cooling season).

ADDITIONAL FINDINGS

Above-the-roof ductwork can greatly hinder system performance.

Most engineers and designers are accustomed to specifying ducting for a packaged RTU, most with natural-gas heating. The ductwork on this project, however, is located above the roof, in rural Montana's outside ambient air conditions.

Typically, the loss of some energy in the supply and return ducting is mitigated with additional heating capacity that keeps delivered air temperatures warm. This reduces the impact on performance and comfort.

However, similar ducting energy losses can significantly impact DOAS ventilation system performance. The loss of 6–8 degrees F of delivered air temperature can reduce winter delivered air temperatures below levels recommended by ASHRAE Standard 55, and reduce heat exchange efficiency. If exhaust air is also conveyed through above-the-roof ducting on its way to the HRV or ERV, then the effect is doubled due to the reduction in available outgoing energy at the heat exchanger. There is a similar effect in the summer during the cooling season, but the impact is lower due to the reduced temperature differentials involved.

The team learned two primary lessons on this topic. First, conversion projects should avoid or minimize above-the-roof ducting whenever possible. Second, the condition and installation quality of the ducting is crucial to system performance, especially for a very high performance ventilation system.

Regular and effective communication ensures project success.

The engineer for the project considered all aspects of post-conversion system operation and carefully specified each component and control element. The engineer and contractor worked closely together to make sure the project met design requirements and specifications. The contractor was conscientious and did excellent work on site, and provided helpful input during the design and specification stage.

It's clear that having an experienced, supportive, thoughtful project team goes a long way toward ensuring project success. Good communication was key to meeting plans and specifications, and to quickly and effectively handling unexpected problems.

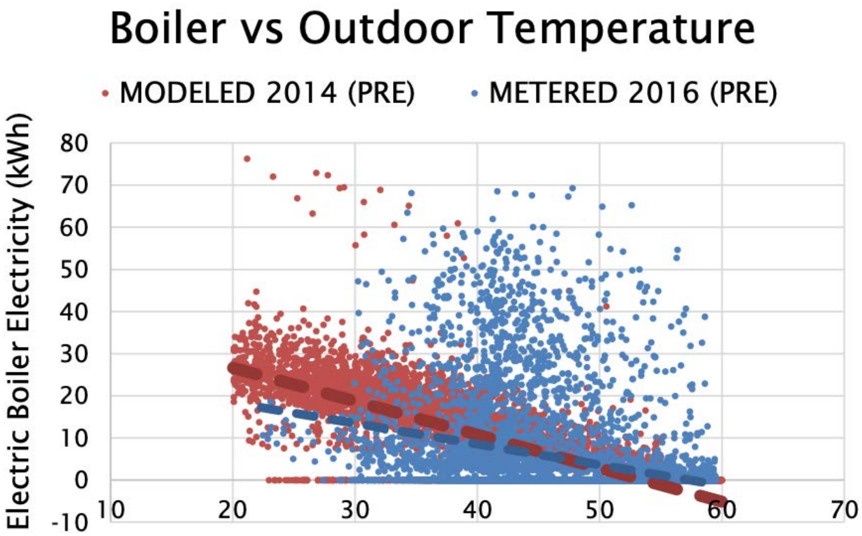
Better controls and zoning strategies are now available.

Thanks to advances in controls, conversion systems can now have very sophisticated controls and more effective zoning, relatively inexpensively, in existing buildings such as this one. Unfortunately, these advanced controls were not quite on the market in time for this project.

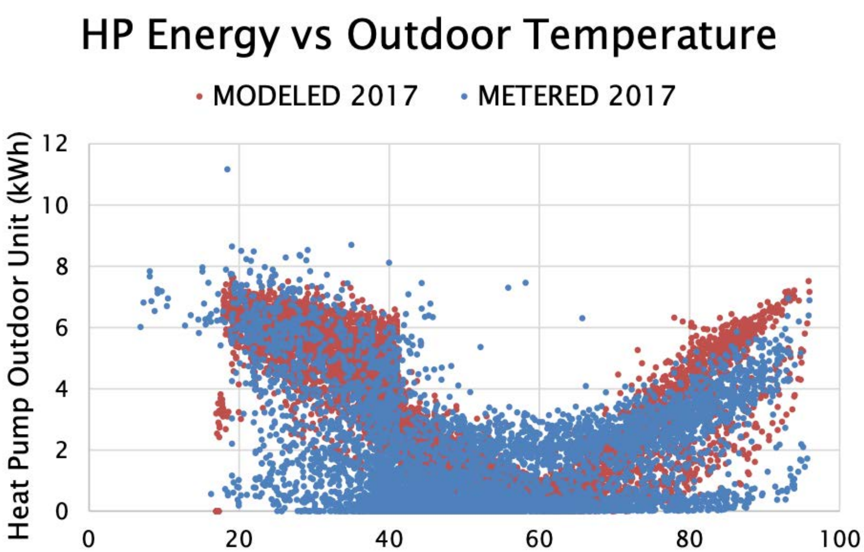
Designed and executed well, simple zoning can provide big benefits in system performance by more closely matching ventilation airflows to the actual need for ventilation air. The whole system operates more efficiently at lower airflows when controls maximize the number of hours at minimum airflow rates, thereby reducing energy use and extending filter life.

It's difficult to model the HVAC systems for projects like this one.

This is true even without unexpected operating modes and unknown occupant operating preferences. Below are examples of the project's positive modeling results.



The modeled range (red) produces a tighter hourly pattern than that exhibited by the metered data. Additionally, there are many more metered readings well above the curve, especially in the 40–50 F ambient conditions. Another example looks like this:



In this case, most of the modeling errors are an over-prediction of heat pump use in the cooling ambient temperature range, and in heating ambient conditions below 40 F. But, in the 40–70 F range, the model often significantly under-predicts energy use. This is likely due to the fact that most variable capacity heat pump systems do not cycle well under low-load conditions. They spend a significant amount of time at relatively high power levels as they cycle. This issue has been validated in the laboratory with the same and similar equipment.

The team learned that performance curves for much of the HVAC equipment being modeled do not reflect actual performance in the field. This is largely due to test and rating procedures not being load-based and not reflecting the actual hourly operation of the systems in most climate zones and in most buildings.

BETTERBRICKS
Powerful Energy Ideas. Delivered by NEEA.

PILOT REPORT

GOVERNMENT OFFICE BUILDING

CORVALLIS, OR

Background

This building is a district office for an Oregon government agency in Corvallis, Ore. With 13,200 sq. ft. of conditioned floor area, the building consists of four separate buildings collected together over time into a single, connected building complex.

This project was one in a series of pilots across the Northwest to help the region better understand the design, installation and expected energy savings of an advanced HVAC solution for small-to-medium commercial buildings. The system approach includes dedicated ventilation air (decoupled from primary heating and cooling air) with high-efficiency heat recovery, a high-efficiency heating and cooling system, and key design principles.

PRE-CONVERSION DETAILS

Conversion Challenges

The building's origins as four separate buildings introduced challenges to the designing and specifying of the conversion system as parts of the building are separated by concrete walls (formerly exterior walls) that extend all the way from the floor to the underside of the roof deck. This makes duct and refrigerant piping runs difficult for some spaces, and limits wireless communication for data collection.

Pre-Conversion HVAC

The existing HVAC system components consisted of two elderly Lennox gas RTUs and their controls, and the plenum portions of their duct systems. One of the units replaced had no legible nameplate, so the project team assumes has a similar heating and cooling capacity to the other unit replaced: 5.25 tons (net to the building) of natural gas heating capacity and 4 tons of cooling capacity. Overall, the conversion project removed 8 tons of cooling and 10.5 tons of heating, for two of the five building zones, comprising 24 percent of the total conditioned floor area.

OVERVIEW

Net Conditioned Area
13,200 sq. ft.

Project Floor Area
3,770 sq. ft.

Starting / Ending EUI
48.9 / 43.4 (Building)
28.2 / 22.7 (HVAC)

Typical Operating Hours
Monday – Friday,
9 a.m. to 5 p.m.

Setpoint/Setback in Heating Season
68° F / 68° F

Setpoint/Setback in Cooling Season
72° F / 72° F

Envelope Thermal Characteristics
Windows: **R-3**
Ext. Walls: **R-4**
Roof: **R-11**

Energy Utility
Consumers Power and NW Natural

SYSTEM CONVERSION DETAILS

Project-Cost Considerations

Both before and after conversion, this building proved to be moderately leaky, with a significant portion of the air leakage happening between the two floors of the building through three large HVAC chases serving the first level businesses. As there was no attempt to seal this connection between floors, the test was effectively measuring whole-building air leakage.

Multi-Zone Mini-Split Systems

The conversion system consists of one Mitsubishi 4-ton multi-zone mini-split system with two indoor air handlers (AHUs) that largely utilize existing ducting, and one Ventacity Systems VS1000RT heat-recovery ventilator (HRV). The HRV is mounted on one of the RTU curbs and the outdoor unit for the cooling and heating system is capped and mounted on the other one.

	Number / Make / Model	Cooling Capacity	Heating Capacity[1]	Zones
Existing System Type				
RTUs	(2) Lennox 4-ton	8 tons	10.5 tons	2

	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Conversion System				
Multi-zone DHP	(1) Mitsubishi MXZ8C48NAHZ	4 tons	4.5 tons	2
Packaged HRV	(4) Ventacity VS1000RT	1,025 cfm		1

The total installed system cost was $11.47/sq. ft., before incentives.

POST-CONVERSION SYSTEM PERFOMANCE

Post-Conversion Air Leakage

While the building has relatively new windows, the rest of the envelope has had limited improvement since its original construction. With no insulation, the roof (at R-11) and the exterior concrete walls create most of the cooling and heating loads.

The pre-conversion air leakage test results showed this building to be relatively airtight, but the HVAC system conversion still resulted in a slight reduction in air leakage. This is most likely due to the removal of the RTU systems and the team's attention to detail when installing the new systems at the existing curbs.

There was a notable difference between the "depressurize" and "pressurize" test regime results, most likely because of two dampers on exhaust fans that opened during the

[1] For natural gas-fired systems to the building after combustion losses (80% AFUE assumed).

"pressurize" test but were pulled closed during the "depressurize" test. Buildings with air change rates at or below 2.0 ACH_{50} require an effective, balanced ventilation system. Lack of such a ventilation system in the building before conversion may be the reason that agency staff almost immediately noticed the improvement in their indoor-air quality.

Air Leakage Test Results – Post-Conversion

Test Condition	Depressurize	Pressurize	Average
Enclosure Airtightness *[cfm/ft²@75 Pa]*	0.203	0.328	0.266
Equivalent Leakage Area *[ft²@75Pa]*	5.284	8.483	6.875
Air Changes *[per hour@50 Pa]*	1.61	2.58	2.09
Air Leakage Text Coefficient (C) *[cfm/Paⁿ]*	515.72	779.27	N/A
Flow Exponent (n) *[dimensionless]*	0.606	0.621	N/A
Squared Correlation Coefficient (r^2) *[dimensionless]*	0.999	0.998	N/A

Post-Conversion Energy Performance

Compared to a modeled base case of existing equipment being replaced with new electric RTU equipment that met current code requirements, the conversion project saved 11 percent of the total building energy use—all of it from heating and fan energy savings. The tables below show the energy impacts of the project. The modeled energy use intensity (EUI) is calibrated using energy bill and system sub-metering data.

Pre-Conversion Code Minimum (As Modeled)		Post-Conversion (As Modeled)		
EUI:	**48.9** kBtu/sq. ft./yr.	EUI:	**43.4** kBtu/sq. ft./yr.	11% Reduction
Fans:	**5.0** kBtu/sq. ft./yr.	Fans:	**4.0** kBtu/sq. ft./yr.	20% Reduction
Heating:	**22.3** kBtu/sq. ft./yr.	Heating:	**17.8** kBtu/sq. ft./yr.	20% Reduction
Cooling:	**0.9** kBtu/sq. ft./yr.	Cooling:	**0.9** kBtu/sq. ft./yr.	No Reduction
HVAC:	**28.2** kBtu/sq. ft./yr.	HVAC:	**22.7** kBtu/sq. ft./yr.	19% Reduction

Partial building area savings as a percentage of whole-building energy use.
Pre- and post-conversion modeled energy consumption are based on a typical meteorological year (TMY).
Models updated based on a full year of sub-metered energy end-use data.

Whole-Building Performance Extrapolation

The project team additionally modeled the estimated whole-building impacts if all the existing systems were converted in the same manner as the first two zones. The projected results, using the model calibrated to actual energy use in the two converted zones for a TMY, are representative of the general conversion results in an office-type occupancy. The base case pre-conversion model again assumes new RTUs that meet current code requirements.

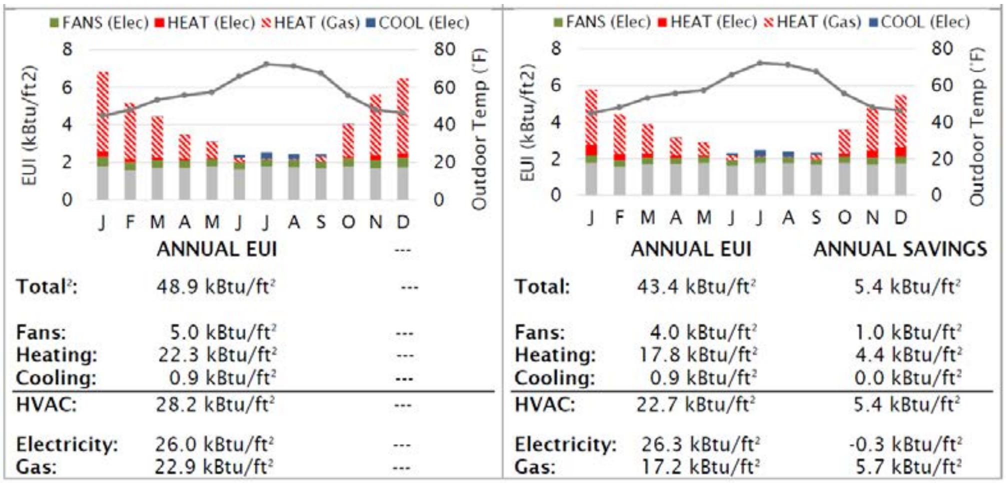
Pre-Conversion Code Minimum (As Modeled)		Post-Conversion (As Modeled)		
EUI:	**45.9** kBtu/sq. ft./yr.	EUI:	**28.1** kBtu/sq. ft./yr.	39% Reduction
Fans:	**5.1** kBtu/sq. ft./yr.	Fans:	**1.3** kBtu/sq. ft./yr.	75% Reduction
Heating:	**19.4** kBtu/sq. ft./yr.	Heating:	**5.3** kBtu/sq. ft./yr.	73% Reduction
Cooling:	**0.7** kBtu/sq. ft./yr.	Cooling:	**0.7** kBtu/sq. ft./yr.	No Reduction
HVAC:	**25.2** kBtu/sq. ft./yr.	HVAC:	**7.6** kBtu/sq. ft./yr.	70% Reduction

Extrapolated whole-building savings as a percent of whole building energy use.
Pre- and post-conversion modeled energy consumption are based on a typical meteorological year (TMY).
Models updated based on a full year of sub-metered energy end-use data.

Before the conversion, the HVAC systems comprised 55 percent of the total energy use at 25 kBtu/sq. ft., while, post-conversion, the HVAC systems comprise 27 percent of the total energy use at 8 kBtu/sq. ft.

Annual Performance Data

Using the building models calibrated with post-conversion energy bills and system monitoring, the figures below show system annual and monthly energy use data. Note that the pre-conversion base case is brand-new RTUs in place of the existing equipment.



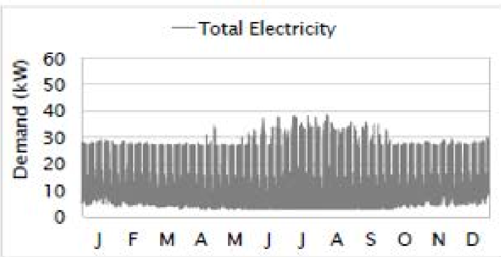
[2] Minor additive discrepancies are due to rounding.

Post-Conversion Demand Impacts

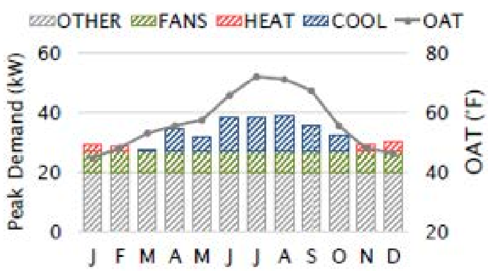
The conversion project additionally focused on quantifying electric demand savings. So as to be most representative of other whole-building projects, the demand savings analysis results are presented below for the extrapolated full conversion case. The conversion demand savings are largely driven by HVAC load reductions, and they vary month to month.

