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Performance of Water-Source Variable Refrigerant Flow

Measurement and Verification of Two Installed Systems

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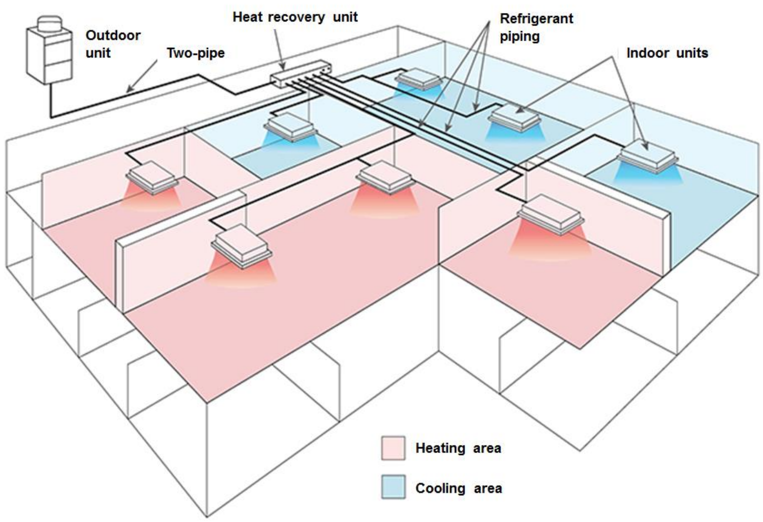
ACRONYMS

BAS	Building automation system
COP	Coefficient of performance
DOAS	Dedicated outside air system
EER	Energy efficiency ratio
EUI	Energy utilization index
GHX	Ground heat exchanger
GS	Ground-source
GSHP	Ground-source heat pump
HVAC	Heating, ventilation, and air conditioning
IEER	Integrated energy efficiency ratio
OA	Outside air
RTU	Rooftop unit
SHGC	Solar heat gain coefficient
VAV	Variable air volume
VRF	Variable refrigerant flow
WS	Water-source
WSHP	Water source heat pump

INTRODUCTION

Variable Refrigerant Flow (VRF) systems have developed into a promising emerging technology. While popular in some places in the world, these systems are quite new to the upper Midwest. The systems are an innovative version of a simple split system air conditioner that utilizes variable speed compressors, multiple zone refrigerant distribution, heat recovery, and low energy fan coils to cool and heat commercial buildings more efficiently than standard split systems and heat pumps. The key components of a VRF system are the outdoor unit, indoor unit, refrigerant, and heat recovery unit, typically laid out as shown in Figure 1. Refrigerant moves throughout the entire system, with both hot and cold refrigerant being produced by the outdoor unit. Both phases of refrigerant are sent to the heat recovery units where, in addition to heat recovery, the controls designate each indoor unit to receive hot or cold refrigerant based on whether they need heating or cooling. An additional, separate system, also called a dedicated outdoor air system (DOAS), is needed to serve the ventilation requirements of the spaces.

Figure 1: Variable refrigerant flow system with supply piping only shown; return piping hidden.



VRF systems save energy in four primary ways:

- **Distribution of heating/cooling using refrigerant instead of air.** Between the energy density and phase change capability of refrigerant, it is several orders of magnitude more efficient at moving heat than air.
- **Variable speed compressors (and fans)**. The inverter-driven compressors in VRF heat pump units are fully variable in speed and capacity. The fans are also variable speed.
- **Zone-level heating and cooling**. VRF systems provide heating only to zones that need heating, and cooling only to zones that need cooling, generally leading to a more efficient overall system. This generally leads to energy savings due to avoided overcooling and reheating that is common in typical multizone HVAC.
- **Recovery of heat from cooling zones to heating zones.** Some VRF systems (including those we studied) also have the ability to directly transfer refrigerant from a cooling zone to a heating zone, resulting in heat being transferred between zones without the aid of a compressor. This is accomplished using heat recovery units between the outdoor and indoor unit as shown in Figure 1.

Estimates in literature, primarily based on modeling, suggest anywhere from 10 to 50% savings from VRF systems compared to traditional HVAC. There is very little field research that has been conducted to measure savings from these systems. And the little field research that does exist is primarily for air-source (or “air-cooled”) VRF systems in warmer climates (Georgia, Alabama, etc.). Most VRF systems are air-source, meaning the outdoor unit is cooled (or heated) by air blowing across an air-to-refrigerant heat exchanger. But these systems perform much better in milder climates because their capacity decreases rapidly at very low ambient temperatures (beginning somewhere between 15ºF and -10 ºF). In colder climates these systems often use supplemental heat to provide heat at these temperatures. Alternately, the system can be upgraded to water or ground-source, thereby saving even more energy in colder climates, but at significant additional first cost. But performance of VRF systems is even less understood than that of air-source systems.

Objectives

To fill this knowledge gap, we have completed a study of the energy savings from two water-source VRF systems. By measuring, in detail, the performance of these systems we sought to meet the following objectives:

- Establish recorded performance of a whole VRF system in a cold climate (the upper Midwest).
- Calculate energy performance values (COP) for individual VRF units as well as for the entire system (i.e. a system COP).
- Calibrate an energy model of the VRF systems and use to compare with more conventional HVAC approaches.
- Establish lessons-learned from design, construction, and operation of water-source VRF systems.

We have completed the field monitoring and analysis, and we’ve calculated the energy performance of the systems as well as their impact on economics and emissions. In addition, we’ve been able to discover several best practices along the way after observing system operation and talking with stakeholders. This report first briefly discusses how and where we completed this work, and then discusses each of those results in turn.

RESEARCH APPROACH

The research study began with a review of VRF performance in literature and identification of some actual system applications to study. We then monitored each system for a period of time, and analyzed the results based on typical performance metrics and calibrated simulation models.

LITERATURE REVIEW

There have been some studies on air-source VRF systems, including those based solely on modeling (Liu, 2010 and Zhou, 2007) as well as some based on field measurement, including Southard, 2014, Gray, 2015, Swanson, 2015.

But there has been relatively limited effort studying water-source systems. Piljae, 2014 is one such study, in which a single, but complex, building with water-source VRF was monitored for a period of time and its performance analyzed and published in terms of standard metrics. Piljae, 2014 is the one other data point available that is similar to the two analyzed here, so in this report we will make comparison of the results of our study to that of the system studied by Piljae.

THE BUILDINGS AND SYSTEMS

In order to study the performance of VRF in a real cold climate building environment, we first located buildings with water-source VRF systems in the upper Midwest. Since this is a newer system type, after a significant search of Wisconsin and Minnesota, we only found four such buildings. Based on the utilities willing to fund the research[1] we chose one building in each of two areas. The first was 749 University Row in Madison, WI ("Madison Office") and the second was 11001 N Hampshire Avenue in Minneapolis, MN ("Minneapolis Office"). Both have water-source VRF systems served by ground heat exchangers (GHXs). More details on these two buildings are given in the two sections below.

Madison Office

The office building that we studied in Madison, WI is known by its address, 749 University Row (see Figure 2). This is a three story, 85,000 ft2 multi-tenant office building with seven tenants, varying in activity from building design to dentistry to financial services. Three above-ground floors are essentially entirely tenant-occupied space, other than bathrooms and a small hallway area. Figure 3 shows a layout of one of the three above-ground floors. A single below-grade level contains parking, storage, and locker rooms.

The building is steel-framed construction with a higher performance envelope. Insulation is composed of purely rigid board insulation, exterior to the structure for both roof (five inches) and walls (three inches). A spray-on air barrier was also applied. Windows comprise 39% of the exterior surface area, and are thermally broken, dual-pane, low-e units (U-value = 0.32, SHGC = 0.28).

Internal loads vary from tenant to tenant. Most tenants utilize primarily fluorescent lighting supplemented with some LED (for overall power densities of between 0.7-0.85 W/ft^2), and all have typical office equipment. The dentist does have additional equipment in their space such as imaging and sterilizing equipment, but its use has been measured to be infrequent enough as to yield a similar annual energy

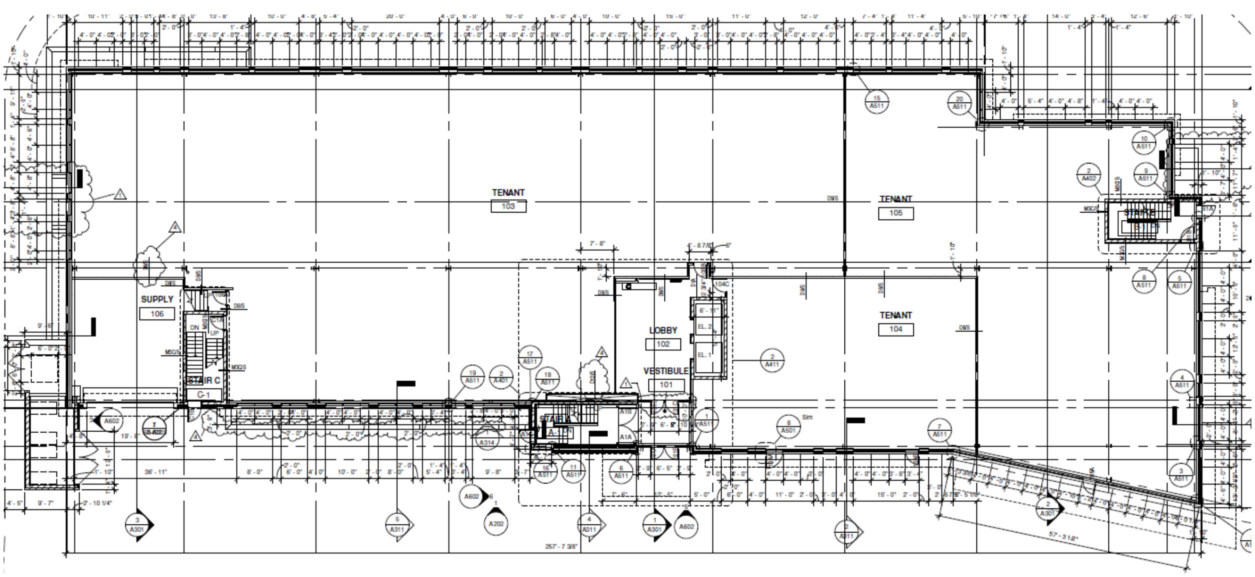
[1] The two utilities were Madison Gas and Electric which serves Madison, Wisconsin, and Xcel Energy which serves much of Minnesota and Northern Wisconsin.

density to that of an office[2]. Operating hours are typical of an office, with most tenants operating from 6:00 AM to 6:00 PM for five days per week, with some tenants – like the dentist – having more limited hours.

Most of the tenants do have server closets, small rooms where they keep servers, routers, switches, and other IT equipment. These closets are important to our study because they represent constant cooling loads, year-round. Each is served by its own fan coil unit on the VRF system; these fan coil units are always in cooling mode.

Figure 2. The Madison Office.

Figure 3. The Madison Office floor layout. There are three floors with layouts similar to this. This is from a core and shell design document, the tenant spaces have since been fitted out, primarily with closed and open office spaces, and one dentist office on the first floor.



[2] More intensive equipment loads that drive mechanical dental equipment are located in a separate unconditioned area.

The ground-source VRF system in the Madison Office is a three-pipe heat recovery system from Daikin, a manufacturer. The units are model RWEYQ _ _ _PTJU-HR. There is a total of 133 tons of installed VRF capacity across all condensing units; each of the seven tenants has a bundle of such units. Each of these bundles serves 1-15 fan coil units in each space (similarly to the layout in Figure 1; see an example of a fan coil in the Madison Office in Figure 4). Depending on the design of the space, there are also anywhere from 1 to 6 heat recovery units (called “branch selectors” by this manufacturer) in each space; the refrigerant first flows from the condensing units to each branch selector, which distribute heating or cooling refrigerant to each fan coil as appropriate (see an example of a branch selector from the Madison Office in Figure 5). The system is capable of direct heat recovery between zones; heat recovery occurs at the branch selector.

Figure 4. Typical fan coil. Note one supply and one return refrigerant line entering the unit in the middle.

