Application of Air Source Variable Refrigerant Flow in Cold Climates

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Authors

Scott Schuetter, Seventhwave

Scott Hackel, Seventhwave

749 University Row, Suite 320

Madison, WI 53705

608.210.7120

www.seventhwave.org

Project Manager

Scott Hackel

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INTRODUCTION

Variable refrigerant flow (VRF) systems use variable speed, split air source (or water source) heat pumps to provide space heating and cooling to a building's conditioned areas. VRF systems can condition multiple zones in a building, each of which may have different heating and cooling needs. Using sophisticated control technologies, VRF systems' variable speed allows them to modulate the amount of refrigerant sent to each zone independently and in tune with diverse and changing space conditioning loads, thereby increasing energy savings. The key components of a VRF system are the outdoor unit (often called a condensing unit), indoor unit, refrigerant, and heat recovery unit. Refrigerant flows between all three components of the system, making this akin to a split system, albeit more complex. VRF systems save energy in four primary ways:

- Distribution of heating/cooling using refrigerant instead of air
- Variable speed compressors and fans
- Zone-level heating and cooling, providing only the needed heating and cooling without reheat
- In some systems, recovery of heat from cooling zones to heating zones

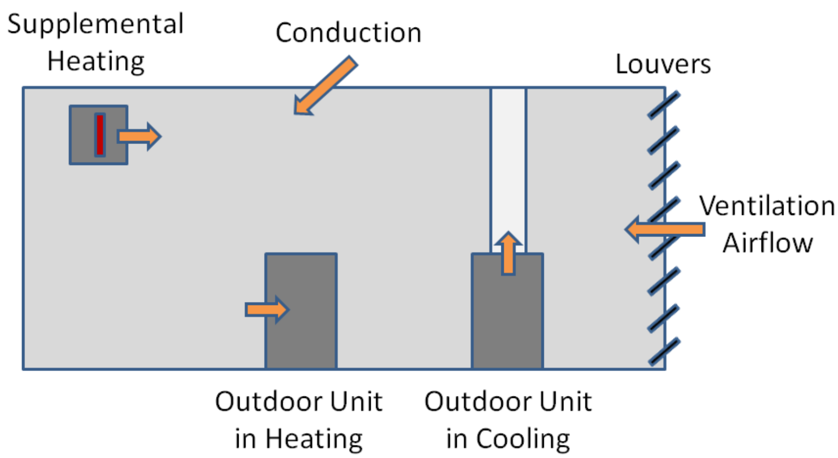
The main heating and cooling plant of a VRF system is usually an air source or water source heat pump. In an air source system, the outdoor unit exchanges heat with the outdoor environment, either by expelling heat (when cooling) or absorbing heat (while heating).

Air source VRF systems perform best in moderate climates, as they typically lose capacity and efficiency at low ambient temperatures – or moderately low wet bulb temperatures where defrost is required – and may be supplemented by an additional heat source. In colder climates, this often necessitates the addition of a supplementary heater within a partially-enclosed mechanical room housing the outdoor units. Often the supplementary heater is gas-fired, necessitating a natural gas connection and its associated cost. Alternately, low temperature performance may be mitigated by upgrading to a water or ground source system, but any additional first cost implications should be considered. Because of their lower capital cost, the majority of VRF systems being implemented in cold climates today are air source. However, significant questions remain regarding how to design and operate air source systems at low temperatures. **This white paper outlines the optimal control strategies for designing and operating air source VRF systems in cold climates.**

METHODOLOGY

To that end, we developed a model capable of predicting the energy consumption and corresponding utility cost of an air source VRF system under a variety of scenarios. At its core, the model is an annual, hourly energy balance of a mechanical room containing outdoor VRF units. A diagram of the components of the model is illustrated in Figure 1.

