APPLICATION AND DESIGN OF AIR-TO-AIR VARIABLE REFRIGERANT FLOW SYSTEMS

by

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Abstract

This report addresses the operation, use, and design of air-to-air variable refrigerant flow systems, also known as VRF. Relatively new to the United States, these HVAC systems have potential to reduce energy consumption and utility costs in the correct applications. Although useful in many applications, the best building types for VRF are those requiring a large number of zones and with low ventilation air requirements. The report explains design and system selection considerations and accordingly presents two flowcharts to help designers implement this system. To show how the system compares to traditional technologies in terms of efficiency and cost, the report presents results from several studies comparing VRF to other systems. In addition, an energy modeling study is conducted to clarify the effect of climate on the system; this study established air-to-air VRF as having highest energy consumption in dry, southern climates, based on energy use and operating costs. With this report, HVAC designers can learn when air-to-air VRF is an acceptable method for providing heating and cooling in a building.

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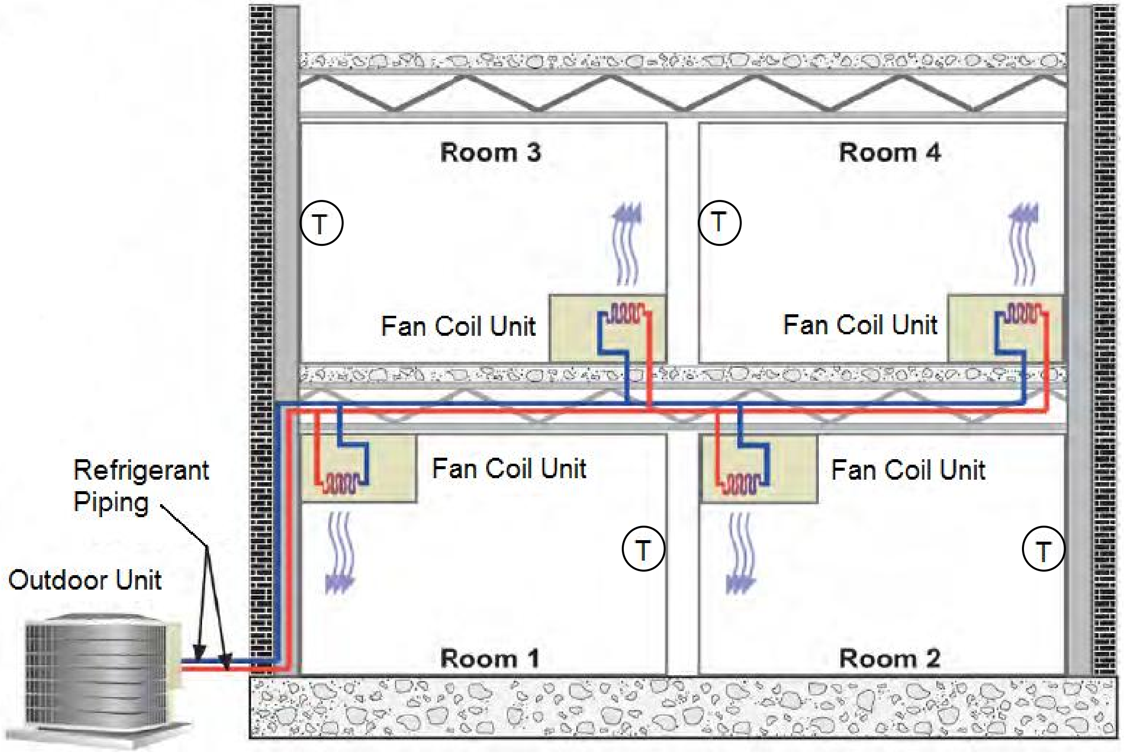
Chapter 1 - Introduction