PRE-CONVERSION HVAC SYSTEM with NEW CODE MINIMUM EQUIPMENT

ELECTRICITY DEMAND



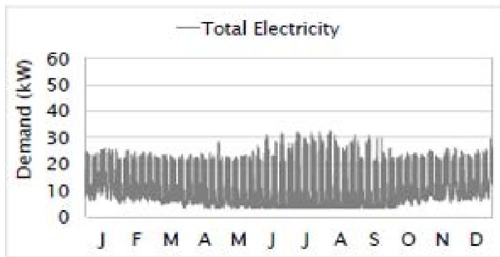
MONTHLY PEAK ELECTRICITY DEMAND



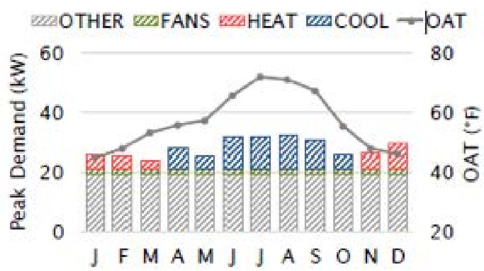
	PEAK DEMAND kW	PEAK OCCURENCE
JAN	29	18-JAN-08:30
FEB	29	08-FEB-08:30
MAR	28	29-MAR-16:00
APR	35	29-APR-15:00
MAY	32	30-MAY-15:00
JUN	38	25-JUN-15:00
JUL	38	10-JUL-15:00
AUG	39	08-AUG-15:00
SEP	36	12-SEP-15:00
OCT	32	01-OCT-15:00
NOV	29	28-NOV-08:30
DEC	30	30-DEC-08:30

POST-CONVERSION HVAC SYSTEM with NEW DOAS HRV + VRF SYSTEM

ELECTRICITY DEMAND



MONTHLY PEAK ELECTRICITY DEMAND



	PEAK DEMAND kW	PEAK OCCURENCE
JAN	26 (↓ 11%)	18-JAN-08:30
FEB	25 (↓ 11%)	08-FEB-08:30
MAR	24 (↓ 15%)	13-MAR-09:00
APR	28 (↓ 18%)	29-APR-15:00
MAY	25 (↓ 21%)	30-MAY-15:00
JUN	32 (↓ 16%)	25-JUN-15:30
JUL	32 (↓ 16%)	29-JUL-15:00
AUG	33 (↓ 16%)	08-AUG-14:30
SEP	31 (↓ 14%)	12-SEP-14:00
OCT	26 (↓ 19%)	01-OCT-15:00
NOV	26 (↓ 10%)	28-NOV-08:30
DEC	30 (↓ 2%)	30-DEC-08:30

→ Reduction in monthly peak demand by up to 21%.

To learn more about this and other efficient commercial HVAC solutions, visit **betterbricks.com/solutions/hvac.**

BETTERBRICKS
Powerful Energy Ideas. Delivered by NEEA.

PILOT REPORT

AIRPORT TERMINAL BUILDING

SEATTLE, WA

Background

While this 2-story 1930-vintage airport terminal building features passenger- and baggage-handling facilities, the majority of its space is used to house airport administration, security and customs functions. In addition to a small deli on the ground floor, there is a small basement where the controls for the airfield lighting system are located. Overall, average occupant density is relatively low compared to other office or mixed-use building spaces.

This building underwent a major renovation in 2002, but HVAC systems were left untouched. The 2002 remodel provided electrical and lighting upgrades, and added some insulation value to the ceilings and windows. The two large multi-zone rooftop units (RTUs) existing at the time were left in place, and a third of a similar type was added.

This project was one in a series of pilots across the Northwest to help the region better understand the design, installation and expected energy savings of an advanced HVAC solution for small-to-medium commercial buildings. The system approach includes dedicated ventilation air (decoupled from primary heating and cooling air) with high-efficiency heat recovery, a high-efficiency heating and cooling system, and key design principles.

PRE-CONVERSION DETAILS

Pre-Conversion System Details

While we don't have full system details, we estimate that the two original RTUs serving the two-story part of the building (1996 vintage) had a nominal cooling capacity of 40 tons each and a nominal heating capacity of 40 tons each (to the building).

The single-story part of the building, about 4,000 sq. ft., probably had a nominal 15-ton unit of the same type, added during the 2002 remodel. Nominal heating capacity for this unit would have been

OVERVIEW

Conditioned Floor Area
25,200 sq. ft.

Starting/Ending EUI
122 / 48.1 (Building)
87.3 / 13.3 (HVAC)

Typical Operating Hours
Monday-Friday,
8 a.m. to 5 p.m.

Setpoint/Setback in Heating Season
72° F / 72° F

Setpoint/Setback in Cooling Season
75° F / 75° F

Envelope Thermal Characteristics
Windows: **U-0.9**
Ext. Walls: **R3**
Roof: **R12.7**

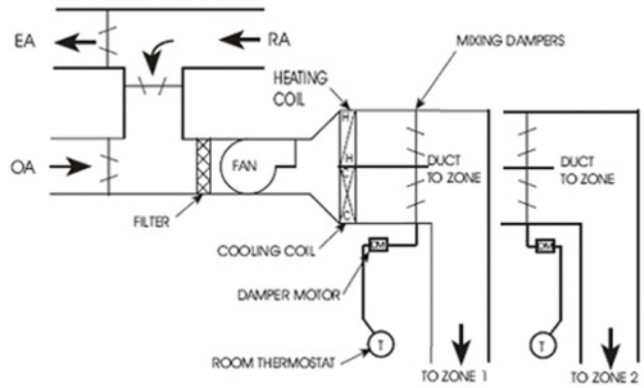
Energy Utility
Seattle City Light and Puget Sound Energy

While we don't have full system details, we estimate that the two original RTUs serving the two-story part of the building (1996 vintage) had a nominal cooling capacity of 40 tons each and a nominal heating capacity of 40 tons each (to the building).

The single-story part of the building, about 4,000 sq. ft., probably had a nominal 15-ton unit of the same type, added during the 2002 remodel. Nominal heating capacity for this unit

Dual-Deck RTUs

Despite the envelope improvements made during the 2002 remodel, the building's Energy Use Intensity (EUI) was still very high. This was almost entirely due to a significant amount of simultaneous heating and cooling, which is inherent in the design of the pre-conversion dual-deck RTUs. In these systems, the supply-air stream is divided into two parts, one part passing through a heating coil, and the other part passing through a cooling coil. Mixing dampers control the proportions of hot and cool air, blending the two air streams to deliver air at a particular temperature for each zone.



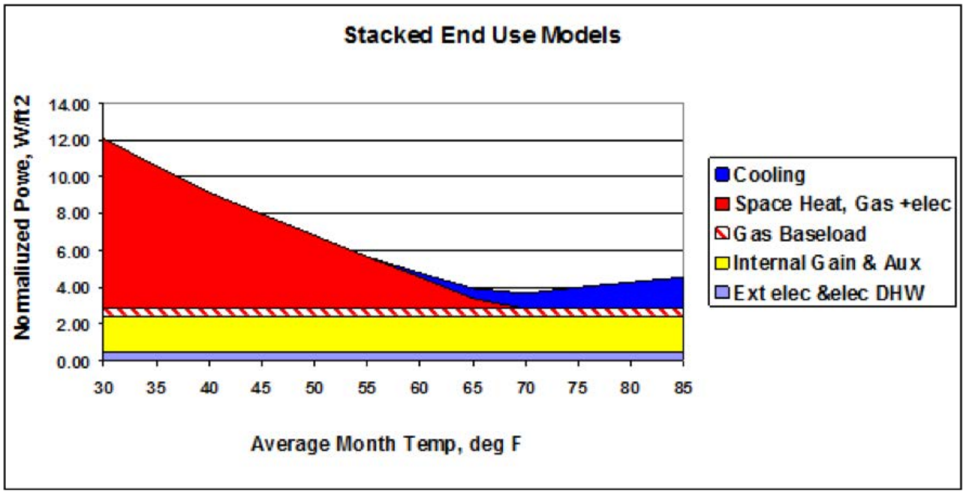
Because the air must be tempered, ventilation air is introduced as part of the supply-air stream that is divided between the two coils. This ventilated air is not based on occupancy in a zone, as zoning is based on heating and cooling loads. All in all, this is a very inefficient way to heat and cool a building.

Pre-Conversion Energy Performance

Due to a massive amount of simultaneous heating and cooling from the dual-deck RTU systems in place before the conversion, the building's EUI was much higher than most buildings with similar occupancies.

During hours with moderate outdoor temperatures (typically 55–70 F), these systems produced both cool air and warm air at the same time, mixing the two air streams to a desired delivered-air temperature. The Seattle climate often falls within this temperature range, resulting in a high frequency of simultaneous heating and cooling. The significant amount of fan power required by these units adds to the overall high EUI, along with the natural gas section that produces warm air at about 70% efficiency.

Relevant end-use categories for this building are shown below, plotted against outside-air temperatures. Simultaneous heating and cooling can be seen at the points at which the heating and cooling wedges overlap one another.



Pre-Conversion Air Leakage

Prior to the system conversion, the building's air leakage was tested using U.S. Army Corps of Engineers test protocols. The results below show the most significant air leakage pathways were found in the lobby entry doors, exhaust fans, and through the building duct system and RTUs.

Air Leakage Test Results – Pre-Conversion			
Test Condition	Depressurize	Pressurize	Average
Enclosure Airtightness *[cfm/ft²@75 Pa]*	0.307	0.321	0.314
Equivalent Leakage Area *[ft²@75Pa]*	10.43	10.92	10.70
Air Changes *[per hour@50 Pa]*	1.99	2.10	2.04
Air Leakage Text Coefficient (C) *[cfm/Paⁿ]*	1198	1412	N/A
Flow Exponent (n) *[dimensionless]*	0.569	0.542	N/A
Squared Correlation Coefficient (r^2) *[dimensionless]*	0.999	0.998	N/A

SYSTEM CONVERSION DETAILS

According to the project engineers, the pre-conversion system was sufficiently oversized for the load served that it was possible to cut system capacity by two-thirds for the new systems. This made the efficient Ventacity heat recovery ventilator (HRV) system an ideal replacement choice. In addition to being more efficient, the installed HRV systems also provide ventilation, with all three required to run at 900 cfm or more in order to meet calculated ASHRAE 62.1 ventilation requirements.

An additional benefit of the Ventacity HRVs is their special filtration ability. As the building is very close to both the airport runways on one side, and very densely trafficked rail lines and an interstate highway on the other, special filtration (activated carbon) is required for the ventilation air. Fortunately, Ventacity HRV systems can easily accommodate this important requirement.

	Number / Make / Model	Cooling Capacity	Heating Capacity[1]	Zones
Existing System Type				
Gas-electric dual-deck RTUs with home-run ducting to each zone, including for ventilation.	(2)	~40 tons each	Unknown	15
	(1)	~15 tons		
Electric resistance units in the vestibule/baggage areas.	(2)	N/A	5 kW (1.4 tons)	3
	(1)		10 kW (2.8 tons)	

	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Conversion System				
VRF Outdoor Units	(2) Mitsubishi PURY-P96YLMU-A	8 tons each	9 tons each	37
VRF Outdoor Unit	(1) Mitsubishi PURY-P120YLMU-A	10 tons	13.25 tons	
VRF Outdoor Unit	(1) Mitsubishi PURY-P72YLMU-A	6 tons	6.5 tons	
Packaged HRVs	(3) Ventacity VS1000RT	1,025 cfm each		4

Ventacity and NEEA engineers recommended that at least five VS1000 units be installed for a variety of reasons: to enable more efficient operation of the ventilation system,[2] to provide some boost capacity for unexpected levels of occupancy and to enable economizing at greater air volumes. These suggestions were not accepted by the contractor and the project engineer. See the Additional Findings section below for more information.

The total installed system cost was $36.85/sq. ft., before incentives.

[1] For natural gas-fired systems, to the building after combustion losses (75%-80% equipment ratings).
[2] Both the heat exchange and fan efficiency of HRVs are higher at lower flow rates.

POST-CONVERSION SYSTEM PERFORMANCE

Post-Conversion Air Leakage

Post-conversion, there was a notable reduction in building air leakage. The largest sources of reduction were likely the elimination of the exhaust fans (the HRV systems now provide exhaust for these spaces) and the elimination of the RTUs. As shown in the table below, the lobby entry doors continue to be a source of air leakage.

Air Leakage Test Results – Post-Conversion

Test Condition	Depressurize	Pressurize	Average
Enclosure Airtightness *[cfm/ft²@75 Pa]*	0.244	0.252	0.248
Equivalent Leakage Area *[ft²@75Pa]*	8.29	8.58	8.44
Air Changes *[per hour@50 Pa]*	1.58	1.64	1.61
Air Leakage Text Coefficient (C) *[cfm/Paⁿ]*	963.4	1018	N/A
Flow Exponent (n) *[dimensionless]*	0.566	0.562	N/A
Squared Correlation Coefficient (r^2) *[dimensionless]*	0.999	0.996	N/A

Post-Conversion Energy Performance

The tables and graphics below show the modeled pre-conversion system baseline typical meteorological year (TMY) energy use compared to the conversion system's TMY energy use. Both cases assume and account for the savings associated with the other efficiency measures installed as part of the 2002 remodel.

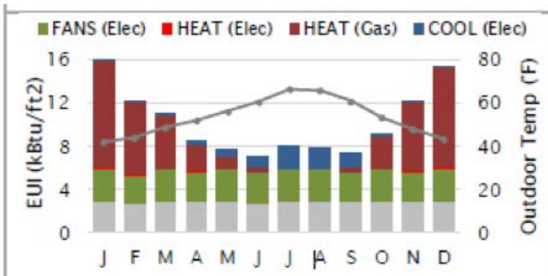
Pre-Conversion Code Minimum (As Modeled)		Post-Conversion (As Modeled)		
EUI:	**122.0** kBtu/sq. ft./yr.	EUI:	**48.1** kBtu/sq. ft./yr.	61% Reduction
Fans:	**33.9** kBtu/sq. ft./yr.	Fans:	**2.8** kBtu/sq. ft./yr.	92% Reduction
Heating:	**45.5** kBtu/sq. ft./yr.	Heating:	**8.1** kBtu/sq. ft./yr.	82% Reduction
Cooling:	**7.9** kBtu/sq. ft./yr.	Cooling:	**2.4** kBtu/sq. ft./yr.	70% Reduction
HVAC:	**87.3** kBtu/sq. ft./yr.	HVAC:	**13.3** kBtu/sq. ft./yr.	85% Reduction

Pre-Conversion (Actual Use)		Post-Conversion (Actual Use)		
EUI:	**175.4** kBtu/sq. ft./yr.	EUI:	**53.2** kBtu/sq. ft./yr.	70% Reduction
HVAC:	**140.7** kBtu/sq. ft./yr.	HVAC:	**18.5** kBtu/sq. ft./yr.	87% Reduction

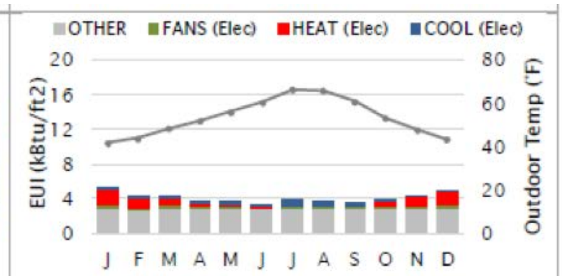
Pre- and post-conversion modeled energy consumption are based on a typical meteorological year (TMY).
Models updated based on a full year of sub-metered energy end-use data.

Annual Performance Data

Using the building models calibrated with post-conversion energy bills and system monitoring, the figures below show system annual and monthly energy use data.



	ANNUAL EUI	---
Total:	122.0 kBtu/ft²	---
Fans:	33.9 kBtu/ft²	---
Heating:	45.5 kBtu/ft²	---
Cooling:	7.9 kBtu/ft²	---
HVAC:	87.3 kBtu/ft²	---
Electricity:	76.8 kBtu/ft²	---
Gas:	45.3 kBtu/ft²	---



	ANNUAL EUI	ANNUAL SAVINGS
Total:	48.1 kBtu/ft²	73.9 kBtu/ft²
Fans:	2.8 kBtu/ft²	31.2 kBtu/ft²
Heating:	8.1 kBtu/ft²	37.3 kBtu/ft²
Cooling:	2.4 kBtu/ft²	5.5 kBtu/ft²
HVAC:	13.3 kBtu/ft²	73.9 kBtu/ft²
Electricity:	47.4 kBtu/ft²	29.4 kBtu/ft²
Gas:	0.7 kBtu/ft²	44.6 kBtu/ft²

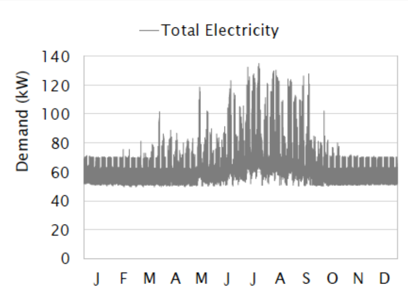
1 Minor additive discrepancies are due to rounding.