Figure 1: Energy balance of mechanical room



The heat flows within the mechanical room include:

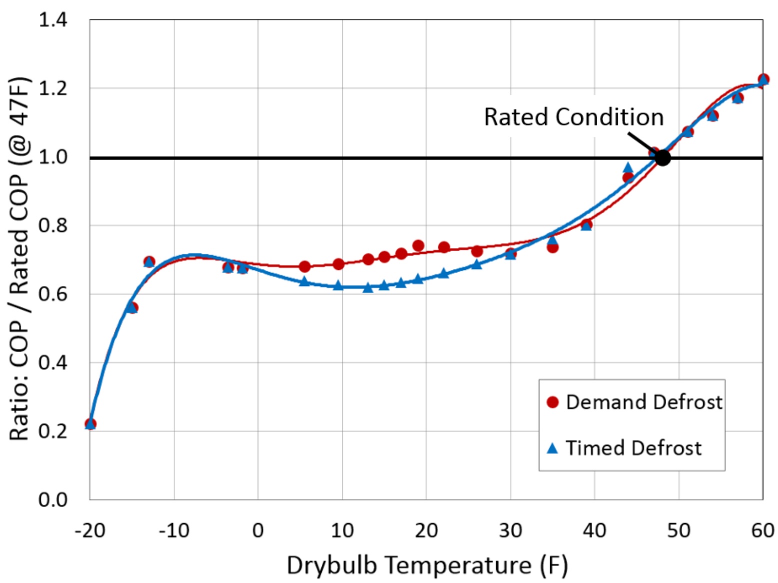
1. Outdoor Units in Heating: Heat extracted from mechanical room and used to heat the building, based on heating demand of building
2. Outdoor Units in Cooling: Heat rejected to mechanical room (and generally from there, directly to the outdoors) and used to cool the building, based on cooling demand of building
3. Ventilation Airflow: Heat associated with air moving through louvers to ventilate the mechanical room, based on wind-driven or fan-assisted airflow (and outdoor air temperature)
4. Conduction: Heat flow either into or out of the mechanical room, based on outdoor air temperature and amount of insulation (this flow is generally very small compared to the others)
5. Supplemental Heating: Heat provided by a supplemental heater, at times when the other heat flows would otherwise cause the temperature in the mechanical room to drop below the given setpoint

The outdoor conditions are based on TMY2 data for the respective location, which represents typical weather, for cold climates. The heating and cooling loads driving the operation of the system in the building are estimated from building energy models created in DOE2 to represent typical office, multifamily, lodging, and public assembly buildings in a few locations in the upper Midwest. These models represent building zone loads *without ventilation*, since most VRF systems utilize a dedicated outdoor air unit with heating and cooling coils that can be served by any system type, not necessarily as part of the VRF.

The predicted heat flows from the energy balance are used to predict the annual energy consumption from the following end uses:

Outdoor Units (in heating or cooling). The building loads define the amount of load put on the outdoor units. Commercial buildings often experience simultaneous heating and cooling loads, as perimeter zones need heating during colder periods, while heavily occupied core zones may still require cooling. Our model is capable of handling this situation, by assigning different outdoor units to either handle heating or cooling loads for a given hour. When coupled with heating and cooling efficiency, these loads define the electricity consumption of the outdoor units. The outdoor units' heating capacity and efficiency are highly dependent on the mechanical room air conditions (dry bulb and wet bulb temperature), as well as their part load operation serving the demand side (zoned indoor units) of the system. Figure 2 illustrates the heating efficiency as a function of mechanical room dry bulb temperature, the shape of which is a major influence on the results of this study. Though this plot is a function of dry bulb, heating performance is also impacted by humidity – or wet bulb temperature – primarily due to defrost from 0º to 35ºF.

Figure 2: Heating efficiency as a function of mechanical room dry bulb temperature; the Rated Condition reflects AHRI testing conditions.



This curve is defined as the ratio of the actual heating efficiency at a given temperature to the rated heating efficiency at 47 °F. The curve is therefore a multiplier that is used to derate the rated efficiency based on a given hour's dry bulb temperature. So, at the rated temperature, the modifier is equal to 1. Below this temperature, the modifier is less than 1, resulting in a decreased heating efficiency. For the majority of the temperatures associated with heating, the modifier hovers around 0.7, resulting in a 30% decrease in heating efficiency. However, at very cold temperatures (less than -10 °F), the modifier rapidly decreases, resulting in heating efficiency degradation of up to 80%. It is also worth noting that there is a difference in heating efficiency for timed and demand defrost methods.[1] These methods are discussed in more detail in the *Defrost Strategy* section below.