As of 2008, the commercial building sector was responsible for 19% of energy consumption in the United States; of which 32% is dedicated to space heating, space cooling, and ventilation loads (Center for Sustainable Systems, 2009). As reducing energy consumption becomes a higher priority, heating, ventilation, and air conditioning (HVAC) designers in the United States' building and construction industry are specifying systems in which sustainable design lowers energy use in commercial buildings. Variable refrigerant flow systems (VRF) are HVAC systems that have great potential to reduce energy consumption when applied correctly. This research addresses the most common type of this technology, air-to-air VRF. Water-to-air VRF systems exist, but little published research exists on them, making accurate discussion about the use and efficiencies beyond the scope of this report. However, research has shown that these systems can expand the use of VRF into areas where air-to-air systems are generally less efficient than water-to-air systems, particularly dry, southern climates. This report defines air-to-air VRF, describes how it works, details how to properly design and select the system, compares the system to more traditional systems, and investigates the effect of climate on the overall performance and operating costs of the system.

1.1 What is Variable Refrigerant Flow

The term VRF stems from ability of the system to modulate the amount of refrigerant flowing to each indoor unit, called a fan coil unit. The system runs constantly at loads between 10% and 100%, but cycles on and off only at loads below 10% (Nye, 2002). To provide heating and cooling, this HVAC system utilizes refrigerant piped via one set of pipes from a single outdoor condensing unit, referred to simply as an outdoor unit, to terminal fan coil units indoors; these will be referred to as fan coil units throughout this report and should not to be confused with traditional fan coil units commonly used in hydronic systems. Instead of using hot water for heating and chilled water for cooling, these fan coils utilize refrigerant delivered via a single pipe for either heating or cooling; in some climates, electric backup heat is required in addition to the basic VRF system. Regardless, each VRF fan coil unit receives only the volume of refrigerant needed to condition the space so that each unit meets the specific needs of each zone when individual temperature controls are provided (Goetzler, 2007). One outdoor unit can accommodate up to 60 indoor units and loads up to 25 tons, rendering the system useful in almost any size structure and extremely versatile in the commercial building industry (Amarnath & Blatt, 2008). Figure 1.1 illustrates how these fan coil units are connected to the outdoor unit.

Figure 1.1 VRF Systems. (Modified with permission by Ammi Amarnath and the Electric Power Research Institute)