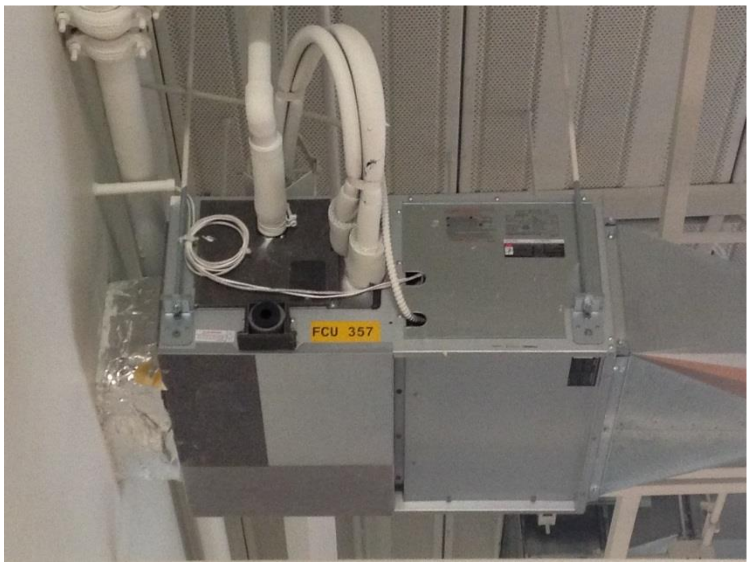
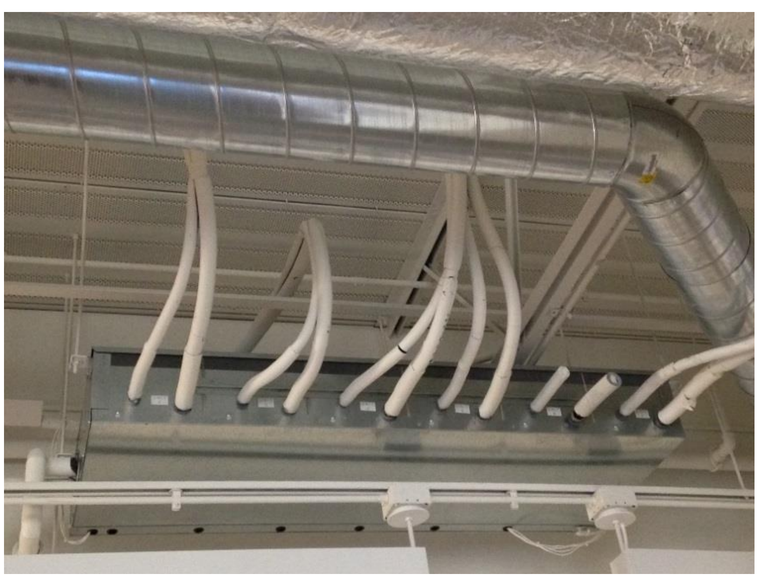


Figure 5. Typical branch selector. Note that this unit has five pairs of refrigerant lines (and one blank) and so serves five fan coil units.



Two typical ‘bundles’ of condensing units are shown schematically in Figure 6 (and in situ in Figure 7), with green lines depicting the three refrigerant lines connected to each tenant space (these lines first extend to a branch selector, then to the fan coils). Note that the refrigerant lines are manifolded together, so that the condensing units are capable of acting as one large condensing unit (e.g. if the space only needed cooling, all condensing units would be providing cooling). Tenants actually have anywhere from one to three units, two are shown in this particular example for simplicity. The blue lines depict the fluid from the GHX that serves as the heat source and sink for the units. All condensing units, as well as the building’s dedicated outside air system (this unit supplies the building’s ventilation), are piped in parallel in this fluid loop.

Figure 6. Schematic of VRF system in the Madison office.

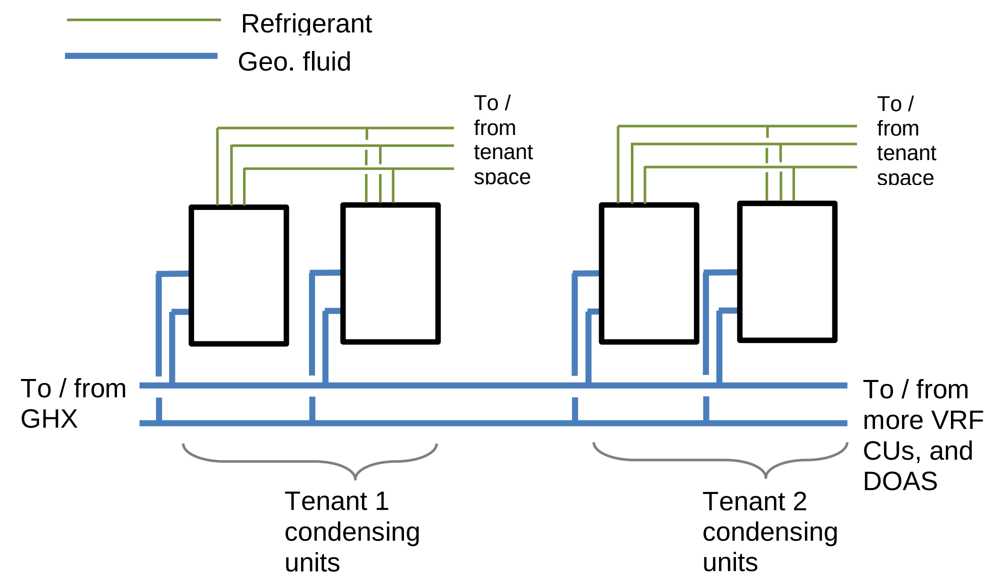


Figure 7. Schematic of VRF system in the Madison office.

The GHX referred to in Figure 6 is a closed-loop, vertical GHX. The GHX was sized for a block load of 119 tons, and 3 gpm per ton of flow. It contains 70 boreholes, each 250 feet in depth. The typical depth for boreholes in the area is 300-400 feet, but this particular site is constrained because it is in a water wellhead protection area. Between the constrained depth and the small footprint of the building site (no parking lot or significant green space) the wells were placed only 15 feet apart in order to attain a large enough GHX. One-inch HDPE u-tubes were used for the GHX.

One other significant system was attached to this GHX: the dedicated outside air system (DOAS). Every VRF system needs some supplemental method of introducing ventilation air. In this building it is a DOAS comprised of a ground-source heat pump rooftop unit, with energy recovery. This heat pump utilizes the same fluid loop for the heat source and sink as the VRF system, but its refrigerant is not in any way connected to the VRF units.

Minneapolis Office

The office that we studied in Minneapolis, MN is an owner-occupied office attached to an industrial building (see Figure 8). The office was served by both a VRF system and a traditional ground-source (GSHP) system – we studied just the portion of the office that was served by the VRF system. The building is occupied by Braun Intertec; the primary activities in the area we studied were engineering and construction administration. The area we studied was 5,450 ft^2, and included a ground floor and a lofted second floor (see Figure 9 for a general layout). The vast majority of the space served is in the core of office (there is very little perimeter space, with envelope, served by the VRF system). This had a significant impact on its average performance.

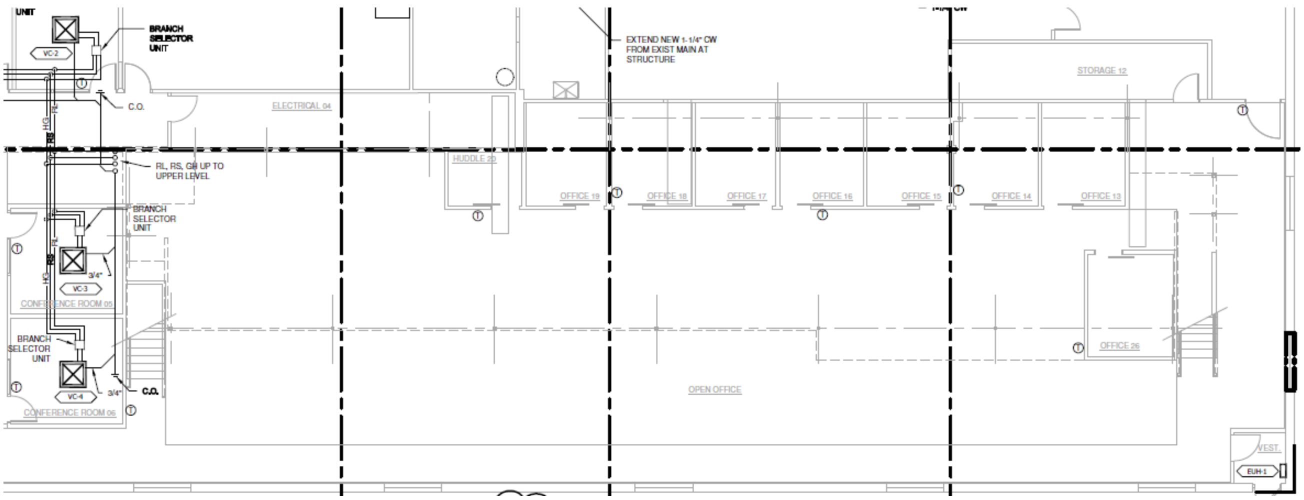
The building is tilt-up precast construction with rigid insulation in the precast. Windows comprise less than 10% of the exterior surface area, and are dual-pane, punched windows (exact properties are unknown).

Internal loads are consistent throughout the space. Private offices and cubicles of similar size all have typical office equipment. The space is occupied during typical office hours, roughly 6:00 AM to 6:00 PM during weekdays, plus limited Saturday usage. Some occupants do spend a significant amount of time in the field during working hours.

Lighting is all fluorescent, with a power density of approximately 0.8 W/ft^2. Unlike the Madison office, there are no server loads cooled by dedicated fan coils. But the fact that the spaces are primarily interior still drove loads to be heavily dominated by cooling.

Figure 8. The Minneapolis Office.

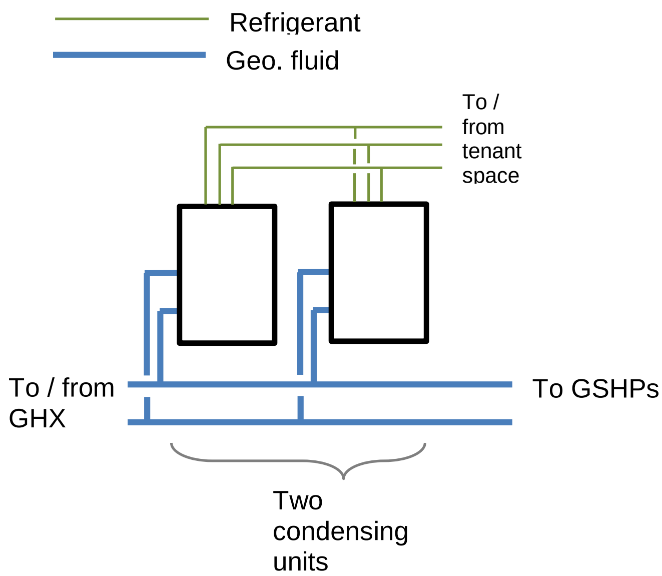
Figure 9. The Minneapolis Office floor layout. Note the majority is open office, with some private offices, and conference rooms off to the left. This is repeated at a smaller scale in a 2nd level mezzanine above this level.



The ground-source VRF system in the Minneapolis Office is a three-pipe heat recovery system from Daikin; the units are model RWEYQ _ _ _PTJU-HR (coincidentally the same units as the Madison office). There is a total of 10 tons of installed VRF capacity across two condensing units. These units serve 9 fan coil units in total. There are also 9 branch selectors, one for each fan coil. The refrigerant first flows from the condensing units to each branch selector, which distributes either heating or cooling refrigerant to its corresponding fan coil. This hardware is technically capable of direct heat recovery between zones, but since each fan coil is tied to its own branch selector there is no potential for direct heat recovery in this system as it’s designed.

The VRF condensing units are shown schematically in Figure 10, with green lines depicting the three refrigerant lines connected to each tenant space (these lines first extend to a branch selector, then to the fan coils). Note that the refrigerant lines are manifolded together, so that the two condensing units are capable of acting as one large condensing unit (e.g. if the space only needed cooling, both would be providing cooling). The blue lines depict the fluid from the GHX that serves as the heat source and sink for the units. All condensing units are piped in parallel in this fluid loop. In addition, a number of GSHPs are similarly piped in parallel – these serve the remaining office space.

Figure 10. Schematic of VRF system in the Minneapolis office.



The GHX referred to in Figure 10 is a closed-loop, vertical GHX. The specific design of the GHX is unknown, as the owner designed it as an ongoing research experiment and retains its design as proprietary. However, it yields temperatures in the fluid loop that are typical for a ground-source system (see Figure 14), so the performance information presented here for the VRF system can still be assumed to be representative even without a detailed understanding of the GHX.

MONITORING

With two buildings identified for study, we next established desired monitoring for the two buildings that would achieve the objectives of the project. Required data points included those in Table 1:

Table 1. Data points measured in the two buildings studied.

Data point	Instrument and Specification (if applicable)	Access for Madison Office	Access for Minneapolis Office
Building electricity usage	Utility house meter and submeter for non-house electricity	BAS	Not available
HVAC electricity usage, VRF equipment	Power meters on a sample of condensing units. These units represent the majority of the load in the space	BAS	BAS
DOAS electricity usage	Power meter	BAS	Not available
Pump electricity usage	Power meter	BAS	BAS
HVAC loads	Temperature sensors (supply and return) and insertion turbine flow meters on all condensing units where power is measured	BAS	BAS, with data loggers for supplemental temperatures

Ground loop supply and return temperatures, and fluid flow	Temperature sensors, insertion turbine flow meter	BAS	BAS
OA flow rate	Airflow monitoring station	BAS	Not available
Space temperatures	Space temperature sensors	BAS	BAS
Equipment mode (heating/cooling/etc.): condensing units, DOAS heat pump, fan coils	Mode output from equipment	BAS	BAS
Outside air temperature and humidity	Weather station	Purchased data	Purchased data
IT tenant loads	Current transducers; spot measured voltage and power factor (to calculate total power)	Data loggers on each major tenant IT circuit (servers, etc.)	Not available
Other tenant loads: lighting, plug loads, process loads	Utility meters	Access to utility data, monthly, via relationships with each tenant	Not available
Space operating schedule	Interview with tenant	Interview	Interview

Spot measurements and data logged measurements were conducted with factory-calibrated sensors. All BAS data points were validated before collection using spot measurement (sampling was used to validate large numbers of similar data points).