Supplemental Heating. The supplemental heating device may be defined as using electricity or natural gas to maintain the mechanical room temperature setpoint. Therefore, supplemental heating is assumed off whenever the outside air temperature is higher than the control setpoint.[2] For periods colder than this setpoint, the energy balance defines the amount of load that the supplemental heating device must meet to maintain the setpoint. A simple heating efficiency is used to calculate corresponding electricity or natural gas consumption of this unit.

Auxiliary Heat (Baseboard). The sharp decline in heating efficiency at low dry bulb temperatures illustrated in Figure 2 is accompanied by a similar decline in heating capacity. For low mechanical room setpoints (less than -10 °F), the VRF system may not have enough capacity to meet the building's heating demand. When this occurs, an auxiliary heating source, such as baseboard, is used to meet the remaining

[1] Note that though the chart shows general heating performance based on dry bulb temperature, the impact of defrost is more directly driven by wet bulb temperature. Our handling of this in analysis is discussed in later sections.

[2] In actuality the supplemental heating is controlled based on the temperature in the mechanical room; our control based on outside air temperature is a simplifying assumption.

demand. This source may be defined as either using electricity or natural gas with a simplified efficiency to then calculate energy consumption.

Fan. In the Fan Assisted control strategy (see below), a fan is assumed to consume electricity in order to deliver enough air from outside to the outdoor unit.

A significant decision for design and operation of an air source VRF system is mechanical room control strategy. The following strategies were analyzed and compared to determine the optimal approach.

Fixed Louvers

Under this strategy, louvers on the wall of the mechanical room are always open, regardless of the outdoor air dry bulb temperature. The airflow rate (and associated heat flow) into the mechanical room is estimated for each hour based on calculations outlined in ASHRAE Fundamentals Chapter 24. The calculations take into account the wind speed, wind direction, and louver geometry.

Operable Louvers

Under this strategy, the louvers are open when the outside air dry bulb temperature is higher than the mechanical room temperature setpoint, and air flows freely to the outdoor VRF units. When the outside air dry bulb temperature is lower than the mechanical room temperature setpoint, the louvers are closed.

Fan Assisted

Under this strategy, a supplementary fan is used to deliver air to the outdoor air unit with a fixed flow rate when outside air temperature is higher than the mechanical temperature setpoint. When the outside air dry bulb temperature is lower than the mechanical room temperature setpoint, the fan is off, with no associated outdoor air flow or electricity consumption.

Table 1 shows the model's input screen, and the corresponding default values of each input.