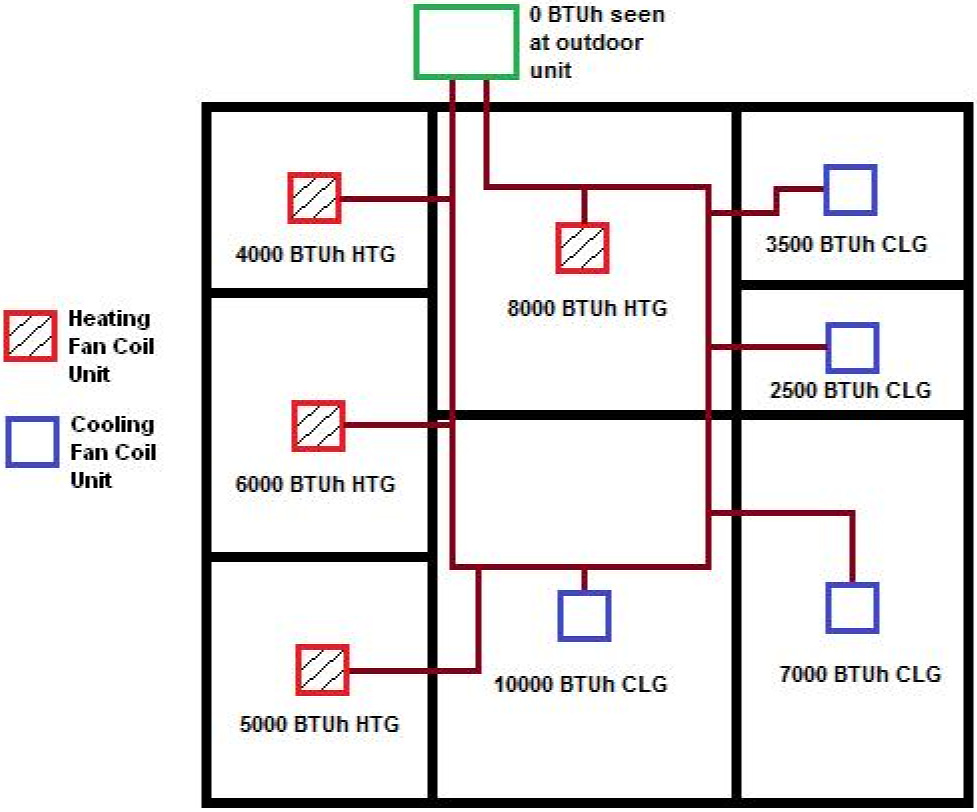


This system can be utilized in a wide variety of commercial buildings, including offices, hotels, historical renovations, additions, hospitals, and schools (Roth, Westphalen, Dickmann, Hamilton, & Goetzler, 2002). In the past, such buildings have been conditioned primarily by conventional HVAC systems, including hydronic systems, variable air volume systems, conventional direct expansion split systems, and heat pump systems. However, designed and implemented properly, VRF can match the zoning and flexibility of these systems much more efficiently and comfortably, making VRF a viable option for an HVAC system.

Two types of VRF systems exist: heat pump systems and heat recovery systems. Heat pump systems require a refrigerant liquid line and suction line be connected to the outdoor unit. This type of system provides either all heating or all cooling because it cannot perform both functions simultaneously, limiting this type of systems implementaion (Amarnath & Blatt, 2008). With heat pump VRF systems, buildings that have zones requiring cooling during the heating season will need two separate heat pump VRF systems; one system can serve the zones requiring year-round cooling, and one can serve the remainder of the zones. However, the non-year-round cooling zones will still have variable cooling and heating loads; if some of these zones require cooling while others require heating, the heat pump system can only provide conditioning to match the load requirements of the majority of the zones.

In contrast, heat recovery VRF systems allow simultaneous heating and cooling operation. These systems have a secondary advantage in that heat is transferred between zones, maximizing system efficiency. Specifically, refrigerant is circulated between zones in either a two- or three-pipe system configuration (Goetzler, 2007). This can greatly reduce the use of the central outdoor unit when the system is heating and cooling simultaneously in different zones and can transfer heat between the zones. Moreover, self-balancing takes place when the demands for heating and cooling are equal. In three pipe systems, heating units absorb heat from a central refrigerant line while cooling units reject heat to the central line; meanwhile, two pipe systems utilize a device called a branch controller, or BC controller, to transfer heat between zones. Figure 1.2 illustrates how the heating and cooling loads of different fan coil units can yield a net load of 0 BTUh in either two- or three-pipe systems. In this case, the compressor will only operate to circulate refrigerant throughout the piping loop, and when one load is greater than the other, the compressor must make up the difference in demand (Inman, 2007).

Figure 1.2 Self-balancing Heat Recovery System.



1.2 Reasons for Slow Adoption in the United States

VRF has been designed and installed for several decades in Europe and Japan but has seen limited use in the United States. This is not because of any particular problem with the system, but rather a result of several external factors: designer and client unfamiliarity with the system, the different construction process, a lack of regulations encouraging the use of alternative systems, limited publicity and advertising, and historically low energy costs.

The first external affecting the use of VRF is unfamiliarity with the system. Most designers in the United States are familiar with ducted direct expansion (DX) systems and hydronic systems and are confident these systems will function properly in specified applications. However, lack of knowledge and experience with VRF has influenced designers to be hesitant to recommend installations. Because VRF are strongly marketed for applications commonly served by traditional hydronic systems, designers typically have not chosen VRF systems over these more familiar, conventional systems. In addition, building owners are not likely to request a VRF system to be considered for their properties because they have not heard of the system and its energy savings potential (Goetzler, 2007). Unfamiliarity by designers and building owners is echoed in the construction sector as well.

The slightly different construction process required by VRF is the second factor hampering the use of this system. Installing a VRF system is different from installing other technologies, and each VRF manufacturer has specific requirements with which a contractor must be familiar. For example, one manufacturer may specify a maximum refrigerant piping length of 100 feet while another might have an allowable length of 175 feet; exceeding these lengths can reduce the ability of the system to meet the space load. Another difference arises as a result of manufacturers developing their own VRF system layouts; pipe fittings and unit configurations can differ among manufacturers. These variations can make the competitive bidding process complicated, since finding comparable products for substitution is difficult. Because of variable construction requirements, contractors first must invest time to learn the specific requirements of a manufacturer's system. Also, most contractors have limited installation experience with VRF and as a result and will charge more for installation, making owners less likely to invest in the system (Goetzler, 2007).