ANALYSIS

Our primary method of performance evaluation is calculation of the coefficient of performance (COP) of the VRF units. We calculated COP for individual VRF units as well as groups of units. In addition, we also calculate a system COP that incorporates the energy requirements from all system components, which in this case means simply adding the pumping power consumption to the COP calculation.

COP for a VRF unit (or units) is calculated from load and power consumption, as shown in equation 1:

$$COP = \frac{\dot{q}_{load}}{\dot{W}_{fans}+\dot{W}_{compressors}} \tag{1}$$

where $\dot{q}_{load}$ represents the cooling or heating load being met by the VRF units in the area of study, and $\dot{W}_{fans}$ and $\dot{W}_{compressors}$ represent power consumption of all the fans and compressors, respectively, in the area of study. $\dot{W}_{fans}$ and $\dot{W}_{compressors}$ are measured directly.

The calculation of $\dot{q}_{load}$ based on measured quantities is facilitated by the fact that the VRF units are water-source. Measurements of water flow and temperature are substantially more accurate than similar measurements of air. In the case of these two projects and many others in cold climates, the “water” cooling the VRF unit is actually a propylene-glycol and water solution, which we’ll simply call “the fluid”. $\dot{q}_{load}$ can be calculated based on a few measurements of this fluid, as well as the power consumption of the equipment, according to equations 2 and 3 depending on the mode of the unit:

$$\dot{q}_{load,cooling} = \dot{V}_{fluid}c_{p,fluid}\rho_{fluid}\left(T_{fluid,out} - T_{fluid,in}\right) - \dot{W}_{fans} - \dot{W}_{compressors} \tag{2}$$

$$\dot{q}_{load,heating} = \dot{V}_{fluid} c_{p,fluid} \rho_{fluid} \left(T_{fluid,in} - T_{fluid,out}\right) + \dot{W}_{fans} + \dot{W}_{compressors} \qquad (3)$$

where $\dot{V}_{fluid}$ is the fluid flow rate, c_p is the specific heat of the fluid, ρ_{fluid} is the density of the fluid, and the $T_{fluid,in}$ and $T_{fluid,out}$ terms are the temperature of the fluid flowing into and out of the VRF unit.

This is a standard method for calculating loads on hydronic equipment. However, it only works if the equipment is in either entirely in heating or entirely in cooling mode. VRF introduces an additional wrinkle, namely that VRF units *can* be providing both heating and cooling at the same time. We get around this by monitoring the heating and cooling modes of each zone, and only calculating COP when either every zone is in heating (true for a certain number of nights and weekends in the winter) or every zone is in cooling (true for several months of time from spring to fall).

In the Madison Office an additional challenge in determining COP is introduced by the fact that the VRF systems also serve a number of server rooms, in which there is a significant cooling load every hour of the year. This is corrected for by measuring the load that these servers impose on their respective rooms, which are essentially equal to electrical power consumed by the servers. With this power measured (via data loggers) then COP of VRF heating becomes:

$$COP_H = \frac{\dot{W}_{fans} + \dot{W}_{compressors} + \dot{W}_{servers}\left(1 + \frac{COP_H}{COP_C}\right) - \dot{V}_{fluid} c_{p,fluid} \rho_{fluid}\left(T_{fluid,out} - T_{fluid,in}\right)}{\dot{W}_{fans} + \dot{W}_{compressors}} \qquad (4)$$

where the COP_H/COP_C term is approximated iteratively as COP_H is solved. Still equation 4 is necessarily an approximation as we don't have a direct measurement of COP_C.

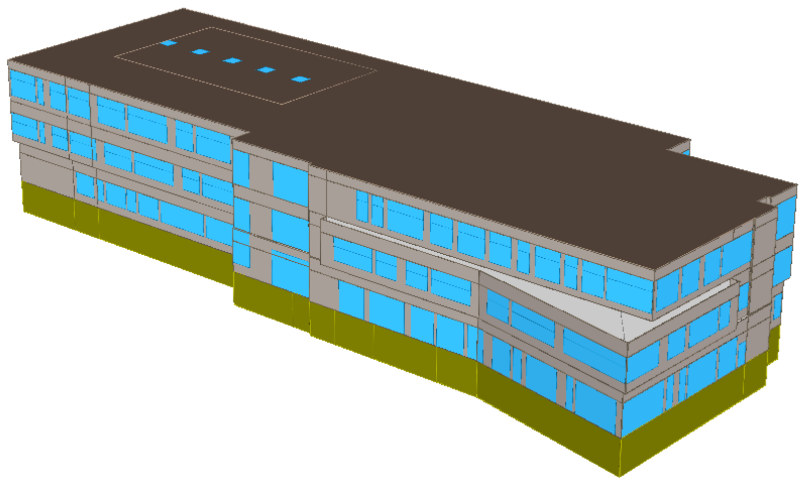
Note that the load and energy usage of the DOAS is separate, and not included in any of these COP calculations.

In addition to COP, it is also beneficial to consider system COP. To calculate system COP for a ground-source VRF system, we simply need to add pump power, $\dot{W}_{pumps}$, to the total power (wherever $\dot{W}_{fans} + \dot{W}_{compressors}$ is shown).

ENERGY MODELING

The calculation of COP values allows for understanding of system performance in relation to manufacturer's published ratings, and comparison to other packaged HVAC equipment. But COP is limited in its ability to be compared with other HVAC system types like multizone VAV, which happens to be the most common system type for most offices of this size (and is recognized as such by ASHRAE 90.1 Appendix G). A well-calibrated whole-building energy model is the best tool for this type of comparison. In this case we used eQUEST 3.64b to build the energy models (see Figure 11).

Figure 11. Whole building energy model for the Madison office.



The initial model inputs were based on full construction documents for the buildings and associated mechanical systems, including building information described in *The Buildings and Systems* section above. The models were adjusted based on additional as-built information, and subsequently calibrated based on measured data. The following prioritization was used for matching data during calibration:

Increasing Priority ↑

1. Measured VRF performance data (energy, loads, temperatures, pumping)
2. Submeters and other equipment-specific data collected (see Table 1)
3. Building monthly utility usage data
4. Observation of actual building operation and use
5. Design and construction documentation

The VRF system model is worth additional description. The system is modeled in two parts, an airside and a waterside. On the waterside, the model consists of a condenser water loop with a variable speed pump. The loop is served by a GHX modeled using eQUEST's g-function model, with properties described in *The Buildings and Systems* section. The Configuration input in eQUEST was then adjusted until the model was calibrated to the measurements of fluid temperature leaving the GHX.

The airside system is modeled as water-to-air heat pumps (type PVVT), with one heat pump for each zone. The fan power is zeroed out, and instead included in the overall energy efficiency of each unit (per AHRI rating methodology). This efficiency is primarily governed by three elements in cooling mode and three elements in heating mode. The three elements for cooling mode are:

a) The rated efficiency ("EIR" in eQUEST), specified based on results from the *VRF PerfORMance Summary* section below
b) The curve describing the cooling efficiency as a function of part load ("COOL-EIR-FPLR" in eQUEST)
c) The curve describing the cooling efficiency as a function of entering fluid temperature ("COOL-EIR-FT")

The same elements are also required for heating mode. All of these performance elements were calibrated based on measured VRF performance data, at a sub-hourly resolution. Per the manufacturer's specifications, the systems were able to unload to 2% of full load. No supplemental heat or defrost is modeled due to the presence of the ground-source loop. Direct heat recovery via the heat recovery units (branch selectors) cannot be modeled directly within eQUEST, so it is post-processed with a zone-by-zone spreadsheet calculation. More details regarding how this spreadsheet was calibrated to measured recovery potential are discussed in the *Heat Recovery Performance* section.

Outside air is handled, as it is in the building, via a model of a completely separate DOAS.

The primary limitation of this modeling approach is that there are multiple condensing units in the model (one for each zone) representing fewer, shared condensing units in the actual building. Elements (a) and (c) above are still handled appropriately in this case, but the accuracy of element (b) is decreased because the part load ratio of each condensing unit in the model will be driven by the local zone part load, and not the overall part load of the tenant space as they are in the actual building. To the extent that an individual zone's load profile differs from that of the larger tenant space, inaccuracy will result. Fortunately, most of the zones in the building (such as most open office spaces) exhibit load profiles that are similar to the larger tenant space.

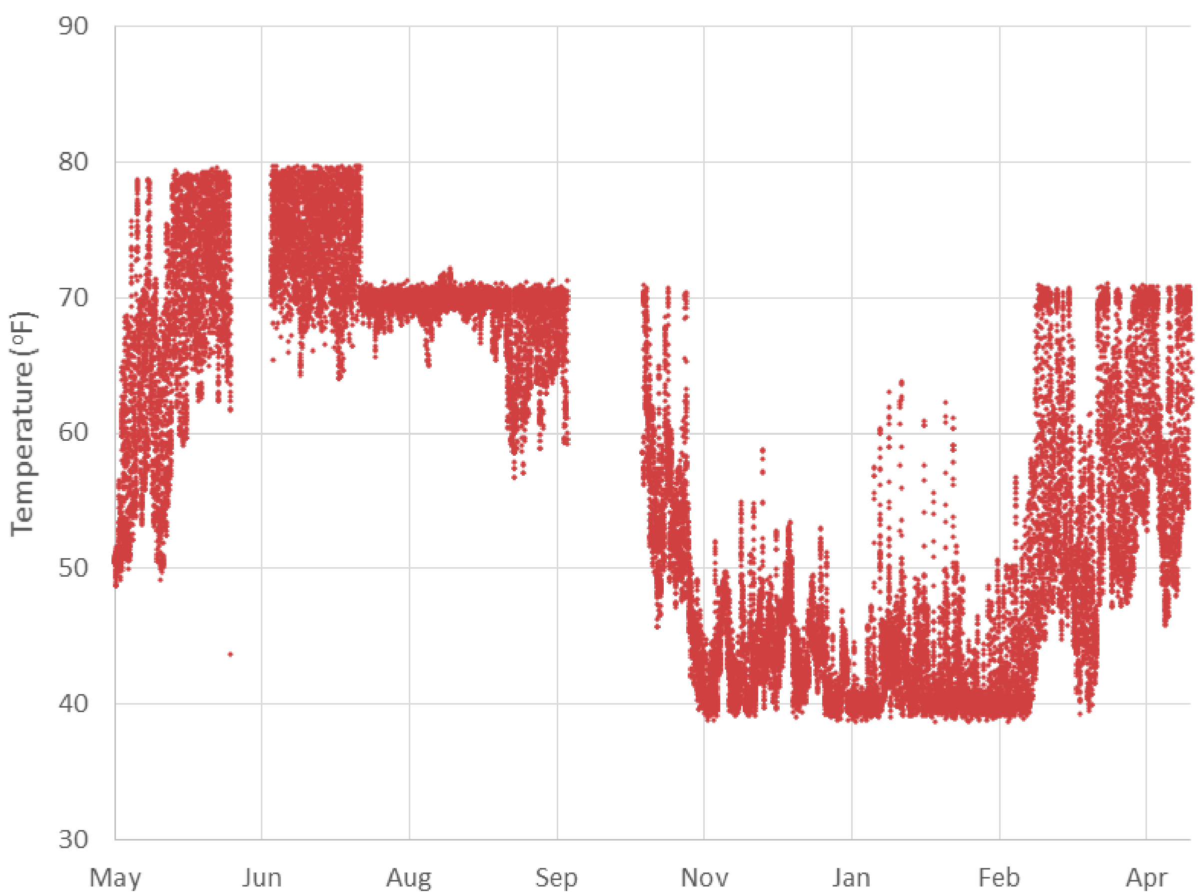
RESULTS

We monitored performance of VRF systems at two buildings; results of that monitoring are presented below.

MADISON OFFICE RESULTS

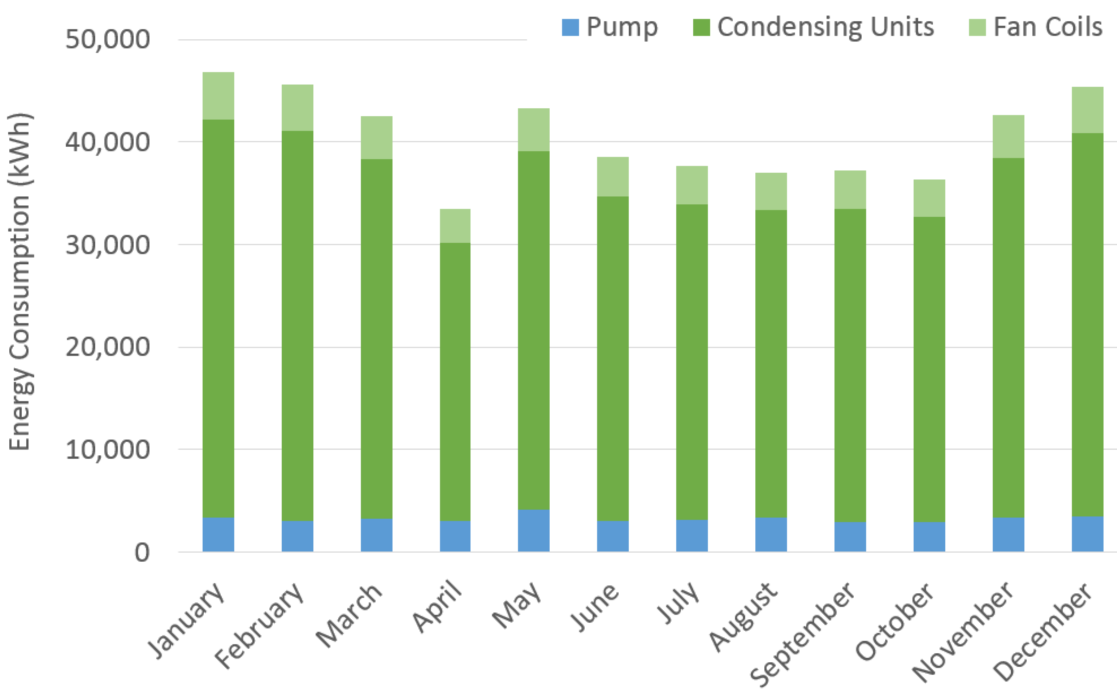
The performance of the ground-source VRF system in the Madison Office starts with the performance of the GHX. This GHX has performed well. Supply temperatures to the building are shown in Figure 12; the GHX is able to maintain fluid temperature to the building between 40ºF and 70ºF throughout all seasons (note that prior to July 1, 2014 the maximum temperature setpoint for the loop was 80ºF, before being modified to 70ºF).