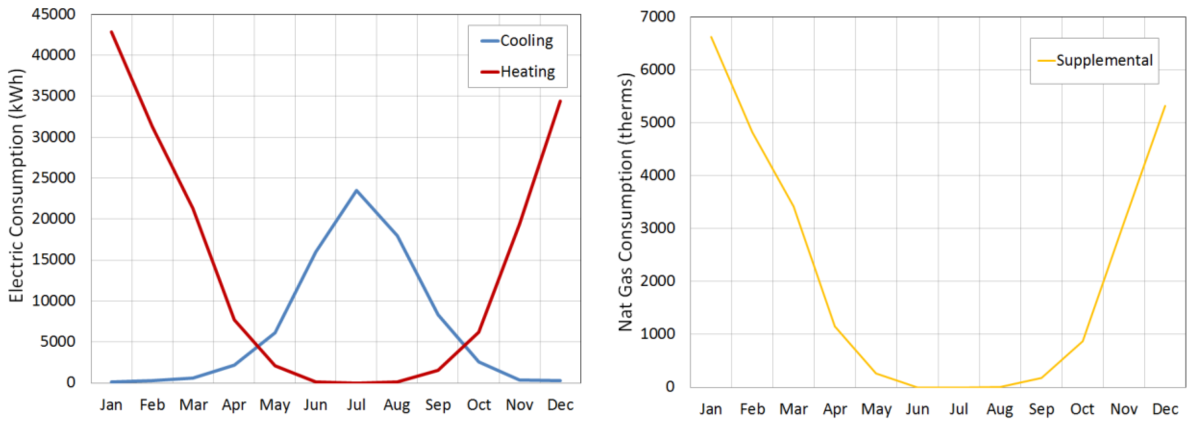
Table 1: Default model inputs

Description	Units	Value
Building location	-	Madison
Building type	-	Office
Building area	ft^2	100,000
Natural gas cost	$/therm	$0.70
Electricity cost - consumption	$/kWh	$0.07
Electricity cost - demand	$/kW	$7.00
VRF model number	-	RXYQ408
Cooling efficiency, EER @ DB = 95 F and WB = 75 F	BTU/hr/W	10.90
Heating efficiency, COP @ DB = 47 F	-	3.25
Defrost type	-	Demand
Supplemental heating fuel type	-	Gas-fired
Supplemental heating efficiency	%	80%
Mechanical room temperature setpoint	F	50.0
Mechanical room ventilation type	-	Operable
Mechanical room ventilation fan power	bhp	0.0
Mechanical room ventilation fan flowrate	ft^3/min	55,000
Baseboard heating type	-	Electric
Baseboard heating efficiency	%	100%

RESULTS

Once a model of the VRF system and mechanical room was built, we used it to explore many different design decisions and their impacts on energy and cost. Our base case for these considerations was a 100,000 square foot office building in Madison, Wisconsin. Figure 3 illustrates the monthly electricity and natural gas consumption predicted by the model for default model inputs, which include a mechanical room temperature setpoint of 50°F, gas-fired supplemental heat, and operable louvers.

Figure 3: Monthly electricity and natural gas consumption using default model inputs



As expected, the heating from both the outdoor units and supplemental devices increases during the colder winter months. On an annual basis, this equates to a utility cost of $0.48/ft^2 for the VRF components of the building (uses such as lighting and equipment, as well as ventilation air, are ignored).

MECHANICAL ROOM CONFIGURATION

Designers of air source VRF systems have approached the challenges of low ambient temperature, and its impact on outdoor units, in a variety of ways.

Table 2 outlines the results for the four most common configurations: no supplemental heat, supplemental heat with fixed louvers, supplemental heat with operable louvers, and supplemental heat with fan assist. Note that these results correspond to a mechanical room setpoint of -10ºF (which is at the low end of the optimum range outlined in the *Optimal Mechanical Room Temperature* section).

Table 2: Annual utility cost, EUI and supplemental heat capacity for different mechanical room configurations

Configuration	**Annual Utility Cost ($/ft^2)**	**EUI (kBTU/ft^2)[3]**	**Supplemental Heat Capacity* (BTU/hr-ft^2)**
No supplemental	$0.43	11.3	n/a
Supplemental, fixed	$0.40	12.0	14.3
Supplemental, operable	$0.39	11.7	9.6
Supplemental, fan assist	$0.44	13.6	9.5

* VRF is assumed to only serve zone loads; our model assumed ventilation for the building was served by a different system.

The operable louver configuration has the lowest annual utility cost and EUI, followed closely by the fixed louver case. Also included in Table 2 is the associated supplemental heat capacity. Note that the

[3] Within this white paper, Energy Utilization Index is defined as site, as opposed to source.

fixed louver case requires a higher supplemental heat capacity than the other cases, in order to accommodate the airflow through the louvers every hour of the year, including those with an outdoor dry bulb below an optimal mechanical room setpoint.

We also determined the impact of supplemental fuel type on system performance. Table 3 illustrates the trade-off in annual utility cost and EUI between different supplemental heat fuel types. Note that these results reflect the supplemental heat with operable louver case.

Table 3: Annual utility cost and EUI for different supplemental heat fuel types

Supplemental Heat Fuel Type	Annual Utility Cost ($/ft^2)	EUI (kBTU/ft^2)[3]
Natural Gas	$0.39	11.7
Electricity	$0.43	11.5