Post-Conversion Demand Impacts

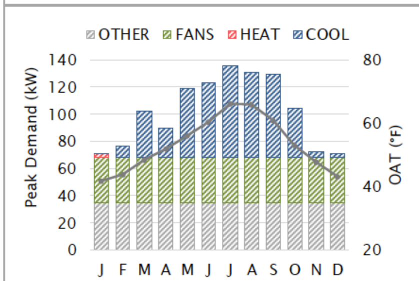
Even with the primary heating source being converted from gas to electric, this system conversion resulted in significant peak demand reduction in every month except January. The modeled demand reduction is calculated to be 43% to 61% (as much as 79 kW) in the spring, summer and fall. The figures below depict the daily and monthly peak electricity demand broken down by end-use in a typical weather year for Seattle, Washington.

PRE-CONVERSION HVAC SYSTEM with NEW CODE MINIMUM EQUIPMENT

ELECTRICITY DEMAND

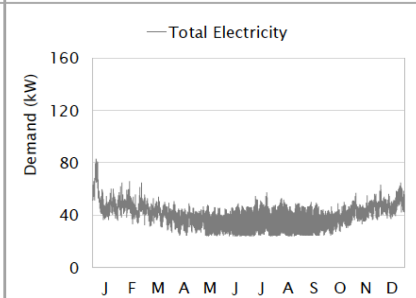


MONTHLY PEAK ELECTRICITY DEMAND

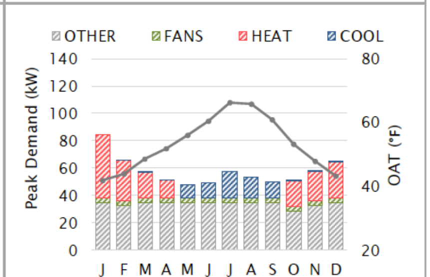


POST-CONVERSION HVAC SYSTEM with NEW DOAS HRV + VRF SYSTEM

ELECTRICITY DEMAND



MONTHLY PEAK ELECTRICITY DEMAND



Pre-Conversion

	PEAK DEMAND kW	PEAK OCCURRENCE
JAN	71	04-JAN-08:30
FEB	76	15-FEB-15:00
MAR	102	29-MAR-14:00
APR	89	11-APR-14:00
MAY	119	15-MAY-15:00
JUN	123	20-JUN-15:00
JUL	136	23-JUL-16:00
AUG	131	12-AUG-14:00
SEP	129	19-SEP-14:00
OCT	104	07-OCT-16:00
NOV	72	07-NOV-14:30
DEC	71	04-DEC-14:00

Post-Conversion

	PEAK DEMAND kW	PEAK OCCURRENCE
JAN	84 (↑ 18%)	05-JAN-08:30
FEB	66 (↓ 14%)	13-FEB-07:30
MAR	57 (↓ 44%)	02-MAR-08:30
APR	51 (↓ 43%)	06-APR-08:30
MAY	47 (↓ 60%)	15-MAY-15:00
JUN	49 (↓ 60%)	20-JUN-15:00
JUL	57 (↓ 58%)	23-JUL-16:00
AUG	53 (↓ 59%)	09-AUG-16:00
SEP	50 (↓ 61%)	13-SEP-16:30
OCT	50 (↓ 52%)	22-OCT-06:30
NOV	57 (↓ 20%)	05-NOV-07:30
DEC	65 (↓ 9%)	27-DEC-08:30

ADDITIONAL FINDINGS

In addition to the above, the project team found a variety of noteworthy lessons, outcomes and challenges unique to this conversion project.

- As a result of the decision to use three HRVs versus the recommended five, the system:
    - runs at nearly full capacity all the time
    - uses more fan power
    - features a lower heat exchange efficiency
    - has no additional capacity for full-flow economizing and boost for periods of higher-than-expected occupant densities

  To help project teams and engineers make more efficient and cost-effective decisions, early market intervention is required. Ideally, the specification ensures system optimization from the beginning.

- The installed HVAC system cost could have been reduced with more advantageous zoning. The building could have been serviced effectively by 24 or fewer indoor units, instead of the 37 being used.

- The project team's design solution will add maintenance costs, as each indoor unit has a filter that must be changed regularly, along with a condensate pump that has the possibility of failure.

- The pre-conversion building had two unusual characteristics that set it apart from other buildings that might undergo a similar HVAC system conversion:
    - First, the existing systems were very inefficient, with an unusual amount of simultaneous heating and cooling. This means that the savings potential was much larger in this project than in most of the others
    - Second, the base load in this building is relatively high, compared to other office-type occupancies. Part of this is due to a disproportionate amount of exterior lighting, and part is due to the airfield lighting controls in the basement.

- Such anomalies make it difficult for modelers to factor in. This, combined with inherent HVAC system modeling issues, can lead to significant modeling errors in the process of describing the building, its systems and its energy performance using different system types. The other lesson, apparent in all of the other pilot projects as well, is that our current engineering models do not consistently provide accurate modeling of either the base case or the conversion case.

- In the end, the unnecessarily high costs of this project were somewhat counterbalanced by the significant energy savings, making the project cost-effective overall.

- Even more significant than the high savings percentage is the ending HVAC EUI—13.3 kBtu/sq. ft.—which reinforces the primacy of high efficiency heat pump and ventilation system performance.

BETTERBRICKS
Powerful Energy Ideas. Delivered by NEEA.

PILOT REPORT

LAW OFFICE BUILDING

PORTLAND, OR

Background

This two-story historic 1907 building resides in an industrial part of Portland, Ore. The HVAC system conversion took place on the second floor of the building—the law office—which hadn't been occupied in almost three years. The remodeled law-office space includes 11,615 sq. ft. of net conditioned floor area, approximately 30 office spaces, 5 conference rooms, a lunch room, an exercise room, two sets of restrooms, and an ample amount of common and utility space. The building's first floor houses a convenience store, a flower shop and a pizza restaurant.

This project was one in a series of pilots across the Northwest to help the region better understand the design, installation and expected energy savings of an advanced HVAC solution for small-to-medium commercial buildings. The system approach includes dedicated ventilation air (decoupled from primary heating and cooling air) with high-efficiency heat recovery, a high-efficiency heating and cooling system, and key design principles.

PRE-CONVERSION DETAILS

Wall and Window Assembly

Exterior construction is 12-inch-thick brick walls, a 2x12" wood-joisted roof deck (newly insulated to a level of R-38 as part of the remodel), and approximately 900 sq. ft. of glazing area (with new windows installed as part of the remodel).

SYSTEM CONVERSION DETAILS

Variable refrigerant flow (VRF) and Very-High Efficiency Heat-Recovery Ventilator (HRV) Systems

The team replaced the existing systems with a conversion package consisting of four Ventacity Systems VS1000RT HRV units mounted on 4 of the 9 vacated roof curbs, and a Mitsubishi variable-refrigerant flow (VRF) system with 8 indoor AHUs, with the twinned outdoor units located on a fifth curb. Four of the original nine curbs were simply capped.

OVERVIEW

Gross Floor Area
11,615 sq. ft.

Starting / Ending EUI
51.4 / 19.1 (Building)
44.6 / 12.3 (HVAC)

Typical Operating Hours
Monday – Friday,
7 a.m. to 7 p.m.

Setpoint/Setback in Heating Season
70° F / 70° F

Setpoint/Setback in Cooling Season
72° F / 72° F

Envelope Thermal Characteristics
Windows: **R-3**
Ext. Walls: **R-8**
Roof: **R-38**

Energy Utility
Portland General Electric and NW Natural

The team replaced the existing systems with a conversion package consisting of four Ventacity Systems VS1000RT HRV units mounted on 4 of the 9 vacated roof curbs, and a Mitsubishi variable-refrigerant flow (VRF) system with 8 indoor AHUs, with the twinned outdoor units located on a fifth curb. Four of the original nine curbs were simply capped.

One HRV serves the large conference room, the second HRV serves four smaller conference rooms, and the remaining two HRVs serve the rest of the space. The VRF system serves 725 sq. ft. of conditioned area per ton of capacity. The same circuit feeds used for the replaced RTUs served the power requirements for the new systems. The near-elimination of the ventilation loads combined with the provision of incentives by Energy Trust of Oregon for new windows and LED lighting, resulted in a substantial reduction in system capacity. The final load reductions allowed the team to be comfortable cutting the system's heating and

	Number / Make / Model	Cooling Capacity	Heating Capacity[1]	Zones
Existing System Type				
RTUs	(8) Carrier 4-ton (1) York 3-ton	35 tons	43 tons	9

	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Conversion System				
VRF w/ heat recovery	(1) Mitsubishi PURY-192	16 tons	18 tons	8
Packaged HRVs	(4) Ventacity VS1000RT	1,025 cfm max. each		4

The total installed system cost was $15.61/sq. ft., before incentives.

POST-CONVERSION SYSTEM PERFOMANCE

Post-Conversion Air Leakage

Both before and after conversion, this building proved to be moderately leaky, with a significant portion of the air leakage happening between the two floors of the building through three large HVAC chases serving the first level businesses. As there was no attempt to seal this connection between floors, the test was effectively measuring whole-building air leakage.

[1] For natural gas-fired systems to the building after combustion losses (80% AFUE assumed).

Air Leakage Test Results – Post-Conversion			
Test Condition	Depressurize	Pressurize	Average
Enclosure Airtightness *[cfm/ft²@75 Pa]*	0.481	0.531	0.506
Equivalent Leakage Area *[ft²@75Pa]*	11.87	12.72	12.30
Air Changes *[per hour@50 Pa]*	4.53	4.96	4.74
Air Leakage Text Coefficient (C) *[cfm/Pa^n]*	1,299.4	1,306.5	N/A
Flow Exponent (n) *[dimensionless]*	0.577	0.598	N/A
Squared Correlation Coefficient (r^2) *[dimensionless]*	0.966	0.998	N/A

Post-Conversion Energy Performance

As the conversion space had not been occupied for about three years prior to the system conversion, utility bill data for the pre-conversion period was not available. The team modeled the base-case energy use intensity (EUI) of 57.4 kBtu/sq. ft./yr. on all new RTUs, properly installed, with functioning economizers, and with the same baseloads (i.e., lighting, plug loads and water heating) as the converted building. Overall, the project's energy performance has been excellent, with actual energy use being less in most months than estimated with the calibrated energy model. The building's performance might be improved even further with better occupant education on optimal use of systems controls.

While the building systems are performing very well from an energy use perspective, the electric base loads are abnormally low in this building (6.7 kBtu/sq. ft., rather than a more normal level of 12-15 kBtu/sq. ft.). This is due to the nature of the business that occupies the building, which leaves it lightly loaded with fewer than 20 people during a large fraction of normal business hours. If the building were more normally loaded (e.g., 60–70 people during normal business hours), the final whole-building EUI might be closer to 25 kBtu/sq. ft. Note that the additional base loads would be added to the base case HVAC alternative, as well, so the baseline EUI would be closer to 62 or 63 kBtu/sq. ft. Whole-building savings would then be 60 percent, and HVAC savings 75 percent, which is about the average expected for office buildings that have converted to from RTUs using this approach.

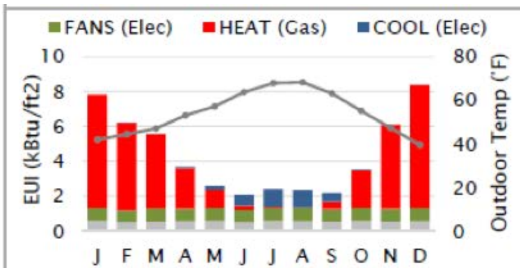
Pre-Conversion Code Minimum (As Modeled)	
EUI:	**51.4** kBtu/sq. ft./yr.
Fans:	**8.7** kBtu/sq. ft./yr.
Heating:	**32.5** kBtu/sq. ft./yr.
Cooling:	**3.4** kBtu/sq. ft./yr.
HVAC:	**44.6** kBtu/sq. ft./yr.

Post-Conversion (As Modeled)		
EUI:	**19.1** kBtu/sq. ft./yr.	63% Reduction
Fans:	**1.0** kBtu/sq. ft./yr.	89% Reduction
Heating:	**8.4** kBtu/sq. ft./yr.	74% Reduction
Cooling:	**2.8** kBtu/sq. ft./yr.	18% Reduction
HVAC:	**12.3** kBtu/sq. ft./yr.	72% Reduction

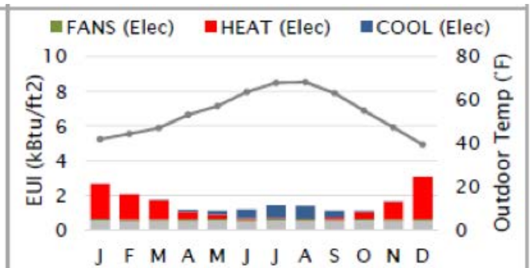
Pre- and post-conversion modeled energy consumption are based on a typical meteorological year (TMY).
Models updated based on a full year of sub-metered energy end-use data.

Annual Performance Data

Using the building models calibrated with post-conversion energy bills and system monitoring, the figures below show system annual and monthly energy use data.

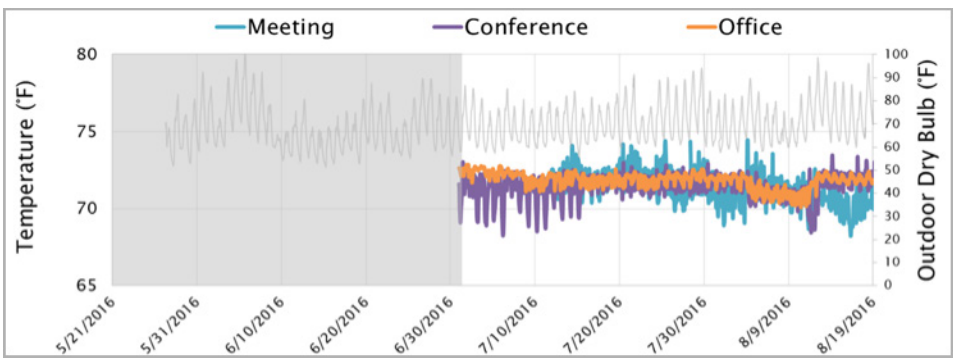


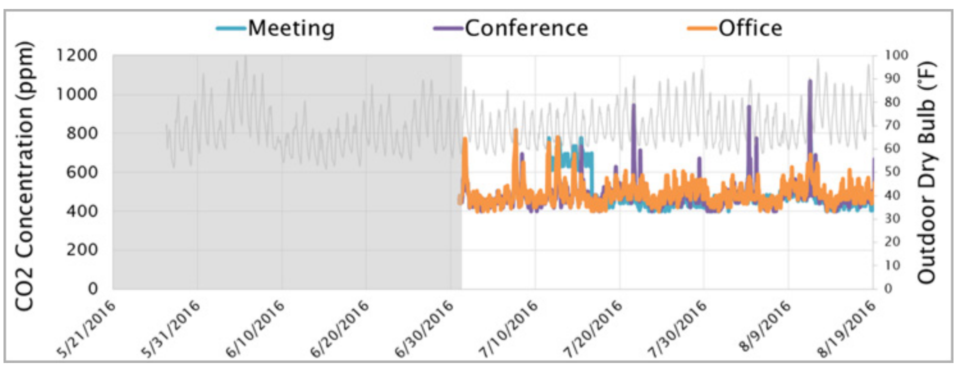
	ANNUAL EUI	---
Total:[2]	51.4 kBtu/ft²	---
Fans:	8.7 kBtu/ft²	---
Heating:	32.5 kBtu/ft²	---
Cooling:	3.4 kBtu/ft²	---
HVAC:	44.6 kBtu/ft²	---
Electricity:	18.9 kBtu/ft²	---
Gas:	32.5 kBtu/ft²	---



	ANNUAL EUI	ANNUAL SAVINGS
Total:	19.1 kBtu/ft²	32.3 kBtu/ft²
Fans:	1.0 kBtu/ft²	7.6 kBtu/ft²
Heating:	8.4 kBtu/ft²	24.1 kBtu/ft²
Cooling:	2.8 kBtu/ft²	0.6 kBtu/ft²
HVAC:	12.3 kBtu/ft²	32.3 kBtu/ft²
Electricity:	19.1 kBtu/ft²	-0.2 kBtu/ft²
Gas:	0.0 kBtu/ft²	32.5 kBtu/ft²

Daily Performance Data

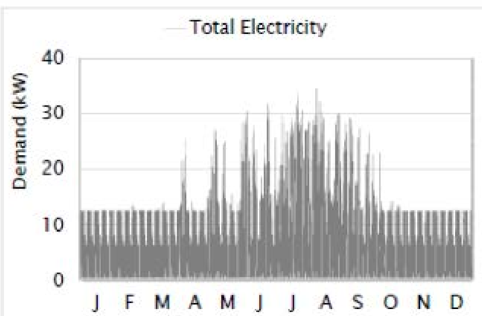


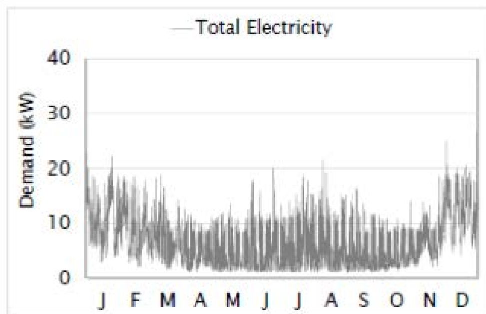


Post-Conversion Demand Impacts

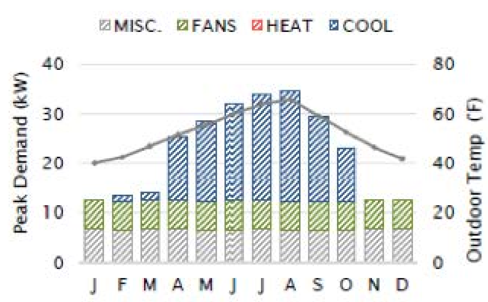
The use of very high efficiency heat recovery and VRF cooling in this conversion project result in a 39% to 52% reduction (9 kW to 15kW) in each month between April and October. However, there is a peak demand increase in the month of November through March due to the primary heating source being converted from gas-fired RTUs to electrically driven heat pumps. The figures below show the daily and monthly peak electricity demand modeled in a typical weather year for Portland, Oregon.