Figure 12. GHX outlet temperature in the Madison Office.



The energy consumption of the Madison ground-source VRF system is shown in Figure 13, organized by component and by month. Note that pump power consumption is largely flat. Consumption of the remaining VRF equipment (fan coils and condensing units) is also relatively flat compared to most HVAC systems, suggesting that the unit's power does not ramp down substantially during milder weather (i.e. the spring and fall seasons).

Figure 13. Energy consumption of VRF components by month.

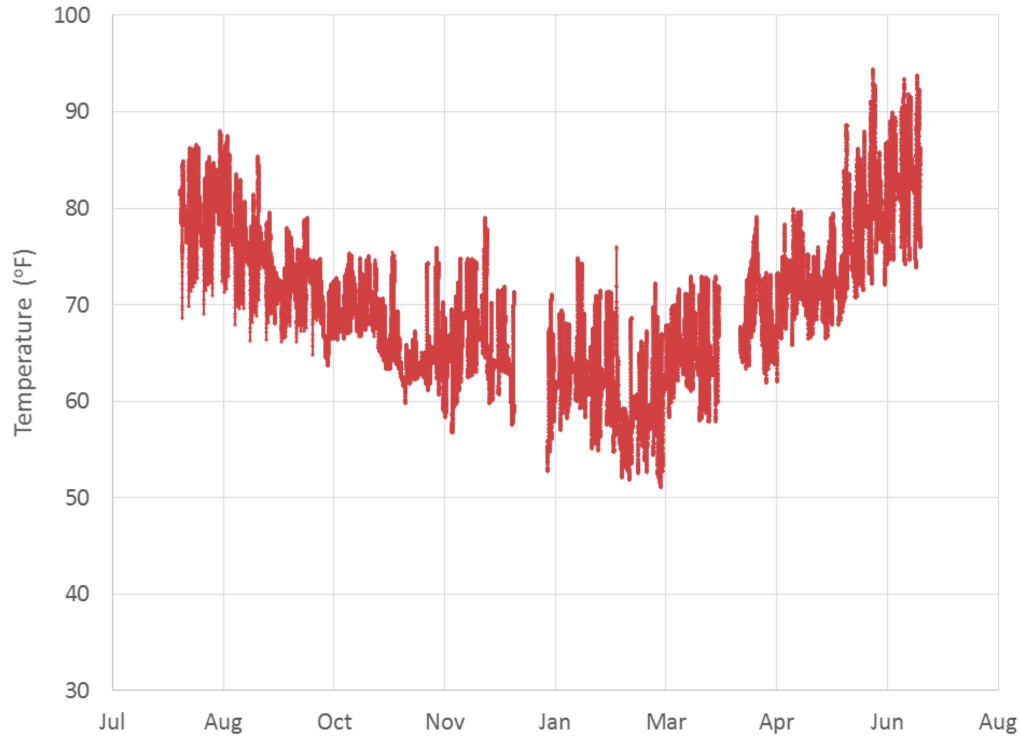


For this system, 82% of the energy consumption is by the condensing units, and 10% is from the fan coils. The remaining 8% of the system energy is consumed by the pump.

MINNEAPOLIS OFFICE RESULTS

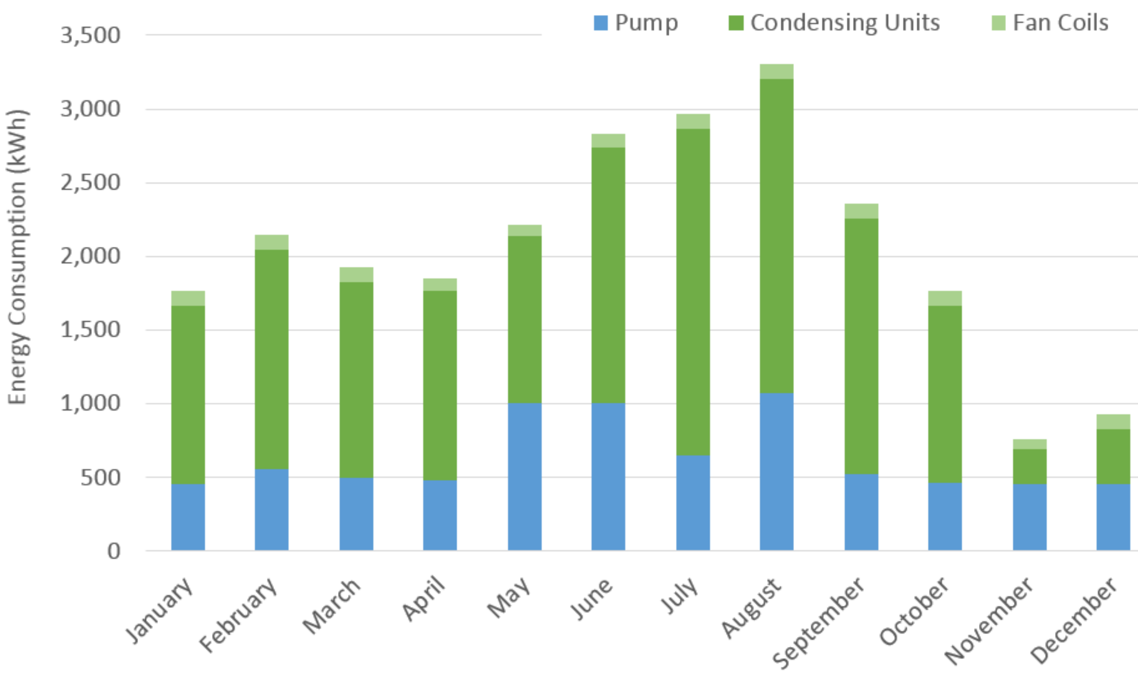
The Minneapolis Office has a GHX that serves both a VRF system and a standard GSHP system. The GHX is sized appropriately for both systems, so the fluid temperatures entering the VRF units are not markedly different from a standalone ground-source VRF system (see this fluid temperature in Figure 14). The temperature ranges from about 50ºF to just below 90ºF. Fifty degrees is relatively high for a minimum geothermal fluid temperature in this climate, suggesting that this GHX is cooling dominated. A cooling dominated GHX absorbs more energy from the fluid (for cooling) each year than it rejects to the fluid (for heating). This in turn suggests that the buildings cooling loads are greater than its heating loads.

Figure 14. GHX outlet temperature in the Minneapolis Office.



The energy consumption of the ground-source VRF system in Minneapolis is shown in Figure 15, organized by component and by month. Note that energy consumption is significantly more variable than for the system in Madison. Consumption is also substantially higher in summer months than winter, again reinforcing that this system is cooling dominated.

Figure 15. Energy consumption of VRF components by month. Pump data is extrapolated for November and December.



The Minneapolis system has a similar ratio of condensing unit energy to fan coil energy as Madison. But the pump component is significantly greater than for the Madison building: pumping represents 31% of the Minneapolis system's energy consumption. Note that because this system was on a common GHX (and therefore used common pumps) with a GSHP system, we did have to use an algorithm based on flow to proportion total pump energy between VRF and geothermal heat pumps. Our algorithm was based on the flow capacity through each unit (VRF and heat pump), as well as how often each unit was on. Even with some uncertainty in pump apportioning, the pump energy is high for this system.

Also note that the condensing unit energy consumption (in Figure 15) for November and December appears markedly different from the other months. We did not find any obvious problems with the data in this time period. However, as we were not able to independently verify power measurement during this time, condensing unit energy consumption for these two months should be assumed to have a much higher degree of uncertainty. That data is neglected in the analysis in the next section.

VRF PERFORMANCE SUMMARY

The primary objective of this study is to determine the energy performance (i.e. efficiency) of these ground-source VRF systems. In this section we analyze that performance from multiple angles.

Performance vs. Weather

To begin to consider the actual performance of the VRF systems in situ, it is first useful to look at the VRF energy consumption seasonally. In Figure 16, we plot power consumption every 15 minutes (over a full year) versus outdoor air temperature for the Madison building[3]. This does not include pumping energy.

[3] In the actual Madison Office, we monitored three separate tenant VRF systems, out of a total of seven tenants. Our monitoring results were scaled up to the scale of the entire building based on an additional load measurement across the GHX (with an

Figure 16. VRF Power vs. outside air temperature, for the Madison office.

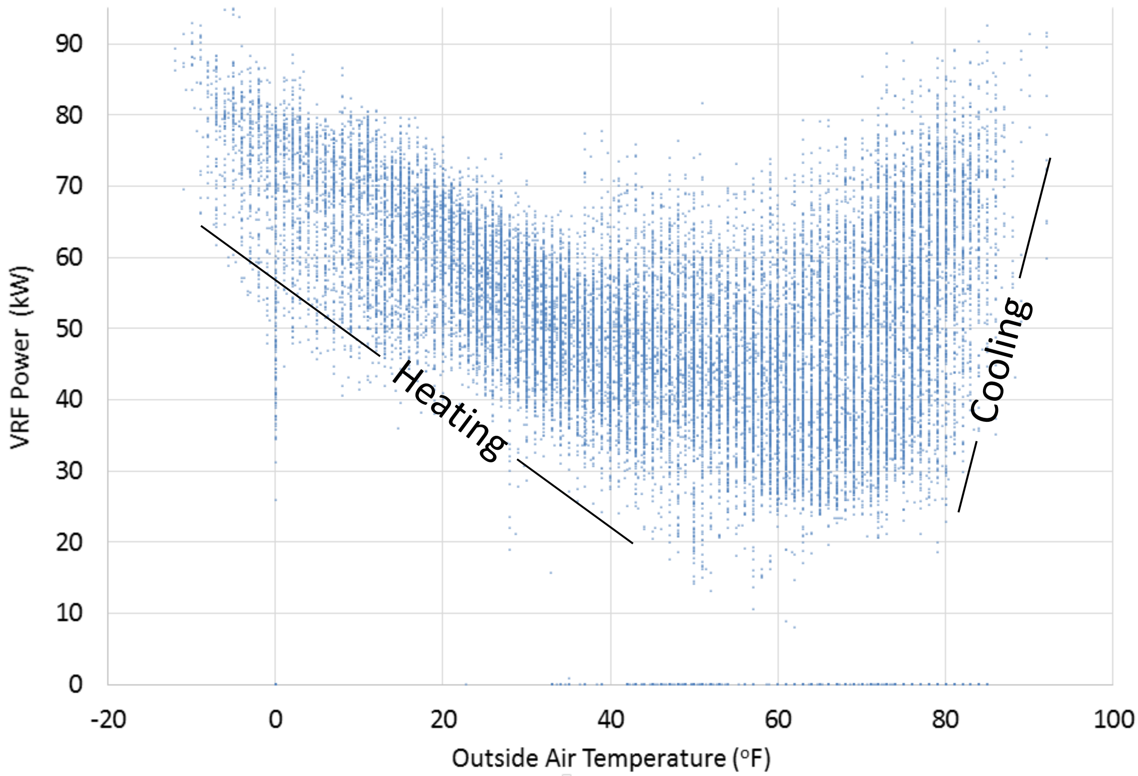
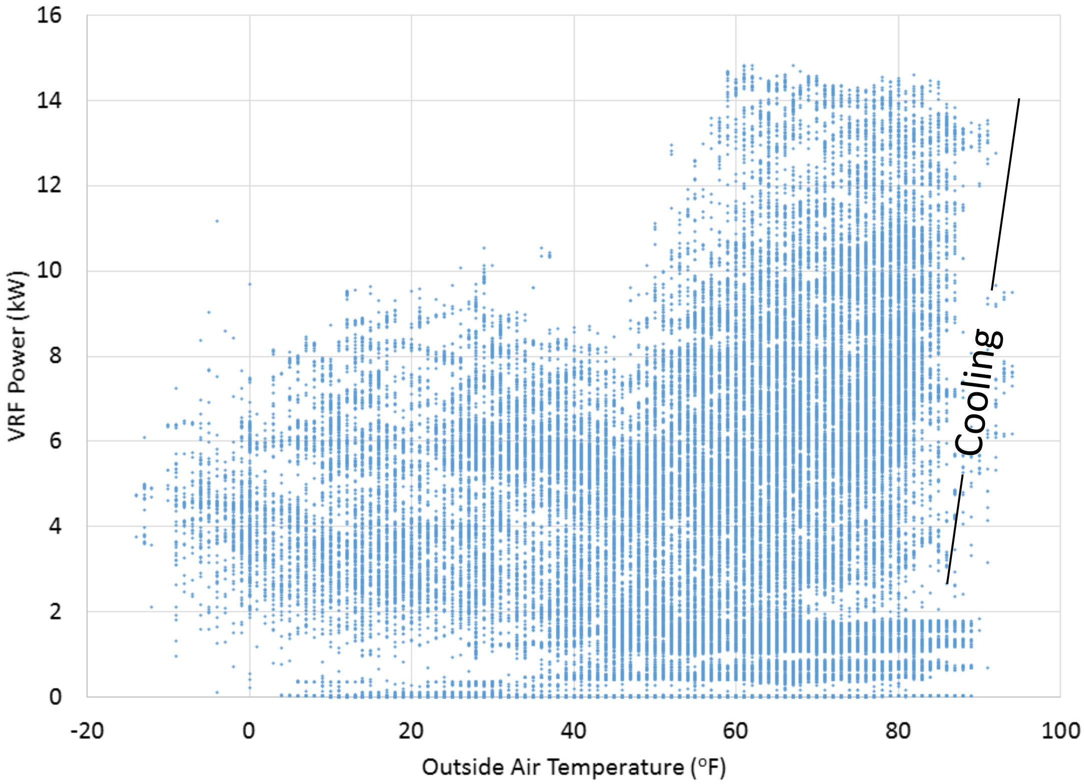