The third factor contributing to the slow adoption of VRF technology is lack of regulations that would encourage its use. In Japan, where the technology was developed, system use was increased by Japanese regulations. To alleviate strain on the electric grid, Japanese authorities prohibit electric chiller installations in large commercial buildings, which consume a significant amount of electricity for cooling (Ryan, 2008). Because of this legislation, designers were forced to seek other options, and as a result, VRF was installed in more projects. If similar regulations were implemented in the United States, American designers would likewise have to explore alternative sources of heating and cooling. As the focus on reducing energy consumption in the United States intensifies and as the popularity of the energy rating system LEED grows, VRF will likely be considered more often, particularly in areas prone to brownouts and blackouts.

Limited publicity and advertising has also contributed to VRF systems not being widely used in the United States. VRF was first introduced by companies based in Asia, which had limited representation in United States markets until recently. This rendered advertising and exposure to the product minimal (Goetzler, 2007). Clearly, designers willing to explore alternative technologies will lead the industry toward VRF systems. Also, as these foreign companies and their products become more widely-known by U.S. designers and building owners, the demand for their VRF systems should increase.

Finally, historically low energy costs have significantly slowed adoption of VRF into the United States. When VRF was introduced into the country during the 1980s, the United States had very low energy costs (Amarnath & Blath, 2008), and low operating expenses associated with running an HVAC system was not the priority it is today. Fortunately, recent increases in energy rates and a push to minimize energy dependence have made systems that are energy efficient and low-cost to run and maintain more desirable. This change in mindset will likely allow VRF to gain in popularity.

1.3 Advantages

VRF systems have several characteristics that make the technology promising: potential for submetering, efficiency, comfort, design benefits, ease of installation. Additionally, some aspects of maintenance also favor this type of system. Each of these benefits will be discussed further in the following paragraphs.

One attractive aspect of the system is the potential for submetering within the system. The cost of energy consumed by the outdoor unit must still be divided between tenants; but each VRF indoor fan coil unit could be submetered, allowing building owners to bill occupants individually for their electric consumption (Goetzler, 2007). This cannot be done with systems utilizing a central unit serving different tenant spaces; for example, a central air-handling unit with variable-air-volume reheat could serve multiple spaces but could not be submetered. From a building owner standpoint, submetering allows more accurate billing capability for buildings leased to multiple occupants. In addition, this could encourage patrons in multi-tenant buildings to reduce their energy use since they would have to pay directly for consumption. This factor, in addition to the high efficiency of the system, can serve to curtail energy waste.

Next, for several reasons, VRF technology is highly efficient. The Department of Energy has listed it as one of the top 15 technologies with potential for a significant impact on energy savings (Dickens, 2003). In particular, VRF has higher part load and seasonal energy efficiencies, because the outdoor unit compressor motor speed can be modulated. In buildings with zones requiring simultaneous cooling and heating demand, heat recovery VRF systems can transfer heat from one space to another, further reducing energy consumption. In addition, VRF eradicates or minimizes air leakage and temperature changes through ductwork since little or no duct work is necessary depending on the fan coil unit configuration selected. Non-ducted units completely eliminate all duct losses; this is significant because duct losses can range from 10% to 20% of the airflow in a ducted system (Goetzler, 2007). Also, the VRF system can be used in the United States Green Building Council, or USGBC, LEED building certification system. Although LEED is a rating system often requested by owners and not necessarily a direct measure of energy efficiency, HVAC designs that meet the requirements of this certification are likely candidates to reduce energy consumption.