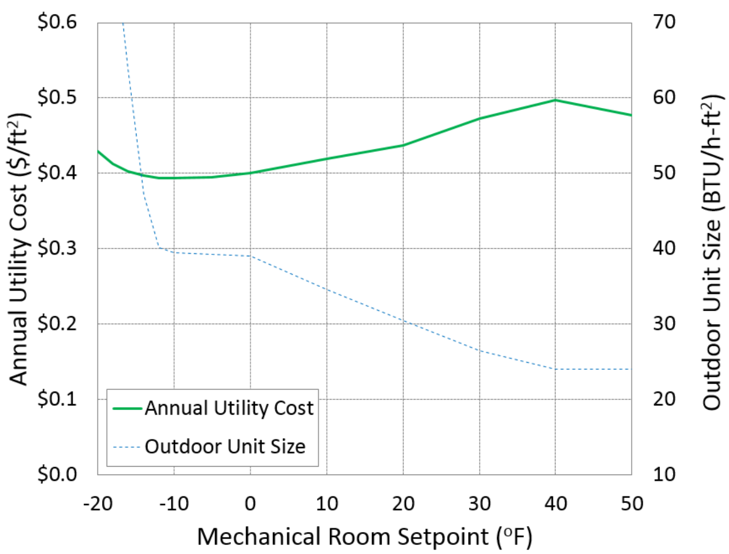
Due to natural gas' relative low cost, using gas-fired supplemental heat results in lower annual utility costs. **Therefore, our analysis indicates that the optimal mechanical room configuration is one with natural-gas fired supplemental heat and operable louvers**.

OPTIMAL MECHANICAL ROOM TEMPERATURE SETPOINT

Assuming that some amount of supplemental heat is provided in the mechanical room, an important consideration for the design engineer is the optimal outdoor air temperature setpoint at which the supplemental device will activate, the temperature at which the louvers will close, and/or the temperature at which the fan assist will turn off (discussed above as the mechanical room temperature setpoint).

Our analysis shows that for base conditions in Madison, this outdoor temperature setpoint that achieves optimal operating cost is somewhere between -12°F and 5°F. Figure 4 shows the impact of this control setpoint on operating cost. Though the exact optimum appears to be around -10°F, the cost impacts are negligible between -12°F and 5°F (they vary by 4% at the most). The designer should select the system in this range that has the lowest first cost and maintenance potential (considering size of unit heater and outdoor units). A conservative approach may be to design for 0°F.

Figure 4. The energy cost of operating the system as a function of mechanical room setpoint

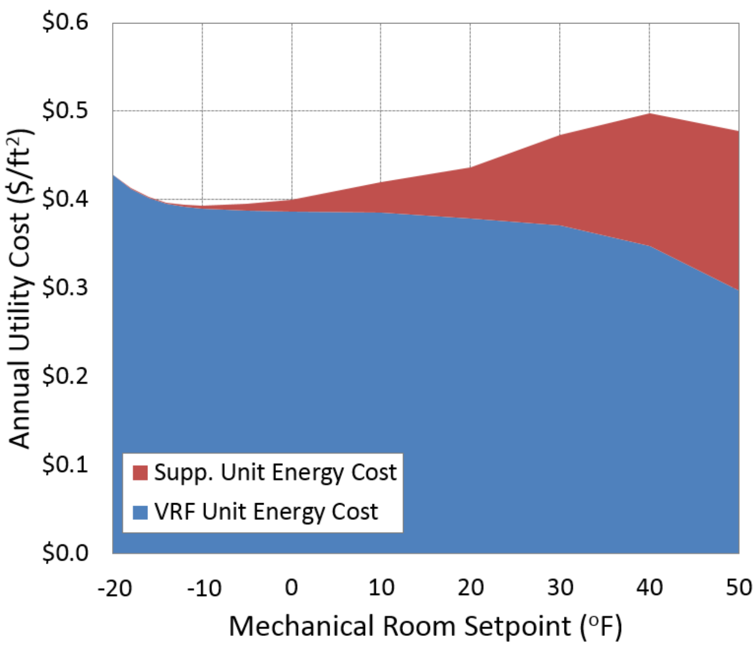


With current gas prices and the default inputs, this optimal range holds for:

- Either demand or timed defrost, and
- All mechanical room configurations we studied.

It is important to note that the most significant factor influencing the mechanical room setpoint for optimal energy performance is the VRF unit's heating performance curve. This is shown more clearly in Figure 5, where the annual cost curve is broken into the supplemental and VRF unit energy costs. Our analysis indicates that **the optimal mechanical room setpoint occurs at a temperature close to, but higher than, the temperature at which the VRF system's heating efficiency and capacity are significantly reduced**. For the performance curve we analyzed (illustrated in Figure 2), this occurred at -10ºF. For other VRF equipment types, the optimum would be higher or lower based on its specific curve.