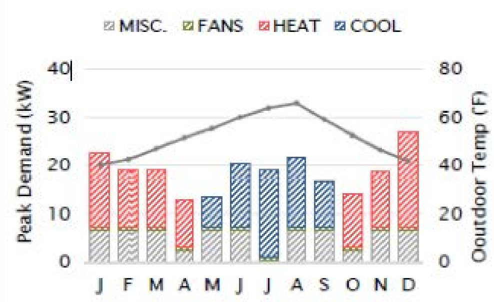
PRE-CONVERSION HVAC SYSTEM with NEW CODE MINIMUM EQUIPMENT	POST-CONVERSION HVAC SYSTEM with NEW DOAS HRV + VRF SYSTEM
ELECTRICITY DEMAND	ELECTRICITY DEMAND





MONTHLY PEAK ELECTRICITY DEMAND	MONTHLY PEAK ELECTRICITY DEMAND





	PEAK DEMAND kW	PEAK OCCURENCE (SEATTLE TMY)
JAN	13	31-JAN-13:00
FEB	13	19-FEB-15:30
MAR	14	19-MAR-13:00
APR	25	08-APR-15:30
MAY	29	31-MAY-15:00
JUN	32	24-JUN-15:30
JUL	34	22-JUL-10:00
AUG	35	09-AUG-15:30
SEP	29	09-SEP-15:00
OCT	23	07-OCT-14:00
NOV	13	28-NOV-13:30
DEC	13	30-DEC-13:30

	PEAK DEMAND kW	PEAK OCCURENCE (SEATTLE TMY)
JAN	23 (↑ 81%)	01-JAN-08:00
FEB	19 (↑ 41%)	05-FEB-08:30
MAR	19 (↑ 36%)	11-MAR-08:30
APR	13 (↓ 50%)	03-APR-06:30
MAY	14 (↓ 52%)	15-MAY-15:00
JUN	20 (↓ 37%)	24-JUN-14:00
JUL	19 (↓ 44%)	21-JUL-14:00
AUG	22 (↓ 37%)	09-AUG-14:00
SEP	17 (↓ 43%)	09-SEP-15:00
OCT	14 (↓ 39%)	30-OCT-06:00
NOV	19 (↑ 51%)	25-NOV-08:30
DEC	27 (↑ 114%)	31-DEC-08:30

To learn more about this and other efficient commercial HVAC solutions, visit **betterbricks.com/solutions/hvac.**

BETTERBRICKS
Powerful Energy Ideas. Delivered by NEEA.

PILOT REPORT

PIZZA RESTAURANT

CORVALLIS, OR

Background

Situated in an older, single-story masonry building with poor thermal characteristics, this pizza restaurant sought better thermal comfort and efficiency with an HVAC system conversion. With a limited seating area and a take-out window located on the end of the connected storefront unit, the project team had to consider how the small floor area relative to the cooking loads would distort the typical metrics by which whole-building efficiency is assessed.

This project was one in a series of pilots across the Northwest to help the region better understand the design, installation and expected energy savings of an advanced HVAC solution for small-to-medium commercial buildings. The system approach includes dedicated ventilation air (decoupled from primary heating and cooling air) with high-efficiency heat recovery, a high-efficiency heating and cooling system, and key design principles.

PRE-CONVERSION DETAILS

Energy-Use Intensity

The starting energy use intensity (EUI) for the project was over 1,400 kBtu/sq. ft., the vast majority of which came from cooking, refrigeration and hot-water loads. With a more typical restaurant dining-area fraction, the EUI might be closer to 750 kBtu/sq. ft., which is very high, but not atypical of this type of occupancy.

Kitchen Hood and Control Complications

The vent hood exhaust in the kitchen and its make-up-air unit substantially complicated the project's full-system conversion. Additionally, the existing equipment was not well-controlled. For instance, the vent hood exhaust fan was not interlocked with the make-up-air fan, and at certain times, restaurant staff would operate the exhaust fan without make-up air, which depressurized the space enough to backdraft the two natural-gas-fired water heaters. The equipment was generally old, marginally maintained, and manually controlled.

OVERVIEW

Gross Floor Area
1,360 sq. ft. (excluding basement)

Starting / Ending EUI
1,470 / 1,352 (Building)
277 / 159 (HVAC)

Typical Operating Hours
Sunday – Wednesday, 8 a.m. to 11 p.m.
Thursday – Saturday, 8 a.m. to 12 a.m.

Setpoint/Setback in Heating Season
68° F / 68° F

Setpoint/Setback in Cooling Season
74° F / 74° F

Enclosure Thermal Characteristics
Windows: **R-1**
Ext. Walls: **R-3**
Roof: **R-6**

Energy Utility
Pacific Power and NW Natural

SYSTEM CONVERSION DETAILS

Ductless Heating & Cooling and Very-High Efficiency Heat-Recovery Ventilator (HRV) Systems

The business owners decided to accomplish the conversion project in two phases. The first phase installed the new ductless heat pump (DHP) heating and cooling system, and configured the existing RTU for ventilation only (which was the best option for ventilation at the time). In the second phase, which began the pilot project, the contractor suggested retrofitting the RTU with a Ventacity heat-recovery ventilator (HRV). Despite the complexity of restaurant HVAC systems and their interactions with cooking and service hot-water systems, the project team decided to acquire data from a project that was only the heating/cooling system portion of a complete conversion. They did this in hopes the data would demonstrate incremental savings provided by the ventilation component of the conversion.

	Number / Make / Model	Cooling Capacity	Heating Capacity[1]	Zones
Existing System Type				
RTU	(1) Lennox 7.5-ton	7.5 tons	10.8 tons	1

	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Conversion System				
Single-zone DHP (Phase 1)	(1) Fujitsu AOU24RLXFW	2 tons	2.25 tons	1
Single-zone DHP (Phase 2)	(3) Daikin RXS36LVJU	3 tons x 3	3.2 tons x 3	3
Single-zone DHP (Phase 2)	(1) Ventacity VS1000RT	1,025 cfm		1

The HRV was set on the existing RTU curb, with a notable amount of supply and exhaust ducting above the roof. There was also an equivalent amount of supply and exhaust ducting in the unconditioned cavity between the dining-room ceiling and the roof deck.

The total installed system cost was $27.50/sq. ft.

[1] For natural gas-fired systems to the building after combustion losses (80% AFUE assumed).

POST-CONVERSION SYSTEM PERFOMANCE

Post-Conversion Air Leakage

Air leakage rates for this building were very high, primarily due to the large make-up and exhaust-air openings for the venting system that serves the pizza oven. In order for the new, balanced HRV system serving the dining room to be effective the vent hood systems must be operated in such a way as to only slightly negatively pressurize the kitchen area, relative to the dining area. Additionally, the HRV must be programmed to operate with a slight surplus of fresh supply air relative to exhaust air to preserve the overall pressure gradient from dining room to kitchen.

During the commissioning process, the indoor-to-outdoor pressure difference was found to be about -30 pascals (Pa) during normal operating conditions (all equipment running, doors closed). Without the make-up air system for the vent hood running, indoor pressure dropped further to about -34 Pa relative to outdoors. With all equipment off and doors closed, relative pressure was about -1.5 Pa. The two vent hood fans and the other two exhaust fans clearly have a significant impact on building pressurization.

The building has a number of supply and exhaust pathways connected to the outdoors, and several of them that did not appear to be in service leaked significant amounts of air. This demonstrates that air leakage and unbalanced airflows also have a significant impact on energy use in this building, particularly with the make-up air unit supplying insufficient flows to provide reasonable pressure balance with the pizza oven exhaust hood in operation.

Air Leakage Test Results – Post-Conversion

Test Condition	Depressurize	Pressurize	Average
Enclosure Airtightness *[cfm/ft²@75 Pa]*	1.616	1.673	1.644
Equivalent Leakage Area *[ft²@75Pa]*	8.517	8.815	8.665
Air Changes *[per hour@50 Pa]*	19.28	19.92	19.60
Air Leakage Text Coefficient (C) *[cfm/Pa^n]*	1,246.1	1,267.4	N/A
Flow Exponent (n) *[dimensionless]*	0.513	0.517	N/A
Squared Correlation Coefficient (r^2) *[dimensionless]*	0.999	0.999	N/A

Post-Conversion Energy Performance

Changes to the other end-use equipment during project implementation complicated the performance assessment for this project. The three most significant changes were new natural-gas-fired water heaters, new commercial refrigerators, and a tune-up of the make-up air system for the vent hood that resulted in a notable increase in make-up airflow. The team had to account for these changes in the modeling.

This project is also irregular in that the original installation of the new heat-pump equipment occurred well before the installation of the new ventilation system. These systems were installed in response to the inadequacy of the cooling system in place at the time (the 7.5-ton RTU). This meant the first stage of the project increased the cooling capacity of the HVAC system by 37 percent and used the RTU system to provide conditioned ventilation air. It's unlikely that the modeling was able to capture the full energy impact of this mode of operation, but the actual increase in cooling energy use has likely been less than the 44 percent shown in the table below. Likewise, the heating and fan energy-use reduction may have been somewhat larger than shown, depending on how efficiently the RTU was estimated to have provided the conditioning of the ventilation air—both in terms of fan power and conditioning energy during the winter. While RTU part-load efficiency has improved quite a bit over the last 10 to 15 years, it still falls off rapidly under low-load conditions. Low-load conditions in the heating mode would dominate during the course of a year's operation at this restaurant.

Restaurants are a challenging occupancy type when it comes to finding ways to significantly reduce commercial building energy use, but the quantitative reductions made possible by this project's systems are not trivial, even if the reduction percentages seem small. Overall, the client and building occupants impacted by the project are pleased with the performance of the new systems. Summer comfort has improved and there has been a notable reduction in monthly energy bills.

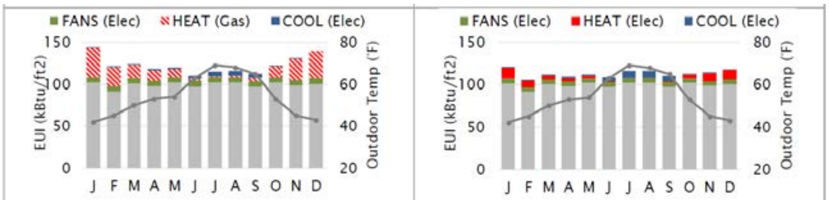
The combination of the energy-intensive occupancy type and the restaurant's small seating area resulted in what appear to be very high EUIs for the project. Even with twice the square footage (resulting in a very small increase in overall EUI), the EUIs would be very high, but still in line with typical restaurant EUIs.

Pre-Conversion Code Minimum (As Modeled)		Post-Conversion (As Modeled)		
EUI:	**1470** kBtu/sq. ft./yr.	EUI:	**1352** kBtu/sq. ft./yr.	8% Reduction
Fans:	**80.1** kBtu/sq. ft./yr.	Fans:	**65.6** kBtu/sq. ft./yr.	18% Reduction
Heating:	**174.1** kBtu/sq. ft./yr.	Heating:	**59.7** kBtu/sq. ft./yr.	66% Reduction
Cooling:	**23.1** kBtu/sq. ft./yr.	Cooling:	**33.3** kBtu/sq. ft./yr.	44% Increase
HVAC:	**277.3** kBtu/sq. ft./yr.	HVAC:	**158.6** kBtu/sq. ft./yr.	43% Reduction

Pre- and post-conversion modeled energy consumption are based on a typical meteorological year (TMY). Models updated based on a full year of sub-metered energy end-use data.

Annual Performance Data

Using the building models calibrated with post-conversion energy bills and system monitoring, the figures below show system annual and monthly energy use data.



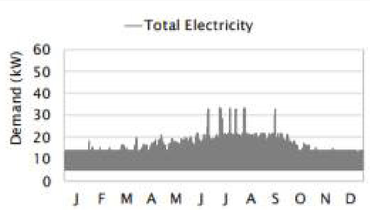
	ANNUAL EUI	---		ANNUAL EUI	ANNUAL SAVINGS
Total[2]:	1470 kBtu/ft²	---	Total:	1352 kBtu/ft²	119 kBtu/ft²
Fans:	80 kBtu/ft²	---	Fans:	66 kBtu/ft²	14 kBtu/ft²
Heating:	174 kBtu/ft²	---	Heating:	60 kBtu/ft²	114 kBtu/ft²
Cooling:	23 kBtu/ft²	---	Cooling:	33 kBtu/ft²	-10 kBtu/ft²
HVAC:	277 kBtu/ft²	---	HVAC:	159 kBtu/ft²	119 kBtu/ft²
Electricity:	268 kBtu/ft²	---	Electricity:	323 kBtu/ft²	-55 kBtu/ft²
Gas:	1202 kBtu/ft²	---	Gas:	1028 kBtu/ft²	174 kBtu/ft²

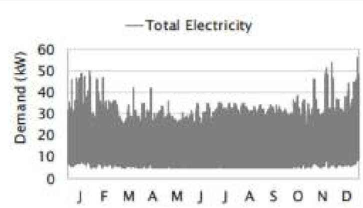
[2] Minor additive discrepancies are due to rounding.

Post-Conversion Demand Impacts

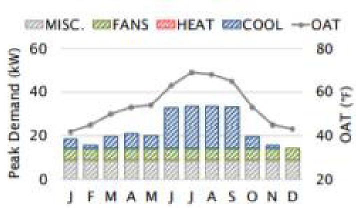
Due to this project replacing gas-fired RTUs with electric heat pumps, there was a net increase in electricity demand for heating. The figures below show the daily and monthly peak electricity demand modeled in a typical weather year for Corvallis, Oregon.