Figure 16 represents a relatively standard method of analyzing HVAC energy usage. We can deduce a few things from this plot. First, this building has both heating loads (below about 40ºF) and cooling loads (above about 70ºF) that are driven significantly by outdoor temperature. It's therefore likely that the building's envelope contributes a significant portion of the load on the VRF system, which is reasonable for a building with a separate ventilation system, a narrow floorplate, highly efficient lighting, and limited process loads. This is corroborated by more direct load measurements. Secondly, the balance point of the building appears to generally be between 55ºF and 65ºF. Thirdly, in this relatively balanced range, system power consumption remains relatively high – generally about 35 kW, or 39% of the peak consumption. This suggests poorer part load performance (i.e. a lower amount of turn-down) then suggested by the manufacturer. And finally, the system almost never shuts off – there are very few power points at 0 kW. So even when the load in the building is essentially zero, the units do not shut down but rather remain on at the their point of lowest modulation.

Similarly, the power consumption of the VRF system in Minneapolis is plotted versus outdoor air temperature in Figure 17, every 5 minutes for a year. This plot also does not include pumping power.

extrapolation based on square-footage serving as a check). So the power consumption data in this section, as well as the COPs reported later in the section, are based on a sample of the VRF systems in the building. Wherever total building data is shown (as in Figure 16), it is scaled.

Figure 17. VRF Power vs. outside air temperature, for the Minneapolis office.



For the Minneapolis office, Figure 17 suggests that there is a significant impact from the building envelope on cooling, but there is very little heating required by the condensing units. The more direct measurement of the load on the system (using ΔT across the units) shows that the units are only providing significant heating for a few days of the year, and then only for limited hours each day. This reflects the fact that the spaces served by the system have: significant internal loads, fairly little exposure to the perimeter, and some solar gain from the perimeter.

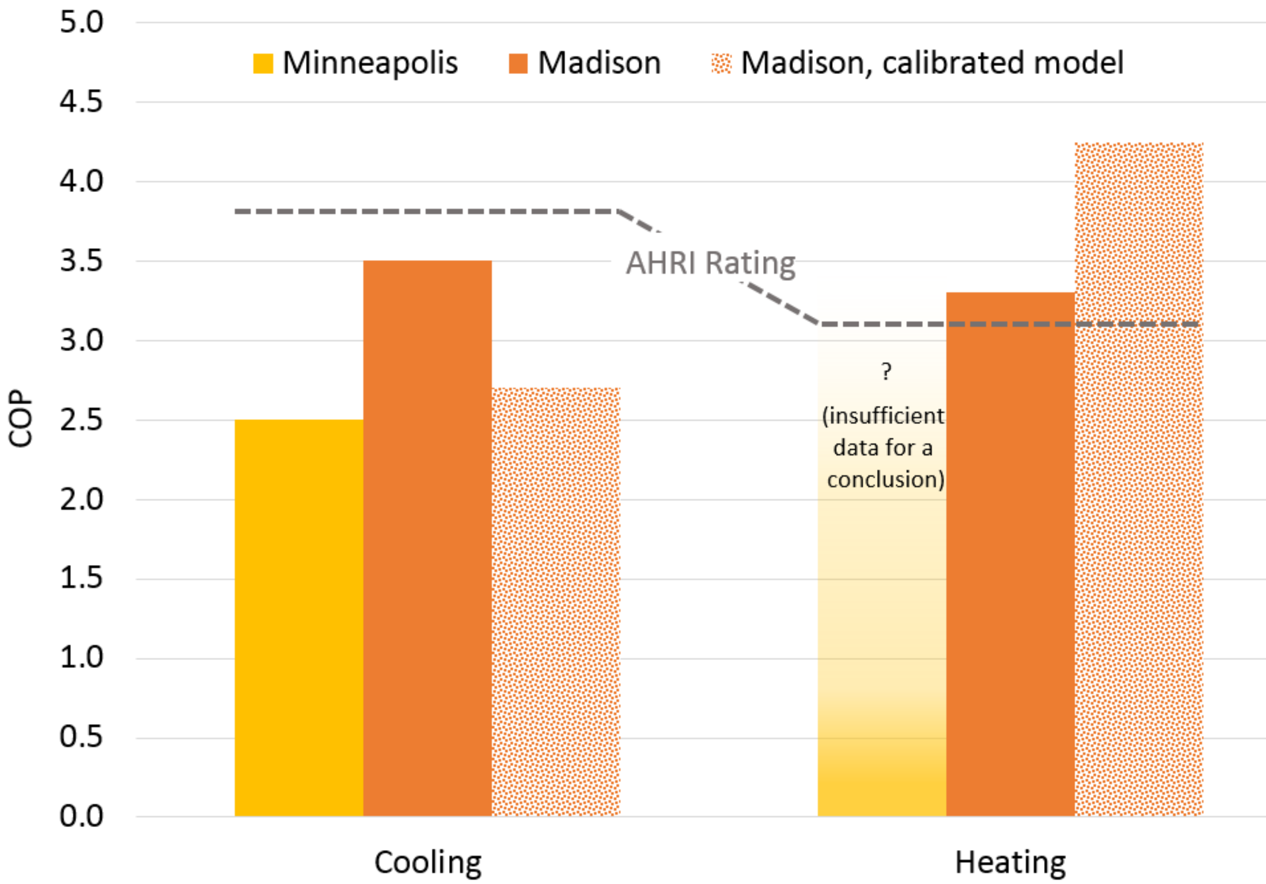
But more importantly, note that in contrast to the Madison system, in and around the balance point this system operates at about 10-20% of its peak load. The system also shuts off relatively often – it is off throughout the night on most nights; also a contrast to the Madison system.

Performance at Rated Conditions

Coefficient of performance (COP) is a more thorough metric for equipment performance than power consumption. COP is the ratio of load served to power consumed, and therefore demonstrates performance relative to the size of the load being met. Manufacturers of VRF equipment publish an expected COP based on a set of typical operating conditions set by ANSI/AHRI Standard 1230. In this Standard the fan coils and condensing units are both included in COP. It is useful to determine what the performance of the systems we studied are at these conditions.

In the 2010 version of the Standard, those conditions are, for cooling, 80.6°F return dry bulb temperature, 66.2°F return wet bulb temperature, and 77°F entering fluid temperature. For heating, the rating conditions are: 68.0°F return dry bulb temperature, 59.0°F return wet bulb temperature, and 32°F entering fluid temperature. For both heating and cooling, the rated condition assumes the units are loaded to 100% capacity. As subsequent plots will show, there were limited data points collected at exactly those conditions. But by selecting those points that were close (and using reasonably small extrapolation for part load), we estimated the COP of each system under rated conditions. Those COPs are shown in Figure 18. For comparison the AHRI rating for this equipment is also shown as a dotted gray line.

Figure 18. COP of each system in actual operation, at conditions close to AHRI rating standards.



Both the Madison and Minneapolis office systems are performing significantly below the rated cooling performance; the <u>rated</u> performance of both systems is 3.8. The Minneapolis system is performing near 2.5, and Madison's system is just barely reaching 3.5. Additionally, when measurements are ignored and the COP of the Madison system is determined by calibrating the whole-building energy model, it suggests the COP may be even a bit lower.

The results for the heating COP were more favorable. Both systems have a <u>rated</u> COP in heating of 3.1. The Madison system performs at about 3.3. There is not enough data for the Minneapolis system in heating mode to state its operating COP with confidence.

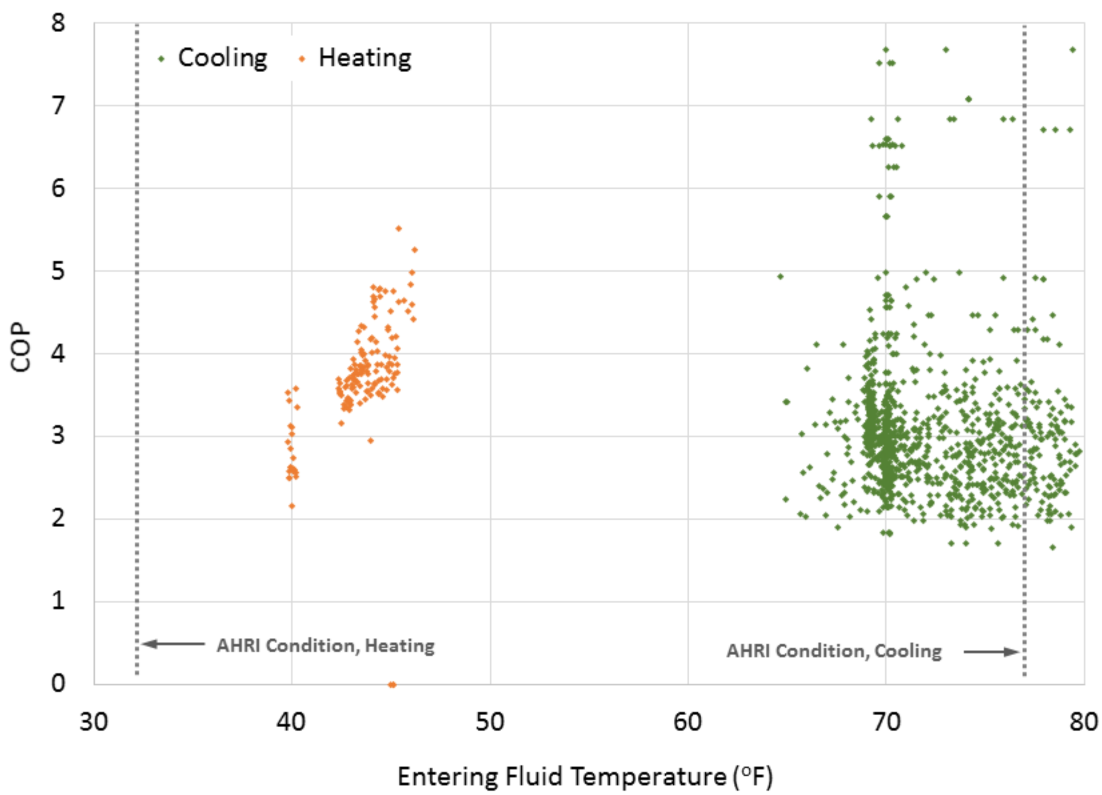
In our review of the literature we found just one other study with measured performance of a ground-source VRF system: Oakland University in Michigan (Piljae, 2014). In that system, the VRF units performed similar to the Madison system in heating (Oakland's system equaled a COP of 3.3). But the cooling performance of the Oakland University system was substantially better, reaching a COP of about 4.9. And this is with the same brand and a similar vintage of equipment. We don't have data to explain the difference in performance.

Performance at Other Conditions

The primary energy saving promise of ground-source VRF systems is in their performance across varying part-loads and across varying seasons. This results from their variable speed operation and the mild fluid temperatures produced by the GHX. So we also looked at the performance (COP) of the VRF systems at different fluid temperatures and part loads.

The COP for the Madison system is shown as a function of entering fluid temperature in Figure 19, for both heating and cooling modes. The fluid temperature at AHRI rated conditions is shown with the dotted gray line. Note that in this plot, and throughout this section, there are substantially fewer heating data points than cooling. This is because a VRF system can, at any time, provide both heating and cooling, and there are very few times when a system is *only* providing heating. So we have very few data points for which we can calculate heating COP in the Madison Office.