Figure 5. The energy cost of operating the system, by unit, as a function of mechanical room setpoint



It is worth considering the impact of natural gas prices, since use of supplemental heat is essentially a trade-off between gas and electricity). We found that this general temperature range of optimal performance held until gas price dropped below $0.25/therm and high efficiency (94%+) condensing supplemental unit heaters were used. Under this scenario, the optimal operating point jumps up to 35-40ºF, suggesting operating the room above freezing. This jump avoids operating setpoints at which the defrost cycle has the largest negative impact on the system. Gas prices have dropped significantly recently, but most customers (other than perhaps the largest buyers) are not paying rates near $0.25/therm yet.

Another option in determining this setpoint (which was not investigated with our model) is a dual temperature control scheme. With this approach, the supplemental heating and louver option is still activated below an outdoor temperature setpoint as discussed above. However, in a dual temperature scheme, whenever the outdoor temperature drops below the setpoint, the supplemental device would be used to maintain the temperature at the outdoor units at a second temperature, well above that setpoint (perhaps at the 35-40ºF range identified in the low gas price sensitivity) to provide improved VRF efficiency. Determining the performance of this scheme should be addressed in future research. The tradeoff between improved VRF unit performance and mechanical room losses at high ΔTs (to the ambient) would have to be calculated. It is important to note that in any scenario (low gas prices, dual temperature, etc.) where the mechanical room air were to be heated to high above outdoor temperature, the mechanical room should be configured to allow the discharge air from the outdoor unit to stay entirely within the room. Some current approaches have 100% of this air hard-ducted to the outside.

But with our base scenario and current gas prices, our analysis suggests that it remains optimal to design mechanical rooms to accommodate freezing temperatures, so that outdoor units can be operated with untempered outside air near 0°F. Significant thought should go into defrost and condensate handling at sub-freezing temperatures: equipment should be built, configured, and placed such that defrost and condensate water can be collected and flow to a drain with limited exposure to the cold mechanical room along the way. A hard connected, insulated drain directly below the outdoor unit equipment is one possible solution.

Condensate may also be generated by the unit heater, if a condensing version is selected for higher efficiency. Based on the results above, if a low temperature (e.g. 0°F) control setpoint is selected, a non-condensing (80% efficient) unit heater should be chosen. The low number of operating hours result at this low setpoint result in only 1% savings from upgrading to a high efficiency unit. But if some other scenario dictates a higher temperature (e.g. 40°F) setpoint or a dual temperature scheme, then the mechanical room will remain above freezing during unit heater operation. In addition, the supplemental heater will be operating much more often. For both of these reasons, a condensing unit heater (94%+ efficiency) should be selected to lower gas consumption in these scenarios.

Note that these optimal temperatures are for minimizing annual energy cost. If an overall economic optimum is desired the first cost of different sizes of outdoor unit must also be considered. Note in Figure 4 the change in outdoor unit size that is required with lower mechanical room setpoints.

DEPENDENCE ON CLIMATE AND BUILDING TYPE

We expanded our analysis to different cold climates and buildings types to understand the effect on optimal design and performance.

Climates

We selected two additional climate locations that span the range of design conditions in the upper Midwest; Detroit and Minneapolis. Table 4 outlines the heating design conditions in each location, as well as the corresponding results of the modeled base case.

Table 4: Design conditions and corresponding results in different cold climates

Location	Heating Design Temp (F)[4]	Heating Degree Days (day-F)[5]	Annual Utility Cost ($/ft²)	EUI (kBTU/ft²)[3]
Detroit, MI	2.9	6103	$0.35	10.0
Madison, WI	-7.0	7104	$0.40	13.0
Minneapolis, MN	-11.2	7472	$0.45	16.1

Detroit is generally not as cold as Madison, with a higher heating design temperature and fewer heating degree days, and Minneapolis is colder than Madison, with a lower heating design temperature and more

[4] Dry bulb temperature (99.6 percentile), from ASHRAE Handbook Fundamentals Chapter 14Heating
[5] Heating degree days (65 °F)

heating degree days. As expected, higher energy costs corresponded to colder climates, with a strong correlation between heating degree days and annual utility cost. Note that these results correspond to gas-fired supplemental heat with operable louvers and a mechanical room setpoint of 0ºF. A study of Minneapolis' optimal mechanical room setpoint corresponded to the results for Madison. Interestingly, Detroit's heating design temperature approaches the upper end of our optimal setpoint. This suggests that Detroit may be at the margin of climates that need supplemental heat at all.