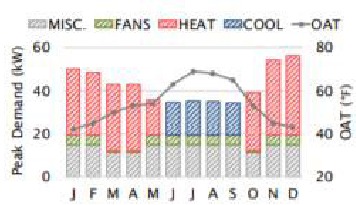
PRE-CONVERSION HVAC SYSTEM with NEW CODE MINIMUM EQUIPMENT	POST-CONVERSION HVAC SYSTEM with NEW DOAS HRV + VRF SYSTEM
ELECTRICITY DEMAND	ELECTRICITY DEMAND





MONTHLY PEAK ELECTRICITY DEMAND	MONTHLY PEAK ELECTRICITY DEMAND





	PEAK DEMAND kW	PEAK OCCURENCE
JAN	18	31-JAN-14:00
FEB	16	04-FEB-13:00
MAR	20	30-MAR-14:00
APR	21	29-APR-16:00
MAY	20	28-MAY-17:00
JUN	33	26-JUN-13:00
JUL	33	09-JUL-16:00
AUG	33	08-AUG-14:00
SEP	33	15-SEP-13:30
OCT	20	01-OCT-13:30
NOV	16	04-NOV-13:00
DEC	14	01-DEC-18:30

	PEAK DEMAND kW	PEAK OCCURENCE
JAN	50 (↑ 172%)	27-JAN-23:00
FEB	48 (↑ 210%)	14-FEB-08:30
MAR	43 (↑ 117%)	24-MAR-07:00
APR	43 (↑ 103%)	15-APR-06:00
MAY	36 (↑ 79%)	06-MAY-22:00
JUN	35 (↑ 6%)	25-JUN-16:00
JUL	35 (↑ 5%)	09-JUL-16:00
AUG	35 (↑ 5%)	08-AUG-14:30
SEP	34 (↑ 4%)	19-SEP-12:00
OCT	39 (↑ 98%)	11-OCT-06:30
NOV	54 (↑ 246%)	28-NOV-08:30
DEC	56 (↑ 302%)	30-DEC-08:30

ADDITIONAL FINDINGS

Vent hoods negatively pressurize the kitchen in the vent hood area by providing less make-up air than exhaust airflow. In general, a slight imbalance (-10 Pa or so) is enough to keep the air in the kitchen area, and from the dining room, flowing toward the exhaust vent hood while maintaining capture effectiveness over the cooking surfaces and equipment. If the supply and make-up air systems are unbalanced, the heat recovery function of the dining-room HRV system will be compromised. This will occur with any HRV system that has unbalanced flows, as a significant fraction of the HRV-supplied fresh air will be extracted through the vent hood and not through the HRV heat exchanger.

In this project, the team did a considerable amount of work to tune the vent hood and make-up air systems so they worked properly together to achieve balance. To measure building envelope leakage, blower-door testing requires all systems that are connected to the outdoors to be in off mode. As such, the team did not initially measure the amount of operational airflow imbalance during business hours.

When the new natural-gas water-heaters backdrafted while firing, a building pressure balance investigation revealed the building to be at -30 Pa relative to outdoors with the vent hood make-up air system on, and at -34 Pa with the make-up air system off. The slight difference between these two measurements suggested that the make-up air unit was falling far short of the needed flow rate to more nearly balance the vent hood exhaust. Additionally, the team tightened a loose drive belt, which stopped the water heater backdrafting and significantly improved the building's pressure balance.

The team's conclusion was that the vent hood and make-up air system in a restaurant can swamp the impact of a good, balanced HRV system. It is recommended to spend the resources required to ensure these systems are performing optimally.

BETTERBRICKS
Powerful Energy Ideas. Delivered by NEEA.

PILOT REPORT

OFFICE SPACE

SEATTLE, WA

Background

A leading Seattle-based energy engineering and research firm leased the third floor of a 3.5-story mixed-use historic building in downtown Seattle. The space includes open office areas, conference rooms, a lunchroom, and a server room. The firm required an HVAC system-conversion to meet their efficiency and comfort standards. Given the building's historic status, no modifications could be made to the envelope.

This project was one in a series of pilots across the Northwest to help the region better understand the design, installation and expected energy savings of an advanced HVAC solution for small-to-medium commercial buildings. The system approach includes dedicated ventilation air (decoupled from primary heating and cooling air) with high-efficiency heat recovery, a high-efficiency heating and cooling system, and key design principles.

PRE-CONVERSION DETAILS

Wall and Window Assembly

The building's exterior walls are uninsulated brick. A portion of the roof has R-11 insulation, and the windows are aluminum-framed, double-paned units.

Pre-Conversion HVAC

Prior to the conversion, the space was served by a multi-zone, rooftop unit (RTU) variable air volume (VAV) system with electric-resistance terminal reheat units in the individual zones. This system provided no heating to the primary section—all heating was done by the electric resistance re-heat VAV units. Post-conversion, this system continues to serve the second floor of the building.

OVERVIEW

Gross Floor Area
6,100 sq. ft.

Project Floor Area
5,911 sq. ft.

Starting / Ending EUI
51.2 / 29.7 (Building)
31.2 / 9.6 (HVAC)

Typical Operating Hours
Monday – Friday,
7 a.m. to 10 p.m.

Setpoint/Setback in Heating Season
72° F / 64° F

Setpoint/Setback in Cooling Season
75° F / 80° F

Envelope Thermal Characteristics
Windows: **R-1.7**
Ext. Walls: **R-5**
Roof: **R-11**

Energy Utility
Seattle City Light

SYSTEM CONVERSION DETAILS

Occupant-Designed System

The engineering firm occupants designed and specified their new systems themselves, as they had experience in the application of the pilot project's concepts. They hired a local HVAC contractor to implement their design. Due to their triple-net lease, the occupants had to pay for the upgrades themselves, with the approval of the landlord.

Variable Refrigerant Flow (VRF) and Very-High Efficiency Heat-Recovery Ventilator (HRV) Systems

The conversion cooling and heating system is a Mitsubishi variable-refrigerant flow (VRF) system with 12 ductless indoor units. Ventilation is provided by a single Ventacity Systems VS 1000 RT heat-recovery ventilator (HRV). The system's heat-recovery module allows simultaneous heating and cooling. The installed system cost was $16.83 /sq. ft.

The original system serving the space had a cooling capacity of 35 tons, but the project team estimated the third-floor capacity to be 14 tons. They determined the system was oversized for the load before the conversion cut the remaining load nearly in half.

The in-house engineering team's drawings called for a 12-ton system (PURY-P144) serving a calculated 10-ton cooling load and an 8.7-ton heating load—a significant downsizing of the system. The climate in the Seattle area typically requires much less cooling than inland climates, so the heating load is most often the governing design load. Further, the building has a significant amount of unshaded west-facing glazing, which imposes a significant fraction of the cooling load on clear days. As the variable capacity heat pump systems typically specified for this type of conversion have more heating capacity than cooling capacity, the space might have been well served with a 10-ton system (600 sq. ft. per ton). The engineering team's design also specified 12 indoor fan/coil units for the space, rather than the recommended fewer air-handling units with short duct runs.

	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Existing System Type				
All-electric RTU w/electric-resistance terminal heat	(1) Carrier 50AK-035CR-511HH	14 tons	16.4 tons	4

	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Conversion System				
VRF	(1) Mitsubishi PURY-P168TLMU-A	14 tons	15.7 tons	12
Packaged HRVs	(1) Ventacity VS1000RT	1,025 cfm		1

POST-CONVERSION SYSTEM PERFOMANCE

Post-Conversion Air Leakage

The project team blower-door tested the space for air leakage using the U.S. Army Corps of Engineers' test protocols. The tests found the most significant air leakage pathways were the windows (perimeter and individual insulated glass units), abandoned conduits, the bathroom exhaust fan stack (which does not have a backdraft damper or any operable damper), and under the exit door to the stair/elevator core.

Air Leakage Test Results – Post-Conversion

Test Condition	Depressurize	Pressurize	Average
Enclosure Airtightness *[cfm/ft^2@75 Pa]*	0.207	0.204	0.206
Equivalent Leakage Area *[ft^2@75Pa]*	2.65	2.62	2.64
Air Changes *[per hour@50 Pa]*	2.66	2.61	2.64
Air Leakage Text Coefficient (C) *[cfm/Pa^n]*	344.4	305.0	N/A
Flow Exponent (n) *[dimensionless]*	0.541	0.566	N/A
Squared Correlation Coefficient (r^2) *[dimensionless]*	0.995	0.999	N/A

Post-Conversion Energy Performance

Despite not upgrading the 100-year old building envelope or downsizing the equipment as recommended, the project is performing very well. As the RTU that's still in place now serves only the second floor of the building, its energy use will likely rise somewhat due to frequent short-cycling in the cooling mode. But in the new third floor spaces, part-load efficiencies are very good and energy use has declined significantly.

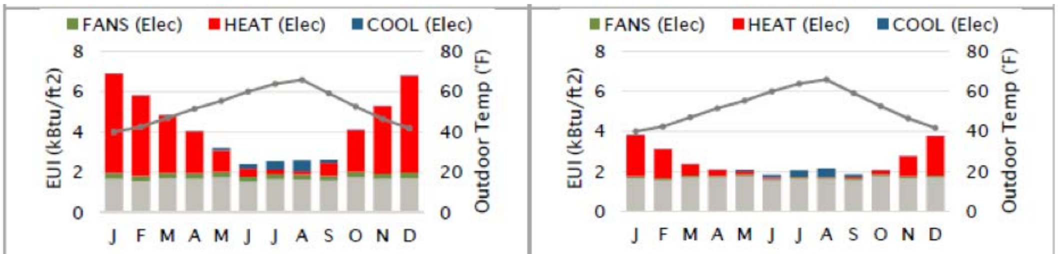
The table below shows the modeled new RTU system baseline typical meteorological year (TMY) energy use compared to the new systems' TMY energy use, with the latter based on a model calibrated with a full year of sub-metered data.

Pre-Conversion Code Minimum (as modeled)		Post-Conversion		
EUI:	**51.3** kBtu/sq. ft./yr.	EUI:	**29.7** kBtu/sq. ft./yr.	42% Reduction
Fans:	**2.8** kBtu/sq. ft./yr.	Fans:	**0.9** kBtu/sq. ft./yr.	68% Reduction
Heating:	**27.1** kBtu/sq. ft./yr.	Heating:	**7.9** kBtu/sq. ft./yr.	71% Reduction
Cooling:	**1.2** kBtu/sq. ft./yr.	Cooling:	**0.8** kBtu/sq. ft./yr.	33% Reduction
HVAC:	**31.2** kBtu/sq. ft./yr.	HVAC:	**9.6** kBtu/sq. ft./yr.	69% Reduction

Pre- and post-conversion modeled energy consumption are based on a typical meteorological year (TMY).
Models updated based on a full year of sub-metered energy end-use data.

Annual Performance Data

Using the building models calibrated with post-conversion energy bills and system monitoring, the figures below show system annual and monthly energy use data.



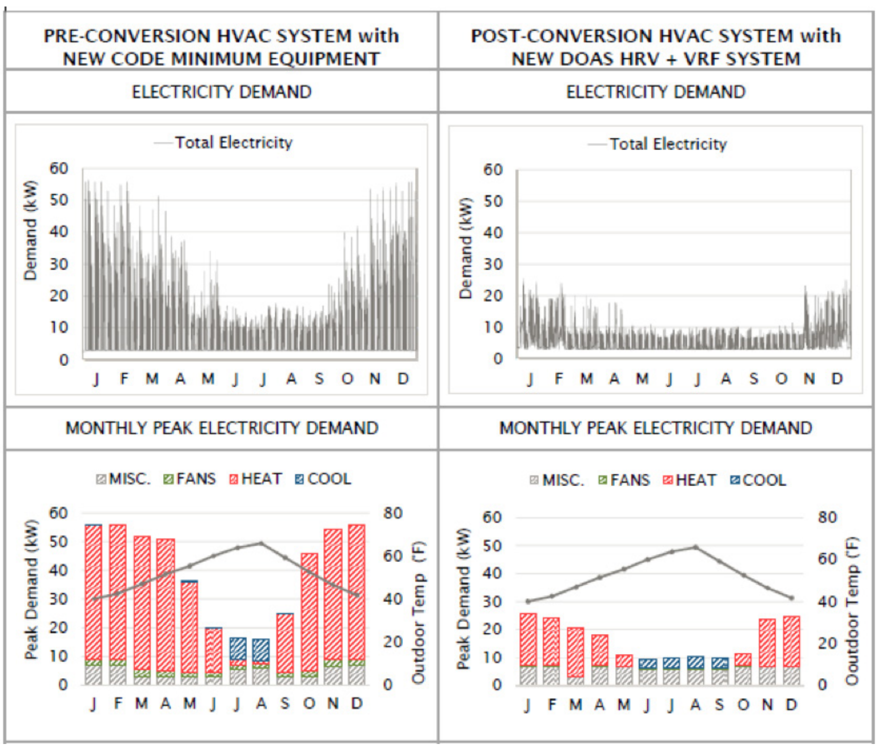
	ANNUAL EUI	---		ANNUAL EUI	ANNUAL SAVINGS
Total:[2]	51.3 kBtu/ft²	---	Total:	29.7 kBtu/ft²	21.6 kBtu/ft²
Fans:	2.8 kBtu/ft²	---	Fans:	0.9 kBtu/ft²	1.9 kBtu/ft²
Heating:	27.1 kBtu/ft²	---	Heating:	7.9 kBtu/ft²	19.2 kBtu/ft²
Cooling:	1.2 kBtu/ft²	---	Cooling:	0.8 kBtu/ft²	0.5 kBtu/ft²
HVAC:	31.2 kBtu/ft²	---	HVAC:	9.6 kBtu/ft²	21.6 kBtu/ft²
Electricity:	51.3 kBtu/ft²	---	Electricity:	29.7 kBtu/ft²	21.6 kBtu/ft²

ipment as
now serves
to frequent
ciencies are

gical year
based on a

Post-Conversion Demand Impacts

This project also delivered significant demand savings. The winter demand savings are especially significant in this case because all space heating in the existing system (and the base-case-modeled system) was done with electric-resistance VAV boxes.



Pre-Conversion

	PEAK DEMAND kW	PEAK OCCURENCE (SEATTLE TMY)
JAN	56	07-JAN-09:00
FEB	56	18-FEB-09:00
MAR	52	25-MAR-08:00
APR	51	01-APR-08:00
MAY	36	20-MAY-07:00
JUN	20	07-JUN-07:00
JUL	16	30-JUL-17:00
AUG	16	23-AUG-17:00
SEP	25	27-SEP-07:00
OCT	46	28-OCT-07:00
NOV	54	25-NOV-09:00
DEC	56	23-DEC-09:00

Post-Conversion

	PEAK DEMAND kW	PEAK OCCURENCE (SEATTLE TMY)
JAN	26 (↓ 54%)	07-JAN-08:00
FEB	24 (↓ 56%)	18-FEB-08:00
MAR	20 (↓ 61%)	18-MAR-08:00
APR	18 (↓ 64%)	18-APR-07:00
MAY	11 (↓ 69%)	27-MAY-09:30
JUN	10 (↓ 52%)	10-JUN-14:00
JUL	10 (↓ 39%)	24-JUL-16:00
AUG	10 (↓ 34%)	30-AUG-14:00
SEP	10 (↓ 60%)	13-SEP-16:30
OCT	12 (↓ 75%)	28-OCT-09:30
NOV	23 (↓ 57%)	11-NOV-10:30
DEC	25 (↓ 55%)	26-DEC-10:30

To learn more about this and other efficient commercial HVAC solutions, visit **betterbricks.com/solutions/hvac.**

BETTERBRICKS
Powerful Energy Ideas. Delivered by NEEA.

PILOT REPORT

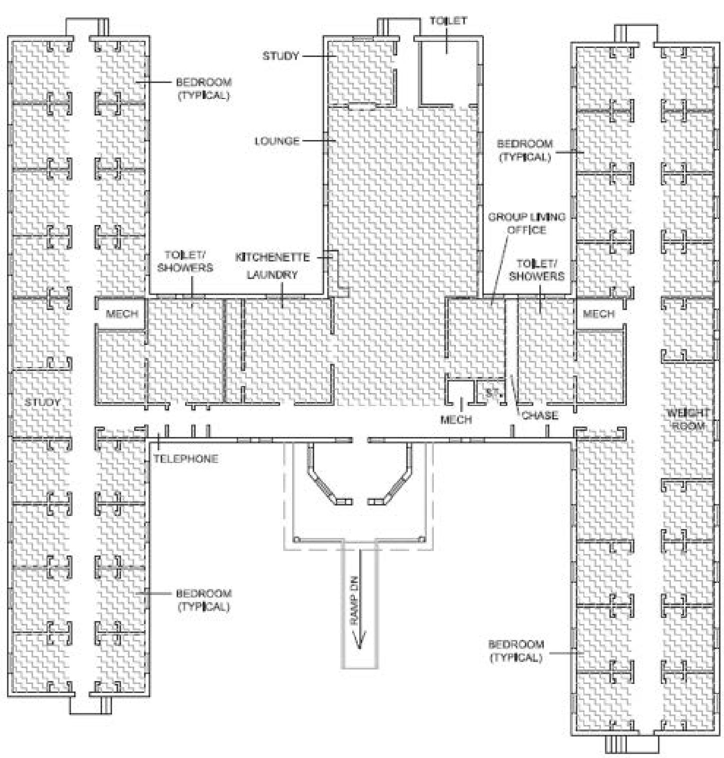
TRAPPER CREEK DORMITORY

DARBY, MT

Background

The Trapper Creek Dormitory is a federal government work campus in rural Montana. Restricted airflow caused summertime overheating and ventilation issues, resulting in tenant discomfort and unsafe $C0_2$ levels in their dormitories. In August 2016, they partnered with Bonneville Power Administration (BPA) and Northwest Energy Efficiency Alliance (NEEA) to determine and implement a cost-effective and energy-efficient HVAC conversion. The system conversion began in February 2017.