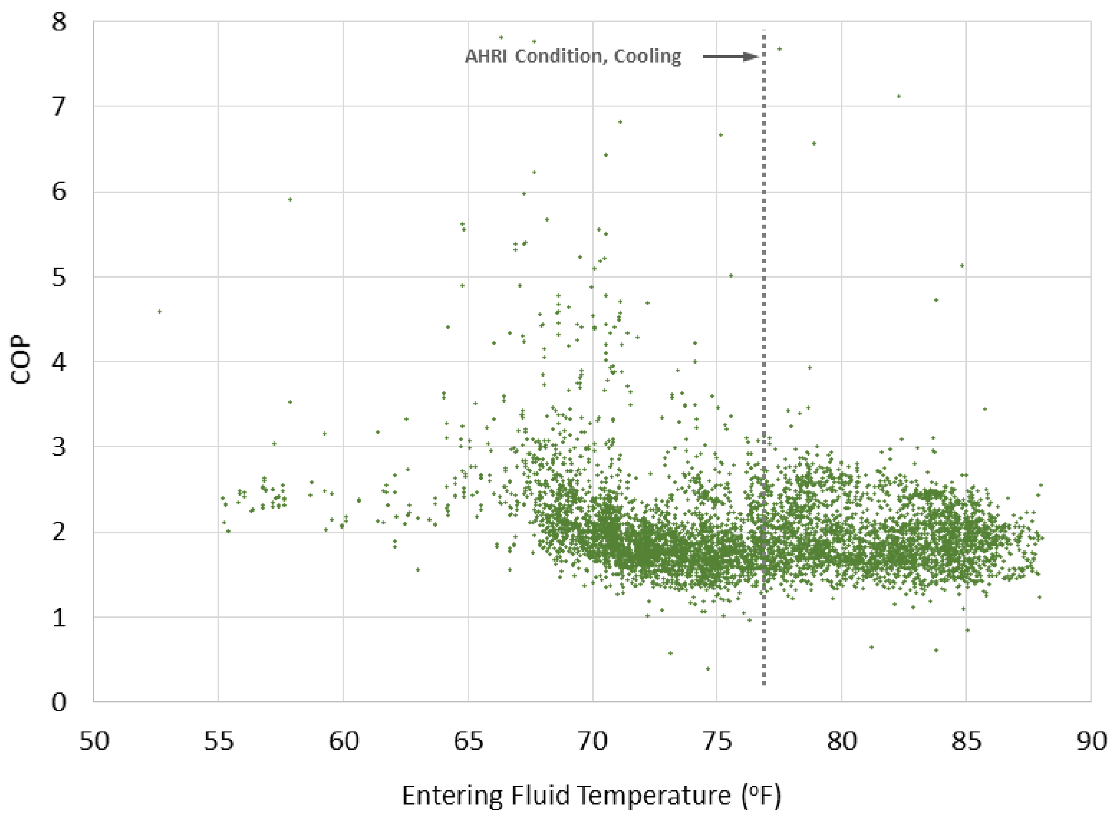
Figure 19. COP of the VRF system at the Madison office as a function of fluid temperature.



In heating mode, the COP increases noticeably as the fluid temperature increases from the mid-30s (near rated conditions); this is as expected from manufacturer's documented performance. The cooling performance is more difficult to analyze, because the GHX controls attempt to maintain the fluid temperature entering the building at 70°F, so the vast majority of the data points simply fall on this line. There are cooling measurements above this line, but they do not show any significant correlation with fluid temperature. This is somewhat unexpected.

The COP for the Minneapolis system is shown as a function of entering fluid temperature in Figure 20, for cooling mode. The fluid temperature at AHRI rated condition is shown with the dotted gray line.

Figure 20. COP of the VRF system at the Minneapolis office as a function of fluid temperature.



Similarly to the Madison system, there is little correlation between COP and fluid temperature above the rated condition. This could be because in this region the part load of the system has a bigger influence on COP. Below about 74ºF, there does seem to be an increase in COP as the fluid temperature decreases, which would be expected in cooling mode. As for heating mode, as described below Figure 17 the Minneapolis Office system serves a set of zones that happen to be in pure heating mode very rarely, so there is not enough data to determine heating COP.

It is also important to consider the performance of the systems as a function of their loading. Due to the variable operation of the condensing units, VRF systems are supposed to operate significantly more efficiently (i.e. have a much higher COP) at low load (part load). In Figure 21 this is demonstrated by the heating COP of the Madison office as a function of load: as the load decreases from its peak, the COP increases.

Figure 21. COP, in heating, of the VRF system at the Madison office as a function of load.

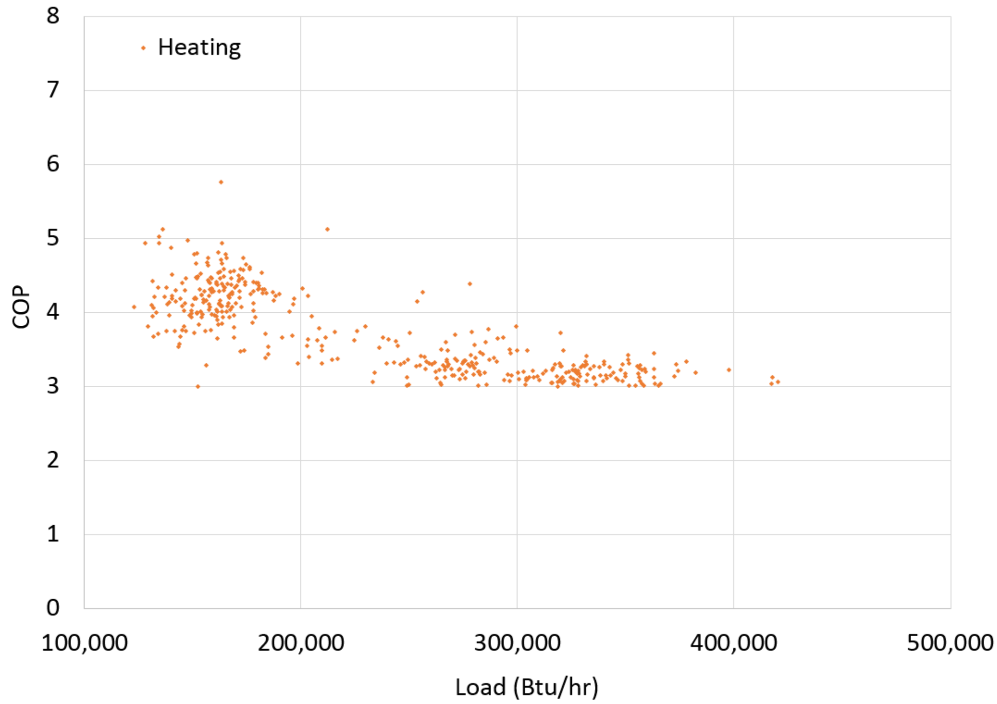
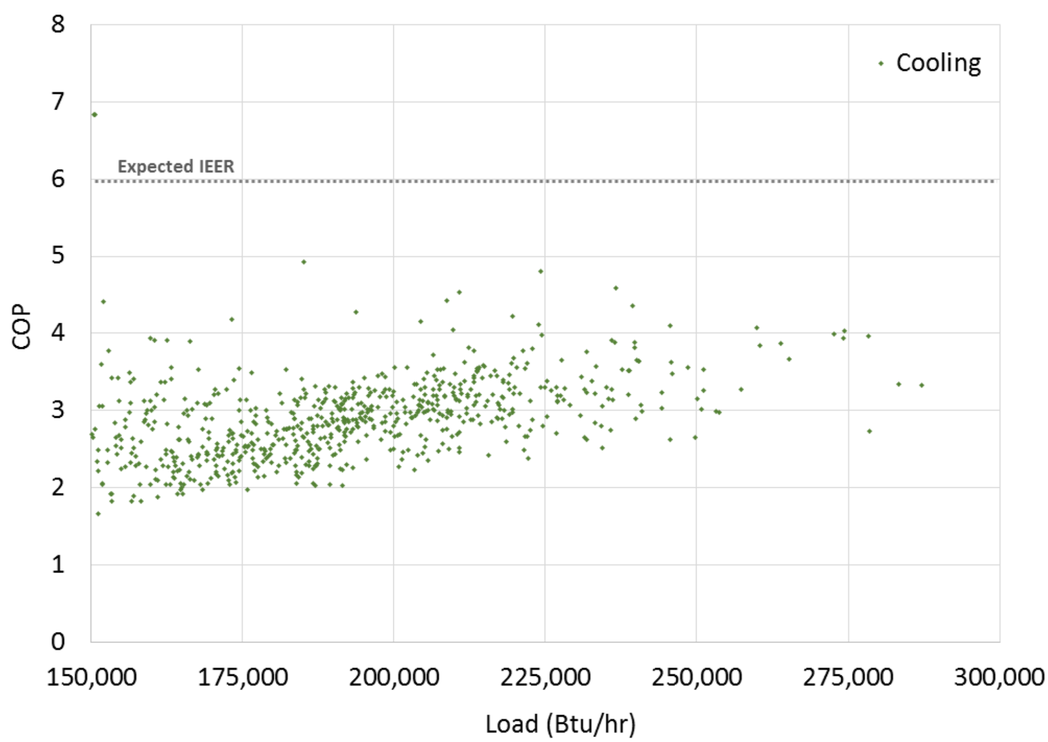


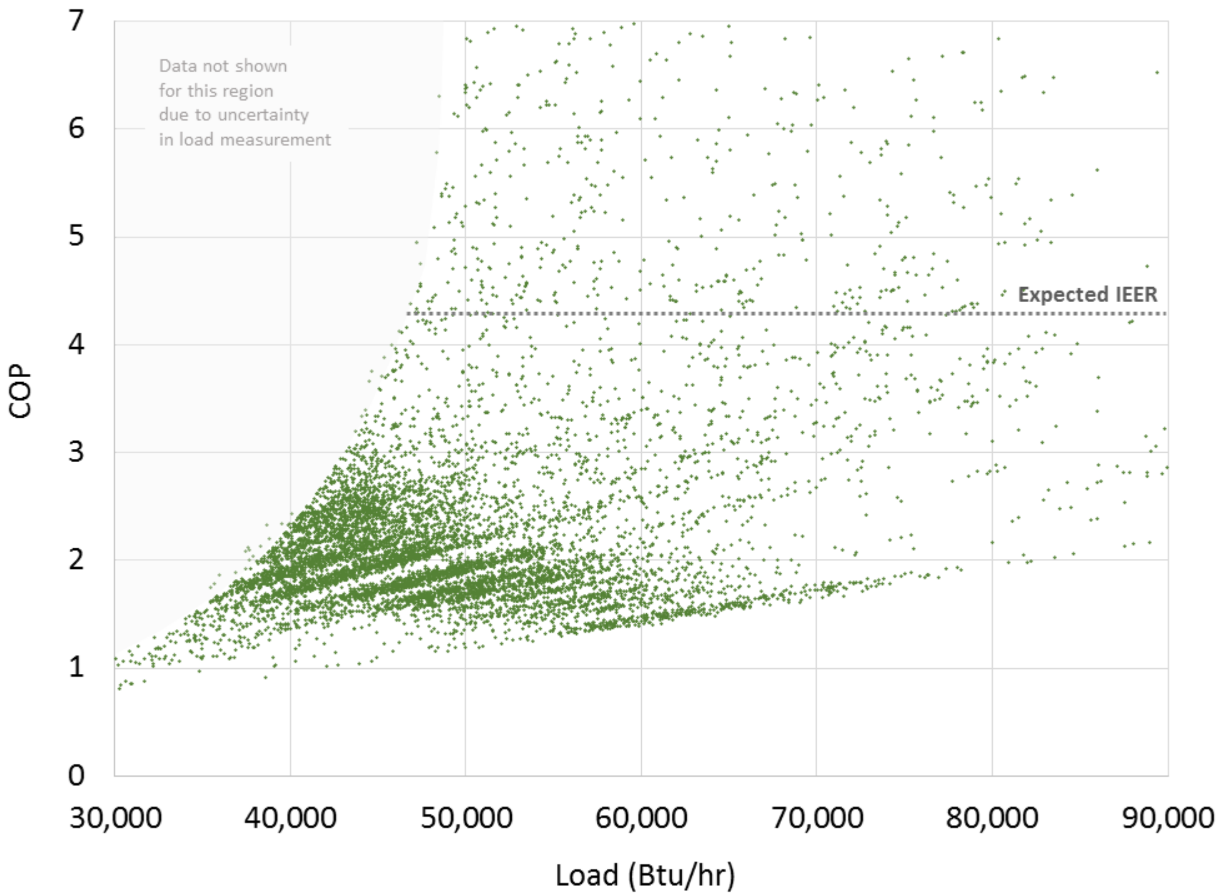
Figure 22 similarly shows the cooling COP versus the load on the system. In cooling, the performance of the system has the opposite trend: it decreases as the load decreases.

Figure 22. COP, in cooling, of the VRF system at the Madison office as a function of load.



Finally, Figure 23 shows the COP as a function of load for cooling mode in the Minneapolis VRF system. This data is less conclusive, showing significant lower COP at part load, but also some data points above the expected IEER.

Figure 23. COP, in cooling, of the VRF system at the Minneapolis office as a function of load.



The rating standard ANSI/AHRI Standard 1230 also includes an indication of part load performance, called IEER. IEER is only given for cooling mode. It describes the performance of the equipment across the spectrum of part loads expected in a typical season. IEER averages this part load performance into one single number. Based on published data, the VRF equipment that we've studied should have an average COP (across all cooling loads) that is 71% higher than the cooling COP at the rated peak condition.

With the peak COP that we measured in Madison at 3.5, the average performance for the year should approach a COP that is 71% higher than 3.5 – specifically, 6.0. Regardless of our ability to describe specific correlations for data in Figure 19 or Figure 22, it is clear that the vast majority of operating performances are well below 6.0. In fact, average cooling COP during just the summer months suggests a value close to 2.3, which is below that of the rated condition.