Third, comfort levels are very good with VRF technology. The indoor fan coil units can each have individual control, allowing for zoning and individual selection of temperatures for each zone (Amarnath, 2008). Also, VRF systems reduce temperature drift in the space by keeping the compressor and indoor unit fans running constantly during most part load situations instead of cycling on and off, resulting in temperature drifts as low as ±1°F. For a disabled fan coil unit in a space with multiple units, that room can likely be comfortably occupied while repairs are being made; meanwhile, load normally handled by the disabled fan coil can be handled by the operational fan coil units. Moreover, if failure occurs in the sole unit conditioning a room, only that room will be negatively affected. Finally, if one of the two compressors in the outdoor unit fails, the other will be available to provide some cooling until the other is repaired (Goetzler, 2007).

In addition to the high degree of thermal comfort, the system also has good acoustic properties, with noise as low as 24 dBA for fan coil units and 56 dBA for outdoor units (Siddens, 2007). However, this is the low end for the system; most indoor units operate around 30 dBA and most outdoor units operate around 60-65 dBA. These ranges are competitive with those of other systems; most diffusers are selected with a maximum noise criterion of 30 dBA, and outdoor equipment can range from as low as 50 to over 75 dBA. The noise level of VRF, combined with the constant operation of the indoor fan, helps create an environment where occupants will not be disturbed by mechanical equipment. Noise can be further minimized by locating ducted fan coils outside the occupied space such as above the corridor or in an adjacent storage space.

Finally, indoor air quality is increased by minimizing ductwork compared to split systems and even some hydronic fan coil layouts; many indoor units supply air directly to the space. Thus, mold, fungi, and bacteria have fewer places to colonize within the VRF system (Nye, 2002). This can provide significant advantages for people with chronic health issues such as allergies and asthma. All of the factors combine to maximize occupant comfort.

Implementation of a VRF system can offer general design benefits. Because the system requires minimal airflow through ductwork, duct sizes are minimized. In turn, this could reduce the height of each floor of a building or minimize the coordination needed to avoid potential conflicts between the structure and other equipment located in the plenum. With each floor requiring less height, a structure can conceivably hold more floors, or the total building height can be reduced, resulting in construction cost savings and less volume of space to heat and cool. Furthermore, the fan coil units of a VRF system do not need to be installed in a separate mechanical room (Goetzler, 2007). This creates more useable floor area for an owner (Cendon, 2009). VRF systems are also expandable; unfinished spaces can be fitted later with fan coil units without significant changes to the system, and space use changes can be accommodated with minimal rework to the system (Goetzler, 2007).

Installation of the VRF system can be straightforward once contractors have familiarized themselves with the installation of the specified VRF equipment. Because the system is all electric, no gas connections are needed (Amarnath, 2008). These systems require limited installation of ductwork for ventilation air and for any ducted fan coil units. Furthermore, the system components are all lightweight and relatively small, meaning lifting equipment is not necessary for moving the components into their final locations.

Finally, maintenance can be an advantage for VRF because the units are small and relatively lightweight. Should an indoor unit fail, it can be removed and replaced without major impact to other parts of the building interior; interior walls do not need to be removed, for example. If units are ducted from unoccupied spaces, maintenance is possible without interrupting building occupants; however, such units should be installed above lay-in ceilings or above an access panel in hard ceilings. Naturally, direct-to-space units are inherently accessible because of their locations within the space. Additionally, some studies indicate that VRF systems have lower overall maintenance costs than hydronic systems in similar applications (Goetzler, 2007); however, these studies could be comparing new VRF systems to old hydronic systems.

1.4 Disadvantages

Although the VRF system has many significant advantages, the disadvantages must also be considered to determine if the technology is appropriate for a specific application. Potential obstacles include refrigerant charge, all-electric heating operation, initial cost, and outdoor air limitations. Additionally, some view maintenance for VRF as a drawback. These potential inadequacies can inhibit the use of this system in certain types of buildings.

The large amount of refrigerant required is perhaps the biggest concern related about VRF. Many designers are hesitant to install systems that route such large amounts of refrigerant within the occupied space. If a leak occurs, the entire system refrigerant charge can escape into the space where the leak is located. A VRF system can contain a significant amount of refrigerant; such high volumes can quickly compromise occupant health. While this does illustrate the need to develop safer refrigerants, VRF can still be safely installed according to ASHRAE 15 - Safety Standard for Refrigeration Systems. As discussed further in Chapter 6 Section 3, designers can achieve compliance by routing refrigerant piping to avoid small rooms, installing mechanical ventilation in rooms with potential for unsafe levels of refrigerant or a refrigerant level monitoring system, utilizing ducted units, and making architectural changes to increase room volumes.