This project was one in a series of pilots across the Northwest to help the region better understand the design, installation and expected energy savings of an advanced HVAC solution for small-to-medium commercial buildings. The system approach includes dedicated ventilation air (decoupled from primary heating and cooling air) with high-efficiency heat recovery, a high-efficiency heating and cooling system, and key design principles.



OVERVIEW

Average Gross Floor Area
~11,000 sq. ft. (per dorm)

Starting / Ending EUI
67.9 / 51.5 (Building)
31.6 / 15.3 (HVAC)

Typical Operating Hours
7 days/week, 24 hours/ day lightly loaded during daytime hours

Setpoint/Setback in Heating Season
72° F / 65° F

Setpoint/Setback in Cooling Season
73° F / 78° F

Envelope Thermal Characteristics
Windows: **U-0.5**
Ext. Walls: **R12.5**
Roof: **R30**

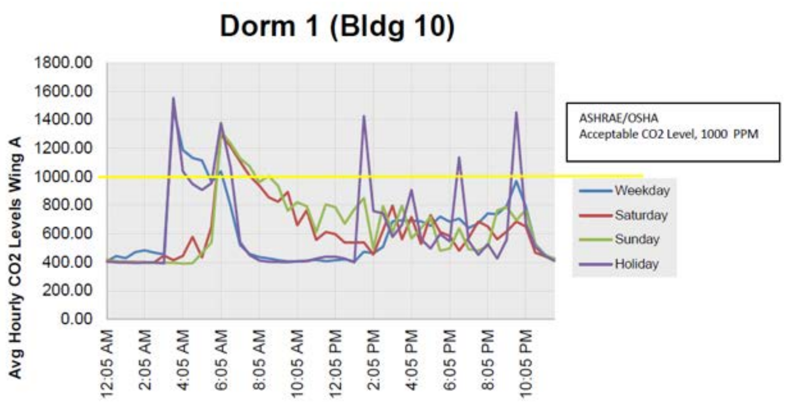
Energy Utility
Ravalli Electric Co-op in partnership with the Bonneville Power Administration

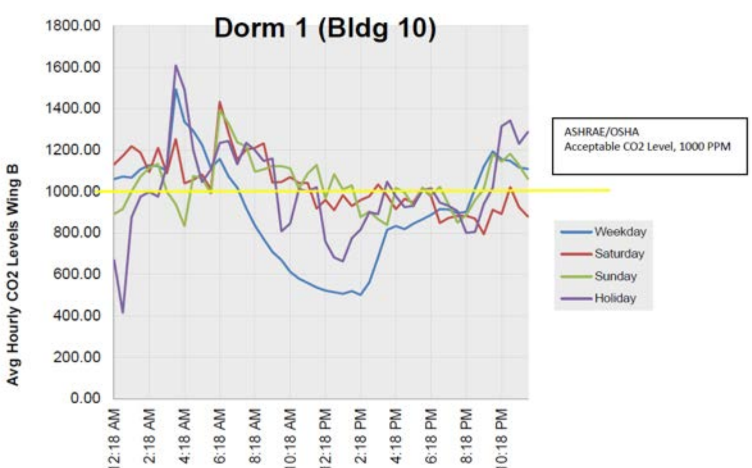
PRE-CONVERSION DETAILS

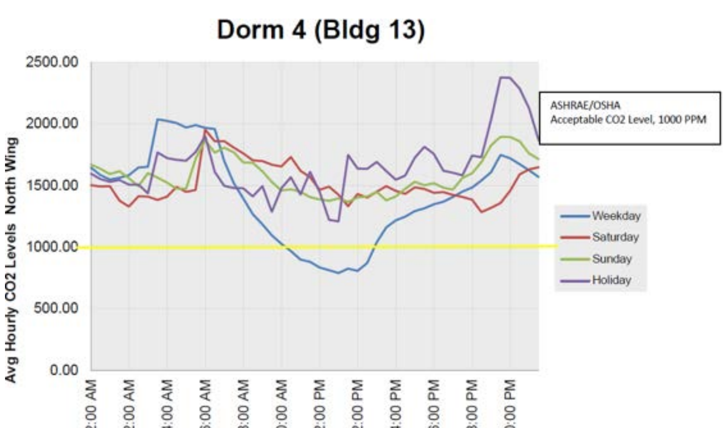
Pre-Conversion CO_2 Levels

A 2011 facility report showed that the lack of airflow in the dormitories resulted in high CO_2 at night when the dorms were fully occupied.

Pre-HVAC conversion sampled CO_2 data, detailed below, shows the weekday working hours pattern with the lowest CO_2 levels occurring during the hours when the occupants are not in the dorms.







Pre-Conversion Air Leakage

Prior to the HVAC conversion, air leakage for all four dorms was tested using the U.S. Army Corps of Engineers test protocols. The results show that air leakage rates among the four dorm buildings were consistent, on average, and that air leakage rates were not excessive.[1]

Air Leakage Test Results – Pre-Conversion			
Test Condition	**Depressurize**	**Pressurize**	**Average**
Enclosure Airtightness *[cfm/ft²@75 Pa]*			
Dorm 1	0.314	0.329	0.321
Dorm 2	0.379	0.394	0.386
Dorm 3	0.332	0.356	0.344
Dorm 4	0.335	0.356	0.346
Equivalent Leakage Area *[ft²@75Pa]*			
Dorm 1	7.46	7.83	7.64
Dorm 2	7.92	8.23	8.08
Dorm 3	7.18	7.70	7.44
Dorm 4	7.95	8.45	8.21
Air Changes *[per hour@50 Pa]*			
Dorm 1	4.00	4.17	4.09
Dorm 2	4.93	5.05	4.99
Dorm 3	4.24	4.48	4.36
Dorm 4	4.20	4.48	4.34
Air Leakage Text Coefficient (C) *[cfm/Paⁿ]*			
Dorm 1	654.3	627.5	N/A
Dorm 2	815.6	717.6	N/A
Dorm 3	605.5	542.8	N/A
Dorm 4	586.3	644.1	N/A
Flow Exponent (n) *[dimensionless]*			
Dorm 1	0.632	0.652	N/A
Dorm 2	0.595	0.633	N/A
Dorm 3	0.641	0.682	N/A
Dorm 4	0.672	0.664	N/A
Squared Correlation Coefficient (r^2) *[dimensionless]*			
Dorm 1	0.999	0.999	N/A
Dorm 2	0.998	0.999	N/A
Dorm 3	0.998	0.999	N/A
Dorm 4	0.999	0.999	N/A

[1]For comparison purposes, the current Washington State Energy Code air-leakage limit for multifamily buildings is 0.4 cfm/ft²@75 Pa.

SYSTEM CONVERSION DETAILS

Split Heat-Pump Systems

Each dorm was previously heated with five electric forced-air furnaces—one for each sleeping wing and one for the central common areas. There was no cooling. Envelope air leakage was relatively low and not sufficient to provide adequate fresh air. There was also a rooftop exhaust fan with a 20-kW make-up air unit serving each dorm's bath and shower area. Airflow for these systems was estimated to be between 800 and 1,200 cfm.

To reduce heating energy use and to solve the summertime cooling issues, BPA's engineering team proposed replacing the five electric forced-air furnace systems in each dorm (in small mechanical rooms) with conventional split heat-pump systems and very high efficiency HRV units.

With BPA's proposal accepted, each dorm received five split heat-pump systems—one for each wing and one for the lounge, bath and laundry areas in the center. The heat-pump systems have capacities of 2 to 4 tons (depending on the areas served), HSPF ratings of 9.0 or higher, and feature electric-resistance back-up. As they are not cold-climate models and have lower rated efficiencies than the systems used in the other Northwest pilot projects, these systems use a significant amount of electric-resistance heating energy, which negatively impacts the post-conversion energy savings.

Heat-Recovery Ventilator (HRV) Systems

In addition to these systems, a Ventacity VS 1000 RT very high efficiency HRV unit was specified for each dorm via new overhead insulated ducting in the attic space. These units are programmed to run at ASHRAE 62.1 ventilation rates during occupied hours (all except normal business hours on weekdays, when the building is unoccupied). The HRVs are physically located outdoors, next to the common areas of the buildings, on the other side of the exterior wall from the lounge, bath and laundry area.

Each HRV is equipped with a CO_2 sensor for the demand-controlled ventilation mode operation. The CO_2 limit was initially programmed for 1,000 ppm. The new fresh-air supply ducting extends only part way down the hallway in each dorm wing to a single diffuser, with an assumed maximum flow of 200 cfm at each supply-air diffuser at full occupancy. Another 100 cfm of fresh air is delivered to the lounge area, and 50 cfm to each of the two restrooms. Exhaust air is extracted from four intakes—two at the wing/common area intersections (220 cfm each) and one in each of the two restrooms (280 cfm each). Ventilation controls were programmed to provide one boost period during the morning shower period, followed by operation at 200 cfm during weekdays when the buildings are unoccupied or lightly occupied.

The HVAC units were placed on stands to elevate them above expected snow levels, with HRV supply and exhaust ducting running from the bottom of the unit and upward for connection to the building near the ceiling level. The 90-degree duct elbows required for these connections have turning vanes to smooth the airflow in these tight turns and lower the external static pressure for the fans. The R-8 duct insulation is on the interior duct surfaces.

Existing System at Each Dorm

System Type	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Electric Forced Air Furnace	(5) Armstrong Air EFC 16MAA-1A for each dorm	N/A	Unknown	1 per AHU
Exhaust Fan	(1) Nutone exhaust fan w/ 20 kW MUA unit for each dorm	N/A	N/A	1

Conversion System at Each Dorm

System Type	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Split System HP	(4) York	3 tons each	3 tons each	1 each
Split System HP	(1) York	4 tons	4 tons	1
HRV	(1) Ventacity VS1000RT	1,025 cfm		

Conversion System Dorm 2

System Type	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Split System HP	(2) York	2 tons each	2 tons each	1 each
Split System HP	(2) York	3 tons each	3 tons each	1 each
Split System HP	(1) York	4 tons	4 tons	1
HRV	1) Ventacity VS1000RT	1,025 cfm		

Conversion System Dorm 3

System Type	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Split System HP	(2) York	2 tons each	2 tons each	1 each
Split System HP	(2) York	3 tons each	3 tons each	1 each
Split System HP	(1) York	4 tons	4 tons	1
HRV	1) Ventacity VS1000RT	1,025 cfm		

Conversion System Dorm 3

System Type	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Split System HP	(4) York	3 tons each	3 tons each	1 each
Split System HP	(1) York	4 tons	4 tons	1
HRV	1) Ventacity VS1000RT	1,025 cfm		

Factoring in engineering and the costs of later modifications to the systems, the total installed project cost was $9.64/sq. ft. before incentives. Of note, the heat-pump systems are residential in nature and do not include significant duct runs or a central control system, which reduces costs compared to larger office or retail spaces.

POST-CONVERSION SYSTEM PERFORMANCE

Post-Conversion Air Leakage

Post-conversion, the leakiest dorm (number 2) became quite a bit tighter with the conversion, and dorm 4 remained about the same. Air leakage rates in dorms 1 and 3 became worse after conversion. Further investigation, with blower doors operating, is planned to track down the sources of the increased air leakage. The same systems were installed in all four dorms, converting the same incumbent systems in all four dorms, by the same contractor. In the end, all four ended up with very similar leakage rates with duct leakage being the most likely source of differences.

Air Leakage Test Results – Post-Conversion			
Test Condition	Depressurize	Pressurize	Average
Enclosure Airtightness *[cfm/ft²@75 Pa]*			
Dorm 1	0.303	0.392	0.348
Dorm 2	0.353	0.326	0.340
Dorm 3	0.347	0.407	0.377
Dorm 4	0.365	0.329	0.347
Equivalent Leakage Area *[ft²@75Pa]*			
Dorm 1	7.18	9.30	8.24
Dorm 2	7.39	6.81	7.10
Dorm 3	7.51	8.80	8.16
Dorm 4	8.65	7.82	8.24
Air Changes *[per hour@50 Pa]*			
Dorm 1	3.85	5.05	4.45
Dorm 2	4.32	4.15	4.24
Dorm 3	4.41	4.99	4.70
Dorm 4	4.51	4.3	4.42
Air Leakage Text Coefficient (C) *[cfm/Paⁿ]*			
Dorm 1	621.4	924.2	N/A
Dorm 2	384.9	556.3	N/A
Dorm 3	586.6	468.9	N/A
Dorm 4	553.1	934.9	N/A
Flow Exponent (n) *[dimensionless]*			
Dorm 1	0.635	0.603	N/A
Dorm 2	0.752	0.648	N/A
Dorm 3	0.658	0.747	N/A
Dorm 4	0.705	0.560	N/A
Squared Correlation Coefficient (r^2) *[dimensionless]*			
Dorm 1	0.995	0.985	N/A
Dorm 2	0.993	0.999	N/A
Dorm 3	1.000	0.985	N/A
Dorm 4	0.998	0.983	N/A

Energy Performance and System Adjustments

The initial performance of the systems did not meet the project owner's or project team's expectations in at least one of the dorms, and it took several months of operation to identify the key issues.

A post-conversion data check in June 2017 revealed that the HRV in the fully monitored dorm (number 4) was not operating at its specified flow rate, even under higher CO_2 concentration conditions. A check of the TAB report suggested some unexpected differences in supply and exhaust fan flows, especially in dorm 1. In particular, exhaust flow rates were lower than specified rates. The project team also received reports that some occupants, especially those at the ends of the dorm wings, were reporting stuffy air conditions, and the facility was having trouble keeping the dorms cool during the hottest weather conditions.

The following problems were found during a system check in dorm 4:

1. The HRV supply fan was not functioning.
2. The HRV air intake filter was significantly clogged with dryer lint.
3. The pressure transducers for the HRV filter pressure differential and supply fan speed control were clogged and not functioning.
4. The schedule for max ventilation flow during shower times was only set for the morning shower period (not accounting for the frequency of late-afternoon showers).
5. Free cooling (economizing) was not implemented in the HRV controls.
6. With exhaust ducting in the equation, maximum HRV exhaust airflow was 800 cfm. With the ducting out of the equation (by sourcing air from directly beside the HRV), airflow increased to 900 cfm.

The project team made the following adjustments in dorm 4 from June 2017 through May 2018:

1. Lint was cleaned out of the HRV and filters were changed
2. The HRV pressure transducers were cleaned.
3. A feature in the dorm thermostat was set up to boost the HRV flow during a second late- afternoon shower period.
4. HRV economizing was implemented at 70 percent of full flow from July 1 through September 1, 3 to 5 a.m.
5. The upper CO_2 limit on the sensor setup was lowered from 1,000 to 900 ppm. This slightly increased ventilation flows and allowed earlier response to rising CO_2 levels.
6. HRV flow was increased from 200 cfm to 300 cfm in schedule period 3—9 a.m. to 4 p.m.).
7. An exhaust intake was added in the dorm shower areas and the exhaust ducting was upsized to lower static pressure and increase flow. (The HRV operates based on its own measured airflow rates, but defaults to the lower of the supply airflow or exhaust airflow. With restricted exhaust airflow in these installations, the supply airflow was also limited to that value in order to maintain neutral pressure relative to outdoors in the buildings. Increasing exhaust airflow under these conditions would also increase supply airflow.)
8. Supply ducting was added to each sleeping wing of each dorm to extend it to the very end of the wings. This helped increase ventilation rates where stuffiness had been reported.
9. To solve the dryer lint problem, supply air intake for the HRV was ducted to move the intake away from the dorm dryer exhaust vents.

Monitoring by the project team, along with observations from facility staff and dorm occupants, revealed that these changes resulted in significant improvements to system performance.

Pre-Conversion Code Minimum w/Cooling (As Modeled)	
EUI:	**67.9** kBtu/sq. ft./yr.
Fans:	**9.2** kBtu/sq. ft./yr.
Heating:	**21.4** kBtu/sq. ft./yr.
Cooling:	**1.1** kBtu/sq. ft./yr.
HVAC:	**31.6** kBtu/sq. ft./yr.

Post-Conversion (As Modeled)		
EUI:	**51.5** kBtu/sq. ft./yr.	24% Reduction
Fans:	**2.9** kBtu/sq. ft./yr.	68% Reduction
Heating:	**11.3** kBtu/sq. ft./yr.	47% Reduction
Cooling:	**1.1** kBtu/sq. ft./yr.	No change
HVAC:	**15.3** kBtu/sq. ft./yr.	52% Reduction

Pre- and post-conversion modeled energy consumption are based on a typical meteorological year (TMY). Models updated based on a full year of sub-metered energy end-use data.