Similarly, with a rated cooling COP of 2.5 in the Minneapolis office, cooling performance data overall should average near 4.3. In actuality, it averages about 3.4 (this disparity is also clear in Figure 23).

System COP

An alternative performance metric that some are using to compare HVAC system choices is "System COP". This metric is meant to compare a larger built-up system, such as a water-source heat pump, to a fully packaged system like a rooftop unit. Rooftop units are generally given a single efficiency metric, like COP or EER, that captures the entire performance of the system (fan, compressor, heat rejection, auxiliary). In a GSHP or ground-source VRF system, the system COP is simply the COP of the VRF units (which includes both the compressors and fans), with the addition of power consumption for the pumps that circulate fluid through the GHX.

By this metric, the system COPs for Madison were 3.2 in cooling and 3.4 in heating. The system COP at the Minneapolis office was considerably lower, at 1.4, due to the pump being larger in proportion to what was a small VRF system. In other work we've done studying system COPs using a combination of field measurement and modeling, we've seen that standard VAV systems often have system COPs just over 2.0. A central plant geothermal system (with a heat recovery chiller) may perform with a COP of about 2.2-2.6. And traditional unitary GSHPs can reach system COPs of 3.5 depending on design. Code-minimum performance for a simple rooftop unit, for comparison, is about 3.0. But this may be a misleading comparison. The literature suggests that rooftop unit COP not only degrades much quicker due to equipment faults and failures, but that a rooftop unit approach may have other drawbacks and constraints that can't be included in a COP (rooftop area requirements, maintenance, comfort and controllability, flexibility, etc.). Rooftop unit COP ratings also do not often include a reasonable fan power for large installations.

COMPARISON TO OTHER HVAC OPTIONS

The performance of the VRF systems we studied relative to other HVAC system choices can best be understood with the results of calibrated simulation, in which we can make direct comparison of VRF to theoretical, conventional HVAC systems serving the same building loads. To enable this comparison, we built and then conducted detailed calibration of a whole-building energy model (see *Energy Modeling* above for methodology) for each ground-source VRF system. These models include the building envelope and loads as we observed them, and the VRF performance based on our measurements. The utility costs for each model were based on local utility rates paid at each site. Blended electric rates were $0.11/kWh for Madison and $0.097/kWh for Minneapolis, and gas rates were $0.75/therm for both offices.

With the performance of the VRF system model reflecting the actual measured system, we could then swap out that HVAC system in the model for a different conventional HVAC system. The HVAC systems we compared to include:

- **VAV, code baseline**: a VAV system with hot water reheat, modeled according to local energy codes (with ASHRAE 90.1 Appendix G modeling methodology).
- **VAV, common deficiencies**: there has been evidence that packaged, zone-level heating and cooling units are less prone to installation, commissioning, and controls issues than more complex multizone VAV. We therefore included comparison to a VAV system that included such deficiencies, including higher minimum flows, imperfect temperature reset, and increased fan power. This represents the most common HVAC system found in the types of buildings we studied.
- **WS VRF**: identical to the ground-source VRF systems that we monitored, but with a boiler and fluid cooler tempering the condenser water loop instead of a geothermal borefield.
- **WSHP**: a typical water-source heat pump, served by a boiler and fluid cooler. Plant equipment in both this and the previous system are code compliant.

- **GSHP**: a typical ground-source heat pump (two-stage heat pumps), served by a similar closed-loop, vertical GHX as in the systems we studied.

The resulting system comparisons are shown in Table 2 for the Madison office.

Table 2. Comparative analysis of HVAC systems for the Madison office.

	Usage			Savings		
	Energy Cost ($/ft²)	Site Energy (kBtu/ft²)	Source energy (kBtu/ft²)	**Cost ($/ft²)**	As % of HVAC (%)	CO_2e (met. tons)
VAV, common deficiencies	**$1.15**	62	120	**-$0.09**	-14%	-67
VAV, code baseline	**$1.06**	56	109	-----	--	--
WS VRF	**$0.99**	36	103	**$0.07**	10%	-74
WSHP	**$0.98**	38	100	**$0.09**	13%	-30
GS VRF	**$0.97**	35	100	**$0.09**	14%	-58
GSHP	**$0.90**	31	90	**$0.16**	25%	19

These comparisons reveal a number of positive attributes of the ground-source VRF system, as well as a few negative attributes. First, the ground-source VRF system achieves a site energy utilization index (site EUI) of about 35 kBtu/ft²/year, which is significantly better than typical buildings. The system has the lowest operating cost of any system other than the GSHP. It saves about $0.09/ft² versus the code compliant VAV system (14% of HVAC energy), and $0.18/ft² (28% of HVAC energy) versus a more typically deficient VAV system. The GSHP saves an additional $0.07/ft² beyond that of the ground-source VRF, primarily because of the poorer part-load performance in the VRF equipment that we studied, as compared to the rated efficiency of a typical ground-source heat pump. Note that the system as modeled is theoretical and not based on field measurements; there is some uncertainty as to whether the two-stage GSHPs that we modeled achieve their rated performance either.

The ground-source VRF system performs better, but only by a small margin, than the WS VRF and WSHP systems. This suggests two things. First, the addition of the geothermal borefield may have a longer payback than the switch to VRF alone. And second, a water-source heat pump system performs similarly to VRF (assuming quality heat pumps that match their rated performance).

Finally, in comparing the systems impact on climate change, the most useful metric is metric tons of CO_2 equivalent (CO_2e) emissions. By this metric, the ground-source VRF system actually exacts an emissions penalty on the building when compared with the code baseline VAV system, and performs just marginally better than the typically deficient VAV system. This is due to the fact that the natural gas used in the hot water-based VAV system yields lower CO_2e emissions than the electricity-based heating of the VRF system. The GSHP system performs better than all of these in this regard. The numbers in the table do assume that the building owner is purchasing electricity from the grid. As a contrasting example, multiple tenants in the Madison office purchase all-renewable electricity for their space, which technically has zero CO_2e emissions.

We also compared the VAV systems and WS VRF systems using the Minneapolis office model. These comparisons are given in Table 3 for that building.

Table 3. Comparative analysis of HVAC systems for the Minneapolis office.

	Usage			Savings		
	Energy Cost ($/ft²)	Site Energy (kBtu/ft²)	Source energy (kBtu/ft²)	**Cost ($/ft²)**	As % of HVAC (%)	CO_2e (met. tons)
VAV, common deficiencies	**$1.22**	60	136	**-$0.08**	-17%	-4
VAV, code baseline	**$1.14**	54	127	-----	--	--
WS VRF	**$1.28**	50	136	**-$0.14**	-30%	-9
GS VRF	**$1.23**	49	132	**-$0.08**	-17%	-7

The Minneapolis office comparisons all suggest that ground-source VRF is not performing effectively for this building. With the VRF model calibrated to the poor level of performance that we measured in the system, it is performing worse than even the typical VAV system performance, in terms of both cost and emissions.

Sensitivity to Utility Rates

When comparing any heat pump based HVAC system, including VRF, to a hot water based system, the relative price of gas versus electricity becomes a major driver of the outcome. Therefore we conducted sensitivities to test the impact of differing utility rates. In the section above, comparisons were based on local utility rates. These rates were $0.11/kWh (blended) for Madison and $0.097/kWh (blended) for Minneapolis, and $0.75/therm for both offices.

In Table 4 below, we re-analyze system performance with new utility rates. We investigate a 30% increase or decrease in natural gas rates, a common rate of volatility for the fuel. For electric rates, we investigate a 30% decrease, which represents some of the lower electric rates being paid in the Midwest[4]. As Minneapolis and Madison are at the upper end of the range for electric rates in the Midwest, it is not useful to consider higher electric rates. The metric used in this comparison is the savings for ground-source VRF as compared to a VAV system with typical deficiencies.

Table 4. Energy cost savings, sensitivity to changes in utility rates.

	Cost Savings ($/ft²)	
	Madison	Minneapolis
Base savings	$0.18	$0.00
30% higher gas rate	$0.26	$0.03
30% lower gas rate	$0.11	-$0.03
30% lower electric rate	$0.21	$0.03

When utility rates are more favorable to VRF –gas rates are higher or electric rates are lower – the savings by the Madison office increases from $0.18/ft² to between $0.21 – $0.26/ft². Savings realized by the Minneapolis Office increase from essentially zero up to $0.03/ft². On the other hand, if gas rates decrease, savings for the Madison Office drops to $0.11/ft2, while in Minneapolis ground-source VRF yields a small energy penalty.

[4] Results for this section of the report are primarily relevant in the upper Midwest. The HVAC systems being described are typical here, and the results of the energy modeling are heavily dependent on the regional climate.

Heat Recovery Performance

One of the ways that VRF systems save energy is by transferring heat from zones that are in cooling mode (and need to reject heat) to zones that are in heating mode (and need more heat). The HVAC comparisons above included this consideration during energy modeling. We also were able to investigate the potential for heat recovery impact empirically. We investigated this potential by investigating trends of the heating and cooling modes across all of the fan coils in two of the tenant's systems in the Madison office (the design of the Minneapolis Office excluded potential for direct heat recovery).

Recall the typical system shown in Figure 1 had one heat recovery unit (called "branch selectors" by this particular manufacturer) shown for the example office space. In the first system we investigated, 12 fan coils were served by 6 different heat recovery units. Designers use multiple units to reduce refrigerant piping (and therefore cost and complexity) – but with 6 such units, the potential for direct heat recovery is significantly reduced. For example, it may be that two of the zones in the core of the space are in cooling mode, and several perimeter zones are in heating mode. But if the core zones happen to be on a heat recovery unit without any of those perimeter zones, they have no way of directly sending heat to those zones. Upon further analysis, we found that for a sample period in cold months (the only time when any zones are in heating mode) only 15% of the cooling loads that were coincident with a heat load were actually connected to those heating zones and able to transfer heat. The remaining 85% of load was on a heat recovery unit that had nowhere to send the heat other than to the GHX via the condensing units (much of it being from IT closet zones).

In the second system we investigated, 12 fan coils were being served by 4 different heat recovery units. This system not only had fewer heat recovery units, it also had slightly more thoughtful layout of the fan coils. In this case 57% of the theoretically recoverable energy during our study period was able to be transferred. This is better, but still demonstrates that even with thoughtful design more than 40% of the heat recovery potential can be lost if several heat recovery units are needed in a system design.

The energy model of the Madison office identified $770 in energy cost that could theoretically be saved with heat recovery, given ideal zone connections. Applying the empirical derating factors above to the all tenants across the building based on the ratios of heat recovery units to fan coils in each, we estimate that the actual savings was probably closer to $330 annually, or 43% of theoretical. Model results were adjusted accordingly, and are reflected in the results throughout this section.

In the Minneapolis Office, every fan coil is piped from its own heat recovery unit, so there is zero potential for direct heat recovery in that system.

ECONOMIC SUMMARY

System First Costs

Regardless of type, advanced HVAC systems generally come with a cost premium either because they contain more complex, costly parts, are more difficult to install and startup, or simply because there is less experience in the contractor community with the systems. We were able to collect first cost premium data for the Madison Office, based on bids received to install various systems from the mechanical contractors on the project. This data is shown in the first column of Table 5. This is compared to data from similar projects in the same timeframe, shown in the second column (this data includes the data from the first column, as well as additional projects). Where 'N/A' is shown, there is no additional cost data beyond the Madison Office. No cost data was available for the Minneapolis Office, so payback analysis will focus on the Madison Office.

Table 5. First cost premiums of VRF and other systems, as compared with conventional VAV

	Madison Office $/ft^2	General $/ft^2
VAV (Baseline)	----	----
WSHP	$0.46	$0.39
GSHP	$3.77	$3.75
VRF, air-source	$1.15	$0.85
VRF, water-source	$3.35	N/A
VRF, ground-source	$5.38	N/A

Considering all these options, the ground-source systems are generally more expensive than the others, with a $4.0 – 5.7/ft^2 premium over the baseline system. The next most costly system is water-source VRF, with a premium of about $3.3/ft^2. WSHP and air-source VRF are intermediate system options with premiums of between $0.4-1.2/ft^2. Note that this data includes a 10% federal tax credit for the cost of the geothermal equipment in each ground-source system.

Payback

In many cases it is not obvious whether this first cost premium above is justifiable based on energy savings. We have therefore completed a payback assessment based on the benefit of the energy cost saved (from study results above, specifically those in Table 2), as well as predicted maintenance cost savings based on conversations with facility operators and owners. We've completed this analysis compared to a baseline VAV system (with typical deficiencies).

This analysis does not include other benefits such as incentives, increased productivity, carbon credits, etc. This assessment is valid for building design teams or owners looking to incorporate the technology, and also for utility program personnel who need this type of information to implement and evaluate programs that may incentivize VRF. Recall that all analysis in this report is for the utility rates being paid in the region at the time: $0.11/kWh (blended) for Madison electricity and $0.097/kWh (blended) for Minneapolis, and $0.75/therm for both offices. The analysis results are laid out in Table 6.

Table 6. Economic analysis of HVAC system options.