Building Types

We also created three additional load profiles corresponding to building sectors typically served by air source VRF: multifamily, public and lodging. As expected each building type's load profile was different, based on their respective DOE2 models. Table 5 outlines the results for the different building types, all set in Madison. Note that these results correspond to a mechanical room setpoint of -10ºF.

Table 5: Annual utility cost, EUI and supplemental heat capacity for different building types

Building Type	**Optimal Mech. Room Configuration**	**Annual Utility Cost ($/ft^2)**	**EUI (kBTU/hr-ft^2)[3]**	**Supplemental Heat Capacity (BTU/hr-ft^2)**
Office	Gas supplemental, operable louvers	$0.39	11.7	9.6
Public	Gas supplemental, operable louvers	$0.36	10.3	8.8
Multifamily	Gas supplemental, operable louvers	$0.34	10.9	7.2
Lodging	Gas supplemental, operable louvers	$0.24	7.7	5.0

Note that the higher utility cost corresponds to the buildings with higher loads; office and public buildings. In all cases, the optimal mechanical room configuration is to use a gas unit heater for supplemental heat, and operable louvers for control of the room.

DEFROST STRATEGY

Defrost has a significant impact on the operation and energy usage of any air source heat pump, including air source VRF units. When these units are in heating mode, below 41ºF, moisture in the air will condense and freeze on the outside of the outdoor unit coils (which is the evaporator coil in this mode). VRF systems generally utilize a reverse cycle for defrost, whereby the outdoor units are cycled to run in reverse (as if cooling were needed) to warm the outdoor coil and melt the frost.

There are two common methods for controlling this reverse cycling. The simplest method is timed defrost, in which the unit will run a defrost cycle for a set portion of each hour whenever the outside wet bulb temperature is conducive to frost buildup, with no consideration for other conditions. However, frost buildup is only problematic when there is sufficient moisture in the outside air, coupled with significant heating mode operation frequency. So, timed defrost tends to over-utilize the reverse cycle, resulting in higher than needed energy consumption. A more sophisticated approach, demand defrost, utilizes a sensor to detect frost build-up, and only operates the defrost cycle when necessary. This impacts the unit efficiency as shown in Figure 2. The resulting energy savings in using a demand defrost approach as opposed to a timed approach is $0.015/ft^2, or about 4% of VRF energy.

Note that defrost also creates waste water that can be a freezing nuisance when temperatures in the mechanical room are cold enough. This concern is discussed in the *Optimal Mechanical Room Temperature* section.

NOTES ON SIZING OF VRF UNITS FOR HEATING

Our primary focus of this analysis was in configuring and controlling outdoor units and supplemental equipment for air source VRF in a cold climate. Much of the discussion thus far focuses on the energy and energy cost impacts of those design decisions, but there are also impacts on first cost.

Basic Sizing of Outdoor Units and Supplemental Unit Heater

One significant first cost is associated with the outdoor units and supplemental devices themselves, making it worthwhile to consider how design decisions impact their capacities. For our base case, which has a peak heating load of 2,400 MBH, if a gas-fired supplemental device is used it should be approximately 1,450 MBH (both numbers assume a 25% safety factor for heating). This equates to 15 Btu/hr/ft^2 for our base building. Note that the electrical power to the compressors actually turns into heat which makes up the difference between unit heater size and heating load.

As for the outdoor units, in this same base case (again with a peak heating load of 2,400 MBH), if we maintain the ambient air to the compressor at a near-optimal 5°F using the unit heater, the outdoor units must have a nominal capacity of 3,600 MBH in order to meet the load (due to derating at low ambient temperatures). If no supplemental unit heater is installed and the units must be able to operate at our -11°F design condition in Madison, these units must increase to 4,000 MBH nominal heating capacity. And this is if the designer chooses to just meet the design condition. Sizing results are summarized in Table 6.