The tables and graphs below show the TMY-based pre- and post-conversion energy use.

TYPICAL YEAR PRE-CONVERSION MODEL

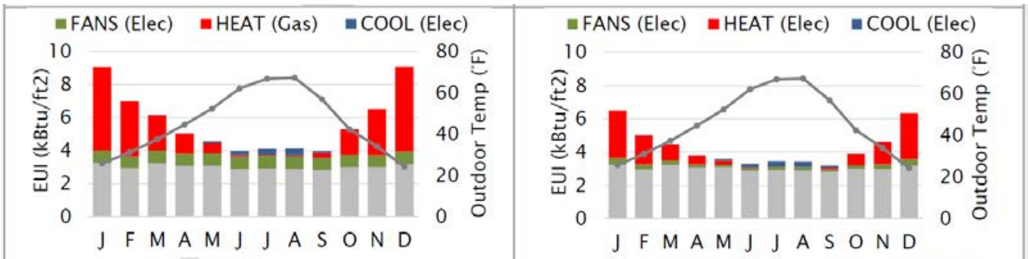
	OTHER kBtu/ft²	FANS kBtu/ft²	HEAT kBtu/ft²	COOL kBtu/ft²	TOTAL kBtu/ft²
JAN	3.2	2.5	6.7	0.0	12.5
FEB	2.9	2.3	5.0	0.0	10.2
MAR	3.2	2.5	4.7	0.0	9.7
APR	3.1	2.4	2.3	0.0	7.8
MAY	3.1	2.5	1.2	0.2	7.0
JUN	2.9	2.4	0.2	0.4	6.0
JUL	2.9	2.5	0.1	0.7	6.2
AUG	2.9	2.5	0.1	0.7	6.2
SEP	2.8	2.4	0.5	0.2	6.1
OCT	3.0	2.5	2.8	0.0	8.3
NOV	3.0	2.4	4.7	0.0	10.1
DEC	3.2	2.5	7.0	0.0	12.7
ELEC	36.2	29.8	34.6	2.3	102.8
GAS	-	-	-	-	-

TYPICAL YEAR POST-CONVERSION MODEL

	OTHER kBtu/ft²	FANS kBtu/ft²	HEAT kBtu/ft²	COOL kBtu/ft²	TOTAL kBtu/ft²
JAN	3.2	0.4	2.9	0.0	6.5
FEB	2.9	0.3	1.8	0.0	5.0
MAR	3.2	0.3	1.0	0.0	4.5
APR	3.1	0.2	0.5	0.0	3.8
MAY	3.1	0.2	0.2	0.1	3.6
JUN	2.9	0.2	0.0	0.2	3.3
JUL	2.9	0.2	0.0	0.3	3.4
AUG	2.9	0.2	0.0	0.3	3.4
SEP	2.8	0.1	0.1	0.1	3.2
OCT	3.0	0.2	0.7	0.0	3.9
NOV	3.0	0.3	1.3	0.0	4.6
DEC	3.2	0.4	2.8	0.0	6.3
ELEC	36.2	2.9	11.3	1.1	51.5
GAS	-	-	-	-	-
SAVINGS	0.0	26.9	23.3	1.2	51.3
%	-	90%	67%	52%	50%

Annual Performance Data

Using the building models calibrated with post-conversion energy bills and system monitoring, the figures below show system annual and monthly energy use data.

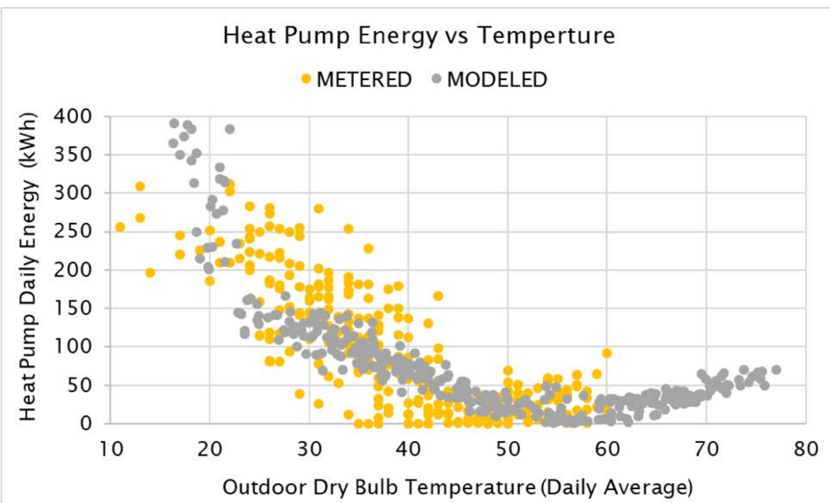


	ANNUAL EUI	---		ANNUAL EUI	ANNUAL SAVINGS
Total:[2]	67.9 kBtu/ft²	---	Total:	51.5 kBtu/ft²	16.4 kBtu/ft²
Fans:	9.2 kBtu/ft²	---	Fans:	2.9 kBtu/ft²	6.3 kBtu/ft²
Heating:	21.4 kBtu/ft²	---	Heating:	11.3 kBtu/ft²	10.1 kBtu/ft²
Cooling:	1.1 kBtu/ft²	---	Cooling:	1.1 kBtu/ft²	0.0 kBtu/ft²
HVAC:	31.6 kBtu/ft²	---	HVAC:	15.3 kBtu/ft²	16.4 kBtu/ft²
Electricity:	67.9 kBtu/ft²	---	Electricity:	51.5 kBtu/ft²	16.4 kBtu/ft²

2 Minor additive discrepancies are due to rounding.

Heat-Pump Energy Performance

The heat-pump systems used in the project are a relatively conventional type that exhibited good efficiency. On average, the models capably track the field monitoring data, however, there are a couple of anomalous sets of conditions where the models do not accurately predict actual energy use.



The modeled performance curves are typically based on current test and rating procedures that do not take into account actual system behaviors in certain ambient conditions. System COPs are assumed to vary directly with the difference between outdoor ambient temperature and indoor temperature, with compressor capacity and efficiency declining smoothly as outdoor ambient temperature declines.

Actual system behavior, on the other hand, is not so predictable. The anomalies can be seen in the differences between the modeled curve and the metered data above. The 25–40 F temperature range is the defrost range, and current test and rating procedures significantly underestimate defrost impacts on system performance. This is in part because the system's diminished low-temperature capacity, which is delivered only during the fraction of the hour not devoted to the defrost cycles, may struggle to keep up with building load.

The impact of the very high efficiency HRV is shown in the 40–50 F range, as it frequently recovers enough energy to keep the system off during a greatly expanded number of hours.

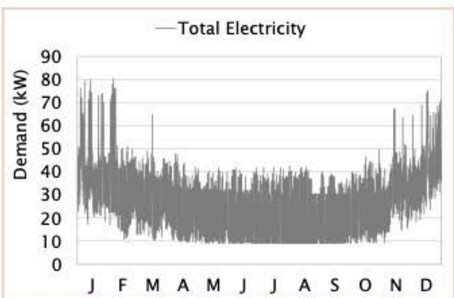
Overall, the average system performances (modeled and actual) over the entire ambient temperature range are reasonably well-aligned.

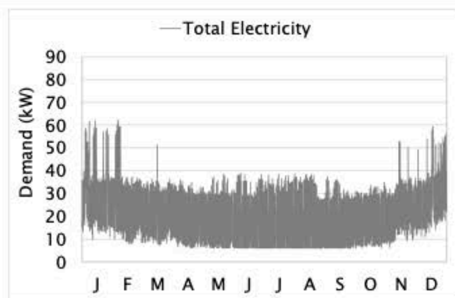
Post-Conversion Demand Impacts

Unlike in other Northwest pilot projects, the conversion systems were required by the owners' purchasing rules to be conventional heat-pump systems rather than cold-climate models that maintain their capacity down to relatively low outdoor ambient temperatures. Cold-climate models could significantly improve energy savings and result in demand savings up to 45 percent in the winter months.

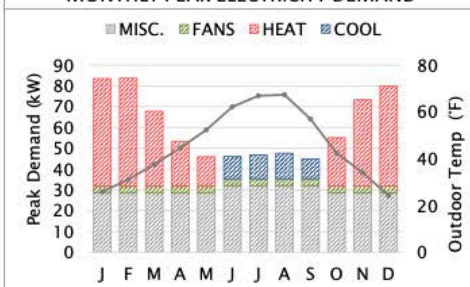
In this project, there is still significant electric-resistance energy use, including during defrost cycles. Because there are a significant number of heating hours where electric resistance is used, both energy savings and demand savings are lower for this project than they would be with a true cold-climate heat-pump system. Despite this, the demand savings are very good during the heating months in comparison to a code-minimum electric-resistance HVAC system.

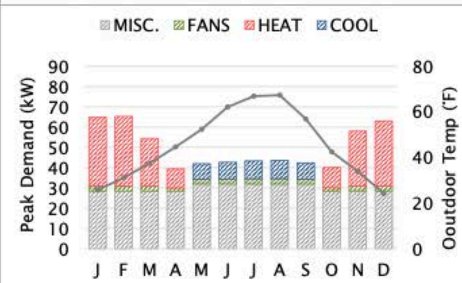
PRE-CONVERSION HVAC SYSTEM with NEW CODE MINIMUM EQUIPMENT	POST-CONVERSION HVAC SYSTEM with NEW DOAS HRV + VRF SYSTEM
ELECTRICITY DEMAND	ELECTRICITY DEMAND





MONTHLY PEAK ELECTRICITY DEMAND | MONTHLY PEAK ELECTRICITY DEMAND





	PEAK DEMAND kW	PEAK OCCURENCE (DARBY TMY)
JAN	84	14-JAN-07:00
FEB	84	06-FEB-07:00
MAR	68	17-MAR-06:00
APR	52	09-APR-06:00
MAY	45	02-MAY-06:00
JUN	45	29-JUN-17:00
JUL	46	23-JUL-17:00
AUG	46	01-AUG-17:00
SEP	44	03-SEP-17:00
OCT	54	30-OCT-06:00
NOV	73	15-NOV-07:00
DEC	80	18-DEC-07:00

	PEAK DEMAND kW	PEAK OCCURENCE (DARBY TMY)
JAN	65 (↓22%)	14-JAN-07:00
FEB	65 (↓22)	06-FEB-07:00
MAR	54 (↓20%)	17-MAR-06:00
APR	39 (↓24%)	09-APR-06:00
MAY	42 (↓7%)	22-MAY-17:00
JUN	43 (↓5%)	29-JUN-17:00
JUL	43 (↓5%)	16-JUL-17:00
AUG	44 (↓6%)	01-AUG-17:00
SEP	42 (↓4%)	03-SEP-17:00
OCT	40 (↓25%)	30-OCT-06:00
NOV	58 (↓21%)	15-NOV-07:00
DEC	63 (↓21%)	18-DEC-07:00

→ Largest peak demand savings are seen in shoulder season because peak savings are reduced when both systems required electric resistance heating to satisfy heating load in colder months.

ADDITIONAL FINDINGS

In addition to the above, the project team found a variety of noteworthy lessons, outcomes and challenges unique to this conversion project.

- This HRV ducting, for both the fresh-air supply ducting and the exhaust/extract ducting, was not optimal. The supply ducting was run in the attic space over the hallways in each wing, but stopped at a point about halfway down the hall. Reports from the occupants strongly suggested that the sleeping spaces in the far half of each dorm wing were not getting much fresh air at all. This is a common mistake made when designing a ventilation-only system for the first time—especially one that runs most of the time without any conditioning of the ventilation air. Best practice in these systems is to provide the fresh air at one side or end of a space, and to extract stale air at the opposite side or end of the space. Otherwise, the ventilation air simply reverts back to the exhaust intakes without providing any benefit to occupants of the space beyond the fresh-air drops.
    - This situation was later remediated by extending the fresh-air supply ducting all the way to the end of the wings.
- The exhaust/extract ducting for the common areas were undersized and overly restrictive, which limited airflow relative to supply flows. This led to complaints of elevated humidity and the potential for mold and mildew growth in the shower areas.
    - This was remediated by enlarging and adding exhaust ducting, and implementing a control strategy more likely to ensure that the system always ran long enough to remove the bulk of the moisture from the shower and bath areas during times when they are most used.
- While the exterior ducting does have some insulation, the substantial amount of exterior ducting, combined with Montana's comparatively extreme winter conditions, compromises the energy recovery performance of the HRVs. Generally, limited duct insulation does not impact performance, but in a very efficiency HRV system, the loss of a few degrees in both supply and exhaust air temperature significantly impacts heat exchange efficiency. This can result in much lower delivered air temperatures during the winter, and higher delivered air temperatures during the summer, which adversely impacts occupant comfort.
    - For the same reasons, duct leakage is also critically important.
- Exterior ducting should be minimized wherever possible, and, where its use is necessary, heavily insulated and fully air-sealed. Supply and exhaust air temperature changes due to exterior ducting losses should be taken into account when determining when or if supplemental heating or cooling of supply air will be needed.
- The HRV airflow rates were greatly affected by the elevation of more than 4,000 feet above sea level. As the air is thinner, the fans will deliver less air at any given speed based on the system's own sensors. For instance, a unit that is rated to deliver 1,000 cfm at sea level will deliver only about 900 cfm at this higher elevation. This difference should be taken into account in the system design.
    - High elevation also diminishes heat pump capacity.

BETTERBRICKS
Powerful Energy Ideas. Delivered by NEEA.

PILOT REPORT

URBAN RESTAURANT

PORTLAND, OR

Background

Verde Cocina is a Portland, Ore., restaurant located within a historic building. The small restaurant's dining room is less than 900 sq. ft., with a kitchen area of 250 sq. ft.

The restaurant owners reported patron comfort issues during both summer and winter, especially near the large roll-up door that comprises most of the street façade, which is the only exterior wall defining the space. This led the property management company to undergo an HVAC system conversion.

This project was one in a series of pilots across the Northwest to help the region better understand the design, installation and expected energy savings of an advanced HVAC solution for small-to-medium commercial buildings. The system approach includes dedicated ventilation air (decoupled from primary heating and cooling air) with high-efficiency heat recovery, a high-efficiency heating and cooling system, and key design principles.

PRE-CONVERSION DETAILS

Pre-Conversion System Details

The original heating, cooling and ventilating functions were provided by an aging 3-ton rooftop unit (RTU) with gas heat. The fresh air intake on the unit was blocked, meaning no fresh air was being delivered, at any time, by the HVAC system.

Pre-Conversion Air Leakage

The team learned from a former restaurant pilot project that not much could be learned about the airtightness characteristics of a space so small, in a building so large, in the presence of a large vent hood in the kitchen. Therefore, no airtightness testing was done for this project.

OVERVIEW

Conditioned Floor Area
1,147 sq. ft.

Starting/Ending EUI
874.5 / 701 (Building)
238.8 / 65.3 (HVAC)

Typical Operating Hours
Sunday-Wednesday,
9 a.m. to 11 p.m.
Thursday-Saturday,
9 a.m. to 12 a.m.

Setpoint/Setback in Heating Season
70° F / 65° F

Setpoint/Setback in Cooling Season
74° F / 78° F

Envelope Thermal Characteristics
Windows: **U-0.6**
Ext. Walls: **R3**
Roof: **R8**

Energy Utility
Portland General Electric and NW Natural

SYSTEM CONVERSION DETAILS

After encountering complications and setbacks, as detailed in the Additional Findings section below, the project was delayed by more than a year. The heating/cooling system conversion ultimately took place the summer of 2017. Then, the Ventacity Systems VS 1000 RT ventilation unit was installed in December 2017 and commissioned in January 2018.

The original RTU was removed and the Ventacity HRV was set in place. The new 3-ton heat pump system has two 15 kBtu/hr. wall units and one 18 kBtu/hr. wall unit, each located in such a way as to be able to direct air to the areas with most of the heating and cooling load, and to mix the air without causing drafts or unwanted air currents for the diners in the space.

In most pilot projects, the project team was able to considerably downsize the heating/cooling system in the conversion process. In this case, there were existing comfort complaints about inadequate heating and cooling, and the internal gains are atypically large for such a small floor area. So, the team chose to select a conversion system with 3 tons of cooling capacity. System details are shown in the table below.