	First Cost Premium $/ft^2	Energy Savings $/ft^2/year	Maintenance Premium $/ft^2/year	**Simple Payback years**
VAV (baseline)	---	---	---	**---**
WSHP	$0.46	$0.18	$0.020	**3**
GSHP	$3.77	$0.25	$0.015	**16**
VRF, water-source	$3.35	$0.16	$0.015	**23**
VRF, ground-source	$5.38	$0.18	$0.008	**31**

The simple payback happens to increase in the order of the systems shown. The WSHP system has the shortest payback at just 3 years. The GSHP system pays back in about 16 years. The two VRF systems have paybacks longer than 20 years, with the water-source approach at 23 years and the ground-source approach at 31 years due to the additional cost of the GHX.

KEY CONCLUSIONS

We can draw several significant conclusions from the results above. In this section, we first discuss the quantitative conclusions that were primary objectives of the study. We then shift to discussing other lessons learned about ground-source VRF systems.

KEY QUANTITATIVE CONCLUSIONS

Performance of the VRF equipment compared to ratings. We were able to isolate and measure the performance of just the VRF equipment, separate from pumps, GHX, and other ancillary equipment. The results show that in heating mode, the VRF systems that we measured meet the high level of performance their ratings suggest, but in cooling mode they do not. To be specific, in cooling mode system performance fell short of ratings by 8-34%. More importantly the part load performance of the systems, which more accurately represents the broader average performance of the systems, fell significantly short of ratings in almost every measurement taken. In the case of the Madison office, part load performance fell short by more than 50%. This is most evident in the seasonal energy profiles of the systems, in that they do not see significant decrease in energy usage in the spring and fall seasons when outdoor temperature moderates, and loads are very low.

We have identified areas for potentially improving this performance. First, the Madison office was not programmed with a night setback due to concerns about morning warm-up. The Minneapolis office did have a setback included. Secondly, there is evidence that in some of the more open spaces in the Madison office there is simultaneous heating and cooling between adjacent zones, as well as oscillation between heating and cooling within a given zone. This is a potential issue for VRF systems in larger open spaces; we are unsure of the capability of adjusting the thermostat/control loop to mitigate this. Finally, none of the systems that we observed were capable of shutting off compressors regularly, or modulating the fluid flow on the water side based on the number of compressors on. This is common in heat pump equipment but for some reason not allowable with the VRF models we observed. This leads to not only significant compressor usage during times of low load, but siginficant pumping penalties as well. It seems like an issue the manufacturer could improve on.

Performance of associated equipment. Any water-source VRF system does not depend solely on the VRF equipment, but is supported by water-side equipment. In this case, pumps and a GHX are used to provide a water loop that acts as a heat source and/or sink for the system. The GHX performed well in both of the buildings we studied, providing fluid between 40-80ºF in the Madison office, and 50-87ºF in the Minneapolis office. These ranges are similar to the ranges that the systems were designed for, and we therefore conclude that the GHX had their intended effect on the VRF systems, and did not serve to irregularly help or hinder VRF performance.

This tempered water loop is not provided free of energy consumption though; pumping is a significant component of HVAC energy consumption in both office: 8% in Madison, and 31% in Minneapolis. So even if VRF equipment does perform well in comparison with conventional HVAC equipment, we must take into account the pumping power (both at full and part load) to understand the full performance of the system, and to design an efficient system. One strategy that the Madison Office utilized with this goal in mind was sizing two pumps each at 70% of full load; this provides a measure of of redundancy while allowing for smaller pumps that are capable of modulating to lower power during times of low load.

Performance compared with other HVAC system types. Though the VRF equipment performed worse than expected, it still had the potential to outperform many conventional HVAC system types in a simulated comaprison (using well-calibrated models). See the *Comparison to Other HVAC Options* section for significantly more detail. In short, for the Madison office, where the VRF equipment performed better, the ground-source VRF system performed better than VAV (savings of $0.18/ft^2), WS VRF ($0.02/ft^2), and WSHP ($0.01/ft^2). We did find that GSHP outperformed the ground-source VRF, with an additional $0.07/ft^2 in savings. In the Minneapolis office where the VRF did not perform as well, it did not save energy compared to any of these systems; it broke even compared to VAV. Note that the comparisons assume that HVAC equipment (heat pumps, rooftop units, etc.) being compared to performs according to its ratings – there is insufficient evidence to guarantee that this is always true. There is a significant enough increment between ground-source VRF and GSHP to conclude that, even with some performance degredation, the GSHP system will generally use less energy.

As the performance of the WSHP, WS VRF, and ground-source VRF systems were quite similar in all cases, a designer and owner of a building like these two might best decide between these systems based more on non-energy impacts, such as space requirements, their own maintenance capabilities, acoustics, and other less quantitative considerations described below.

Impact of gas and electricity prices. In comparing the ground-source VRF system to HVAC system types that use natural gas for heating, such as VAV and WSHP systems, the price of natural gas relative to electricity has a substantial impact on the results. If gas prices decrease by 30% from their current level, the stated savings decrease by 39%. If gas prices increased by 30%, the stated savings would increase by 44%.

Economics. Using costs from the time of construction, the VRF systems analyzed here tend to have a longer payback than other typical heat pump systems at current utility prices. The higher performing of the two systems, in the Madison office, suggests a payback of over 20 years for VRF (31 years for ground-source VRF, and 23 years for water-source). Admittedly, simple payback is an oversimplification of most owner's financial operations – a rate of return may be more applicable. But with the significant spread in the payback results, it is fairly clear that though VRF does save energy, its cost creates a more challenging economic scenario than other high performance systems such as GSHP or WSHP. Owners who are choosing which high performance HVAC system to employ may therefore want to choose VRF over other heat pump options only if there were significant qualitative reasons for doing so. In the next sections we explore some of those qualitative impacts.

Note that this economic analysis used VRF costs from these systems' construction in 2013; one mechanical contractor told us that, relative to other systems, VRF costs have decreased as contractors become more experienced. They even suggested that in their area, VRF (without a GHX) is 'nearing' the cost of VAV with hot water.

Emissions. In comparing different systems' impact on climate change, the ground-source VRF system actually exacts an emissions penalty on the building when compared with many system types, due primarily to the comparison to cleaner burning natural gas. This does assume that the building owner is purchasing electricity from the general grid; as a contrasting example, multiple tenants in the Madison office purchase all-renewable electricity for their space, which technically has zero CO_2e emissions.

OTHER LESSONS LEARNED

Throughout this study we made other, qualitative observations on the design and operation of VRF systems. We also had broad access to many people with experience designing, installing, commissioning, operating, and owning the two systems we studied. We've compiled these qualitative lessons below.

Design practices. First of all, VRF is a relatively new system type for many building professionals. Some design/build mechanical contractors are now more than able to handle design of VRF. But if an owner or general contractor does not have extensive experience with VRF, it is best to have an engineer complete a full design first, especially one with experience. Otherwise the owner/contractor won't have the confidence necessary to interact with the mechanical contractor and/or VRF vendor.

From the engineer's perspective, a few design challenges were specifically mentioned on these projects that other engineers should be aware of. First, in a multi-tenant building it can be a challenge to size the condensing units for a space before the tenant is known. With the modularity of the units, consider oversizing the mechanical room a bit, but waiting to actually select and purchase the condensing units until the tenant is known. Also, the engineer should be aware of specific constraints in the length and rise of refrigerant piping – there are limits to all such dimensions. It is best to keep runs as short as possible to prevent losses in capacity; multiple condensing unit locations (such as one per floor, as in the Madison Office) can help with this. In any case, you can rely on the manufacturer for these rise and run constraints.

Size and shape. The small space requirement of VRF, due to the size of its refrigerant piping compared to ductwork or even hydronic piping, can lead to additional cost efficiencies. Some new construction projects can reduce their floor-to-floor heights, thereby saving substantial first cost for building structure and finishes. VRF's small space requirements can also increase a buildings usable square footage, as was done in the Madison Office where increased rentable space led to increased revenue. Additionally, one mechanical contractor stated that the modularity and small size afforded excellent flexibility to move system components around and make adjustments in the field, which reduced their cost, especially in contingency.

Additionally, the modular nature of the condensing units in a VRF system allow heating and cooling energy to be easily metered at the tenant level, increasing savings due to tenant behavior.

Sound. The operator of the Madison Office stated that the VRF systems they've used (on multiple buildings) have been quieter than heat pump systems.

Maintenance and operation. Maintenance has been reported to be similar to most higher-performance systems. VRF may still be slightly more costly to maintain than a traditional rooftop VAV system due to the additional knowledge needed for some operational tasks. The operator of the Madison Office has been very happy with the operation of their system; while they have had significant compressor issues on air-source VRF buildings, they have not needed to replace any ground-source compressors in the couple of years of operation. The operator of the Minneapolis office has not had any maintenance issues, but has had an issue with cold calls in conference rooms.

Water-source vs. air-source. The operator of the Madison Office has operated both air-source and ground-source systems. In the cold winters of the upper Midwest, air-source units dramatically lose capacity at low ambient temperatures and often need to be supplemented by an additional heat source at the outdoor units in order to meet loads. Often this is done with gas-fired equipment, necessitating a natural gas connection, some type of enclosure, defrost, freeze protection, associated controls, and all the maintenance required for each of these items. At the air-source site operated by this particular operator these items have been a maintenance and control "disaster" (in the owner's words), adding significant labor and cost. In contrast, they have had none of these issues with the ground-source VRF system. They do note that their air-source VRF system was installed several years ago, and design of air-source systems in cold climates has improved significantly since then (though it still remains arguably more complex to operate than a water-source system).

Occupant satisfaction. Occupant satisfaction from those we've worked with has been mixed. As we mentioned above, the Minneapolis office has had issues with cold calls in conference rooms. On the other hand, they mentioned that comfort with the system in office spaces was similar to other systems. Contractors have reported a similar number of post-occupancy issues that they've had with other system types. One tenant in the Madison Office reported uncomfortable swings in conference room temperature as well.

Quantitatively, our data shows that temperature control is about as cyclical as other more conventional HVAC systems. There is generally a 2-3ºF range to the control loop (the air temperature in the space increases and decreases within a range of 2-3ºF as the control determines how much heating or cooling to provide).

Installation. The primary feedback we received on installation of VRF systems was regarding refrigerant piping. Specifically that with the amount and complexity of the refrigerant piping and connections on a VRF project, the refrigerant piping skill of the contractor is critical.

Commissioning. Commissioning of these systems did take somewhat longer than a traditional system; it's difficult to say how much of this was due to the uniqueness of the systems. Thermostats needed to be tuned to avoid swings in temperature (or between heating and cooling mode) that did cause significant occupant discomfort in the first few weeks of operation. Complaints were particularly bad in this early period during the coldest days. Simultaneous heating and cooling between adjacent zones and some oscillation between heating and cooling within a zone were also addressed in this commissioning period.

These issues were solved by both the vendor and contractor spending additional time at the site, primarily tuning thermostat control loops. VRF systems, including both of those we observed, generally have integrated control systems from the same manufacturer as the HVAC equipment itself. This means that contractors and those providing startup and commissioning must know these specific proprietary systems. In the case of the Madison office, this meant leaning heavily on the vendor to provide some of these services.

The nature of VRF's proprietary controls will often also require that a second, higher-level building automation system be installed. For example, in the Madison Office the VRF thermostats were Daikin controls, and these were tied into a central interface where basic elements of these controls could be modified across the building. Outputs from this Daikin system were then connected to an overarching BACnet compatible BAS. As a final layer, the BAS also fed a building dashboard where performance could be seen by the public via the internet. Not all buildings should pursue this depth of control capability due to its associated costs and complexity. But those that need the functionality should realize that this type of control system hierarchy will often result with VRF systems.

Since the systems were commissioned, they have tended to work well with little additional oversight.

Safety. VRF systems can be built and operated safely. A large number of systems, including both those we studied, have operated without any incident at all. But due to the large amount of refrigerants in the system, the designers need to take key precautions. ANSI/ASHRAE Standard 15 *Safety Standard for Refrigeration Systems* must be adhered by to ensure safety when designing the refrigerant loop. In short, the refrigerant in a given loop must not cause a danger if a leak occurs in the smallest room(s). In the designs that we looked at, designers made their smallest rooms comply by simply moving fan coils outside such rooms (and ducting the air into the room) or by eliminating dropped ceilings (increasing room volume). Installation can also mitigate safety hazards. Contractors should braze all joints; no flared fittings should be used. Thorough pressure testing should also be conducted for each circuit.

FUTURE WORK

As with any newer technology, there is certainly more research to be conducted. First, this study, along with the study of Oakland University, result in a total of just three buildings with this system type that have been studied. More field study is needed to understand performance in different building types, or with different system configurations such as a water-source VRF system with a boiler and fluid cooler serving the water loop. In addition, all three of these systems happened to use Daikin equipment; other types of equipment need to be measured.

There is also research needed to improve the operation of the equipment. First and foremost, the poor part load performance of these systems needs to be better understood. Is it due to simultaneous heating and cooling with adjacent zones? Is it a controls issue integral to the condensing units? Is it simply the compressor performance? How much is the lack of modulation on the waterside impacting part load performance? These questions were not able to be answered with the level of metering that we had available, but their answers would inform future systems signficantly.

Finally, air-source VRF systems are noticeably absent from the *Comparison to Other HVAC Options* in this report. This is because we do not feel that field research of air-source VRF systems in cold climates has been adequate enough to make such a quantitative comparison. Essentially every air-source VRF study that has created a specific benchmark of air-source VRF performance in a cold climate has relied on modeling. A significant field study of the specific performance of air-source VRF systems in cold climates is badly needed, as these systems gain a signficant share of the market.

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