Table 6: Nominal equipment sizes required to serve base case office building in Madison, WI, for two different mechanical room temperatures. Units are shown in both Btus and tons for convenience.

Mechanical Room Temp. Setpoint °F	Outdoor Unit Nominal Heating Capacity MBH	tons	Supplemental Unit Size MBH
5	3,600	300	1,450
-11	4,000	330	1,350

Block Load vs. Peak Load

Sizing of the internal VRF components: fan coils, piping, heat recovery units, etc. must all be done according to the peak load in each zone. However, as with some other plant equipment (e.g. cooling towers) the outdoor units can be sized according to peak *block* load, which is lower than the sum of all zone peaks due to diversity in use of the building. This often leads to a connectivity ratio (ratio of indoor equipment capacity to outdoor equipment capacity) of greater than 1.0. Note that based on ratings for some VRF equipment, this can lead to an increase in the efficiency – above AHRI ratings – of between 5-10%.

Comfort Concerns

Much of this section has focused on sizing of the outdoor VRF units. It is worth noting that accurate sizing of the indoor units is also important, not just for meeting of design requirements and cost, but for overall comfort. If indoor units are oversized by too large of a margin, the PID loop that maintains temperature control in the zone will not be able to control the refrigerant flow to maintain a zone temperature close enough to the setpoint; increases in refrigerant flow will too easily ‘overshoot’ the zone temperature on the low end, and the room will become colder than desired (leading to cold calls).

Sizing for Simultaneous Cooling

Finally, in sizing the system the designer needs to consider that even at very low outdoor temperatures some zones (such as core zones, server rooms, etc.) will still be in cooling mode. Therefore, at least one compressor must be available to modulate to meet the cooling load. An N+1 sizing method for heating must therefore be considered (at the *least*), and only the N considered available for heating. Our models accounted for this as described in the Methodology section.

KEY CONCLUSIONS

During our investigation of optimal configuration and control strategies for air source VRF systems in cold climates, we reached a variety of different quantitative and qualitative conclusions:

- Ideal system design is highly dependent on the shape of the heating performance curves for the VRF units. The optimal mechanical room setpoint occurs at a temperature close to, but higher than, the temperature at which the system’s heating efficiency and capacity drop off significantly.
- The configuration with the lowest utility cost and EUI for buildings in Madison, WI is a mechanical room with operable louvers and a supplemental gas unit heater. The operable louvers also result in lower supplemental heater size than the fixed louver case.
- For base conditions in Madison, the optimal mechanical room temperature setpoint is somewhere between -12ºF and 5ºF, and that this range held true until gas price dropped below $0.25/therm.
- Significant thought should go into defrost and condensate handling at sub-freezing temperatures: equipment should be built, configured, and placed such that defrost and condensate water can be collected and flow to a drain with limited exposure to the cold mechanical room along the way.
- The resulting energy savings in using a demand defrost approach as opposed to a timed approach is about 4% of VRF energy.
- Sizing of equipment is critical for VRF systems, not just due to impacts in first cost, but also because sizing heavily affects efficiency and comfort.

The energy model that we utilized for this study is planned to be made available to the public for analyzing different, custom scenarios. Check at the project’s webpage, www.ecw.org/cold-climate-vrf to determine availability.

FUTURE WORK

Though this study attempted to be as thorough as possible, there are several areas of investigation that remain for cold climate VRF:

- Most importantly, field research is needed to validate how these systems perform in actual buildings. A project that measures energy usage and other performance parameters from several air source systems would be best. Other studies have shown that actual field performance of such complex systems is often different – worse in some cases, better in others – than the theory or even lab ratings suggest. Field work would also result in a significant body of lessons learned from those operating these systems every day.
- The dual temperature control scheme described above should be investigated further.

- Other sources of supplemental heat could be investigated. One suggested approach has been a hybrid system that could allow the outdoor unit to utilize a water-source heat exchanger below a certain temperature setpoint, and cease absorbing energy from cold outdoor air.
- Full life cycle cost economics could be calculated, especially with consideration for the cost of maintenance, and of building the mechanical room itself.