	Number / Make / Model	Cooling Capacity	Heating Capacity[1]	Zones
Existing System Type				
RTU	(1) Ruud URKA A036JK08E	3 tons	5.3 tons	1

	Number / Make / Model	Cooling Capacity	Heating Capacity	Zones
Conversion System				
Multi-zone DHP	(1) Daikin 4MXS36NMVJU	3 tons	3 tons	3
Packaged HRV	(1) Ventacity VS1000RT	1,025 cfm		1

As shown in the photo to the left, the existing curb for the original RTU was relatively small. The curb adapter required careful redirection of the HRV airflow without imposing significant external static pressure. As in all exterior ducting, careful attention to detail in airflow restrictions, air-sealing, and insulation are required to enable optimal performance of the ventilation system.

The heat pump outdoor unit, with its relatively small footprint and limited power requirements, was easy to locate near the HRV. The team was able to take advantage of existing through-the-roof penetrations for wiring and refrigerant-line connections.

[1] For natural gas-fired systems to the building after combustion losses (80% AFUE assumed).

The existing evaporative cooler make-up air unit (MUA) for the kitchen range hood was left in place. After the significant fan power reductions that came with the new HVAC system, monitoring data showed that the range hood and its swamp cooler MUA now use most of the restaurant's fan power.

Most of the existing RTU ducting was re-purposed for the ventilation function, with only minor modifications required. This simplified the installation and kept project costs down.

POST-CONVERSION SYSTEM PERFORMANCE

The project delays and complications led to a significant shift in the post-conversion monitoring period.[2] The models and analysis here are based on 6 months of pre-conversion data in 2016 and 12 months of post-conversion data, with more than a year for the conversion process in between.

Pre-Conversion Code Minimum (As Modeled)		Post-Conversion (As Modeled)		
EUI:	**874.5** kBtu/sq. ft./yr.	EUI:	**701** kBtu/sq. ft./yr.	20% Reduction
Fans:	**49.6** kBtu/sq. ft./yr.	Fans:	**32.7** kBtu/sq. ft./yr.	34% Reduction
Heating (gas):	**181.0** kBtu/sq. ft./yr.	Heating:	**25.9** kBtu/sq. ft./yr.	86% Reduction
Cooling:	**8.3** kBtu/sq. ft./yr.	Cooling:	**6.6** kBtu/sq. ft./yr.	20% Reduction
HVAC:	**238.8** kBtu/sq. ft./yr.	HVAC:	**65.3** kBtu/sq. ft./yr.	73% Reduction
Other:	**635.7** kBtu/sq. ft./yr.	Other:	**635.7** kBtu/sq. ft./yr.	N/A

Pre- and post-conversion modeled energy consumption are based on a typical meteorological year (TMY). Models updated based on a partial year of sub-metered energy end-use data (Aug.-Feb.).

Data and owner feedback during the post-conversion monitoring period suggest that the new systems are working very well. The savings are substantial, though a large fraction of annual energy use in this restaurant is non-HVAC related. The largest loads are cooking and hot water, and base loads of 500 kBtu/sq. ft. or more are not uncommon.

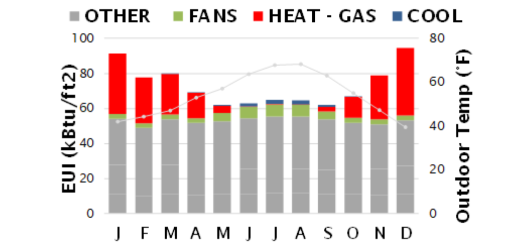
Restaurants with small dining areas and/or a large fraction of take-out meals have very high base load EUIs (large process loads, small conditioned floor areas), but HVAC savings of more than 100 kBtu/sq. ft. are impressive in any setting. The base load EUI for this project is 635.7 kBtu/sq. ft.

As experienced by the team, thermal comfort and indoor air quality after conversion were excellent and all system airflows were silent and unobtrusive, even in the very small space with relatively high occupancy per square foot (i.e., with relatively high airflow rates).

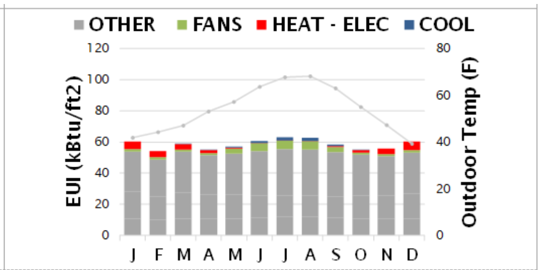
[2] A mid-winter 2019 water pipe break in the building flooded the restaurant out of operation for a short period of time, but twelve months of post-conversion data was collected, ending the month before the pipe break.

Annual Performance Data

Using the building models calibrated with post-conversion energy bills and system monitoring, the figures below show system annual and monthly energy use data.



	ANNUAL EUI	---
Fans:	49.6 kBtu/ft²	---
Heating:	181.0 kBtu/ft²	---
Cooling:	8.2 kBtu/ft²	---
HVAC:	238.8 kBtu/ft²	---
Electricity:	189.8 kBtu/ft²	---
Gas:	684.7 kBtu/ft²	---
Total:	874.5 kBtu/ft²	---



	ANNUAL EUI	ANNUAL SAVINGS
Fans:	32.8 kBtu/ft²	16.8 kBtu/ft²
Heating:	25.9 kBtu/ft²	155.2 kBtu/ft²
Cooling:	6.7 kBtu/ft²	1.5 kBtu/ft²
HVAC²:	65.3 kBtu/ft²	173.5 kBtu/ft²
Electricity:	197.3 kBtu/ft²	-7.5 kBtu/ft²
Gas:	503.7 kBtu/ft²	181.0 kBtu/ft²
Total:	701.0 kBtu/ft²	173.5 kBtu/ft²

Post-Conversion Demand Impacts

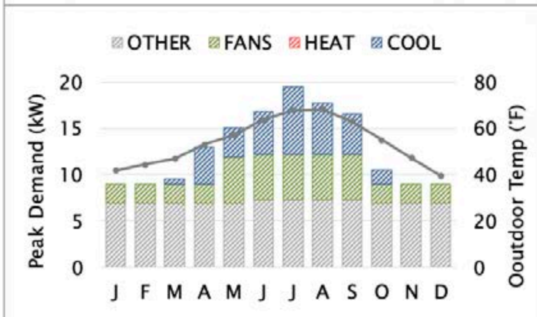
Data and owner feedback during the post-conversion monitoring period suggest that the new systems are working very well. The savings are substantial, though a large fraction of annual energy use in this restaurant is non-HVAC related. The largest loads are cooking and hot water, and base loads of 500 kBtu/sq. ft. or more are not uncommon.

Restaurants with small dining areas and/or a large fraction of take-out meals have very high base

PRE-CONVERSION HVAC SYSTEM with NEW CODE MINIMUM EQUIPMENT

ELECTRICITY DEMAND

MONTHLY PEAK ELECTRICITY DEMAND

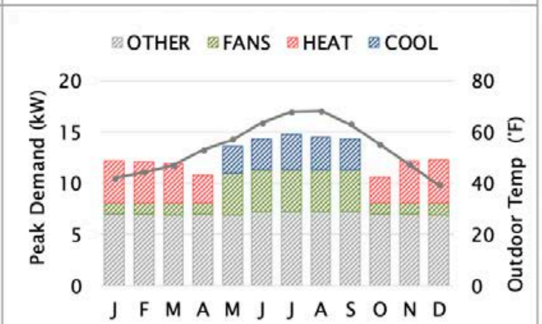


	PEAK DEMAND kW	PEAK OCCURRENCE
JAN	9	01-JAN-22:00
FEB	9	23-FEB-09:30
MAR	10	11-MAR-09:30
APR	13	18-APR-20:00
MAY	15	31-MAY-16:30
JUN	17	23-JUN-16:30
JUL	19	21-JUL-15:00
AUG	18	10-AUG-16:00
SEP	17	09-SEP-15:00
OCT	11	29-OCT-20:00
NOV	9	30-NOV-09:30
DEC	9	01-DEC-22:00

POST-CONVERSION HVAC SYSTEM with NEW DOAS HRV + VRF SYSTEM

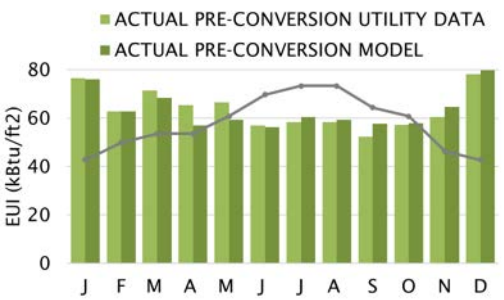
ELECTRICITY DEMAND

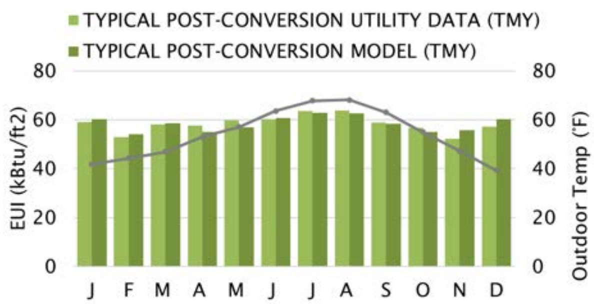
MONTHLY PEAK ELECTRICITY DEMAND



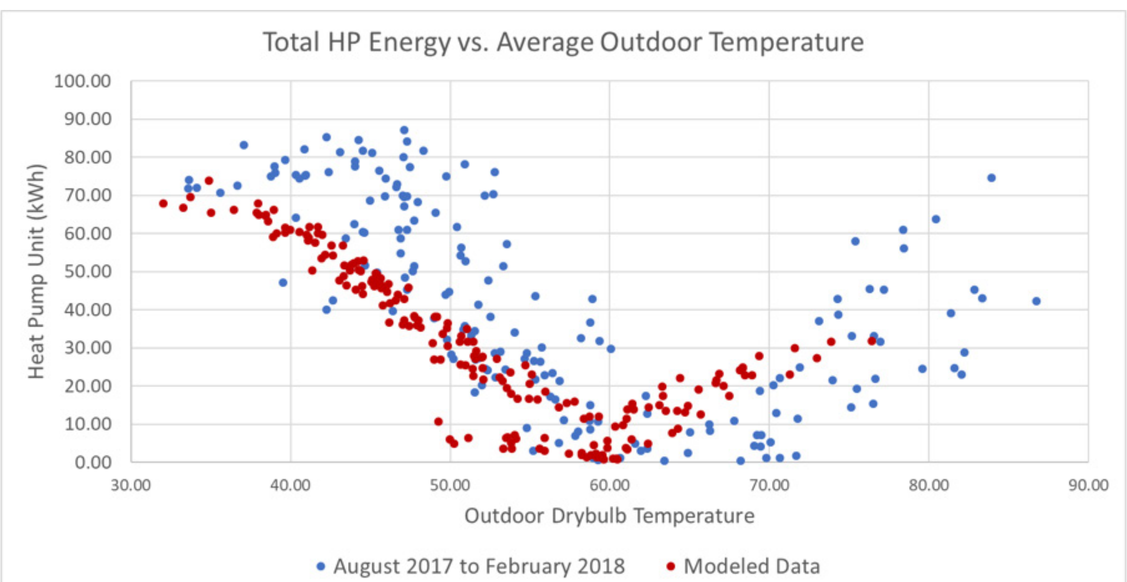
	PEAK DEMAND kW	PEAK OCCURRENCE
JAN	12 (↑ 36%)	01-JAN-19:00
FEB	12 (↑ 35%)	03-FEB-18:00
MAR	12 (↑ 25%)	07-MAR-11:30
APR	11 (↓ 17%)	18-APR-20:00
MAY	14 (↓ 10%)	31-MAY-16:30
JUN	14 (↓ 15%)	23-JUN-16:00
JUL	15 (↓ 24%)	21-JUL-15:00
AUG	14 (↓ 18%)	10-AUG-17:30
SEP	14 (↓ 13%)	09-SEP-15:30
OCT	11 (↑ 1%)	29-OCT-20:00
NOV	12 (↑ 36%)	09-NOV-20:00
DEC	12 (↑ 37%)	06-DEC-11:30

Energy model calibration results were quite good. However, the high energy baseload of over 600 kBtu/sq. ft. is common for a restaurant, and much higher than the energy use of the new systems after conversion. The models assume the same base loads for pre- and post-conversion calibration, so what appear in the monthly model EUI comparison data as small differences are created by much larger HVAC reduction percentages than seen in the non-restaurant pilot projects.





The scatter plot below shows the modeled performance of the new heating and cooling system under the range of ambient conditions (in red) versus the metered performance (in blue), during a typical 7 months of operation. The metered data is much more scattered than the model predicted, especially in the heating mode, under ambient temperature conditions that are often likely to have exceeded the balance point temperature. The cooling predictions were likely thrown off not only by variable occupancy, but also by highly variable solar gains from the west-facing glazing in the only exterior wall of the space. To add another complication, this glazing is located in a large roll-up garage door that was probably operated by staff in a more or less random fashion in milder weather.



Overall, the project team and the restaurant owners were quite pleased with both the energy and comfort performance of the new systems. For a restaurant project, the energy savings are impressive, albeit challenging to predict or explain.

Project costs, including permitting, and before incentives, were $30.99/sq. ft. The proposed cost for the base case alternative—a brand-new 3-ton RTU with gas heating and economizer—was $11.14/sq. ft., with the same square footage and occupancy caveats described above. Note that this price is for the lowest priced unit of its size available on the market.

ADDITIONAL FINDINGS

In addition to the above, the project team found a variety of noteworthy lessons, outcomes and challenges unique to this conversion project.

- This project is one of two restaurants in the pilot portfolio. Both had very limited floor area and very high process loads (vent hood, refrigeration, hot water) per square foot. A restaurant's dining square footage relative to its meals served, more than any other single factor, determines the EUI for the space. The numbers seem high here, even for a restaurant, but they are lower than for the other pilot restaurant project, in part because that restaurant had a high proportion of take-out business (process loads that require no dining square footage to service).

- As mentioned above, this project faced many delays and complications, including:
    - Typical of small projects where a single system (often an RTU) is replaced on the roof by two separate systems (in this case, the heat pump outdoor unit and the HRV unit), the existing system occupied the roof curb where the HRV would be located, and the HRV would use the power circuit serving the RTU. The heat pump outdoor unit had to sit nearby, on the roof, and be operated with a new circuit installed at the electrical panel and a new conduit run to the roof.
    - An electrician deemed the electric panel insufficiently accessible for adding another circuit due to the walk-in cooler located within 16 inches of the front of the panel. Some project delay occurred while the restaurant owners removed the walk-in cooler and replaced it with a packaged commercial refrigeration unit with a smaller footprint.
    - After the refrigeration equipment change-out, the electrician learned that the main cut-off for the whole electric panel was located in another store in the building. This additional code issue had to be remedied before the panel could be worked on.
    - New code requirements went into effect requiring a complete roof plan of the whole building identifying all of the existing and proposed new equipment for the roof. Another set of provisions limited the total number of pieces of equipment on the roof and limited the equipment color to a few common colors. In a multi-tenant building, this can be a significant burden, especially for a project that will typically either not increase the amount of equipment on the roof, or more often, decrease it. It was necessary to obtain several permits from the city in order to set up and utilize a crane to lift the old equipment off the roof and put the new equipment on it. Acquiring the necessary city permits delayed the project for several more months. By now the project was more than a year behind schedule.

- As noted in the System Conversion Details above, most of the existing RTU ducting was converted to provide ventilation air. This can often be done where an RTU serves a single zone, particularly if the zone consists of a large open space with multiple duct branches and diffusers. In an installation with multiple zones, with multiple individual enclosed spaces, it is generally advisable (and more cost-effective) to use the existing ducting, with its balanced supply and return ducting, for the heating and cooling function, while using ducted air handlers in the VRF system rather than individual ductless indoor units in each space. This is because the individual ductless units are frequently significantly oversized for the loads they serve, and, when added together, the sum of their capacities results in a significantly oversized outdoor capacity as well.

- This project offered the ideal situation for utilizing existing heating/cooling ducting for the ventilation function because of the single large space being served by the multi-zone mini-split heat pump system and the HRV. All three of the heat pump indoor units are ductless, properly sized for the loads being served. It's also important to note that the existing ducting, sized for the existing RTU's 1,200 cfm nominal airflow rate, was properly sized for the HRV's 1,000 cfm maximum airflow rate, under low external static pressure conditions.

- Modeling heat pump energy use is always a challenge as typical modeled performance curves are derived from sub-par testing and rating procedures. Restaurants are especially challenging to model because of the highly variable occupancy and because assumptions must be made about the airflow balance between the range hood fan(s) and the MUA unit fan(s). The fan balance assumptions may or may not reflect reality in the field.

- It is important to note that costs can vary by building type and depend on existing conditions and local code requirements. For example, a system conversion of this size would capably serve a typical office occupancy twice as large as this restaurant—in that case the cost per square foot could be up to 30–40% lower (as low as $20/sq. ft.).