Another drawback to VRF is its all electric heating operation. In northern climates, gas heating is a more cost-effective means to condition than electric resistance (Goetzler, 2007). This is because natural gas is a less expensive utility than electricity in these regions. Nonetheless, VRF can save on electricity costs and consumption in northern climates when used to replace older, inefficient equipment. In a study by Minnesota Power, a VRF system installed in a building located in Minnesota is projected to reduce the electrical demand of the facility by up to 98,483 KWh per year; the facility had been previously conditioned by a steam boiler, several chillers, split systems, and window mounted units (Variable refrigerant flow zoning research profile, 2009). Such significant savings will not be possible with all buildings; however, the system should be considered to determine if the operating costs are competitive with those of gas-heating based systems. In climates where gas heating predominates, a life cycle cost analysis should be performed to determine if a payback exists. Any disadvantage could be offset because the all-electric technology of the system allows on-site generated energy from photovoltaics, wind, or other means to power the system, further reducing energy costs.

The electric operation of the system eliminates any need for a flue for proper gas combustion; also, some utility companies offer rate discounts when the building operates solely on electricity.

The initial cost of the VRF system can be a hindrance to selecting the system for a building. In new construction, VRF system installed costs can be anywhere from 5-20% higher than for conventional systems (Roth, et. al., 2002). This cost difference will likely decrease as contractors become more familiar with the product since they charge higher rates to install a system with which they are not familiar. Also, VRF may be more difficult to justify from a cost standpoint when renovating an existing HVAC system. For example, to replace a traditional hydronic fan coil system in an existing application, all terminal units and piping would require replacement (Goetzler, 2007), but if those components of this system were in good condition, the cost of replacing them would result in unnecessary expense. For such reasons, VRF is much more likely to be installed in new construction.

Next, VRF systems have limited capacity to heat or cool ventilation air (Afify, 2008). This drawback can be addressed by installing a designated outdoor air system, or DOAS (Goetzler, 2007). Because this may reduce the energy efficiency of the system, a designer must carefully consider whether a DOAS is really a viable option. Because of this limitation, VRF may not be useful in high ventilation applications.

Maintenance for VRF systems has two main drawbacks, including shorter equipment life than some conventional system components and multiple pieces of mechanical equipment. The first, short equipment life span, requires that VRF outdoor units be replaced more often than chillers and boilers in hydronic HVAC systems, which can last up to 30 years (Goetzler, 2007). DX systems are inherently short lived, lasting 10-15 years (Roth, et. al., 2002). However, the cost of replacing a large chiller or boiler may be so high that this drawback is negated; also, access for unit replacement is better for VRF systems. Because filters must be replaced at a room level, and maintenance must be performed on each indoor VRF fan coil unit, variable air volume (VAV) and central air-handling unit systems do have an advantage as far as ease of maintenance is concerned. Although these are noteworthy drawbacks, any HVAC system will have maintenance issues that could be equally costly or problematic. Table 1.1 provides a simplified view of VRF advantages and disadvantages.

Table 1.1 VRF Advantages and Disadvantages.

VRF Advantages & Disadvantages	
ADVANTAGES	DISADVANTAGES
Submetering potential	High refrigerant charge
High efficiency	All-electric technology
Good comfort levels	Low ventilation air capacity
Minimal ductwork	High initial cost
Easily expandable	Equipment life-span
Ease of installation	Maintenance at each fan coil
Equipment easily replaced	
Accessibility of fan coils	
Maintenance costs	

1.5 Useful VRF Applications

VRF is useful for a variety of building occupancy types, particularly those with multiple zones. The flexibility and potential for low plenum space required by the system make it a viable choice for additions and historical renovations. Since it is best suited for applications with lower ventilation requirements, VRF is most practical in offices, hotels, and condominiums. Although hospitals and schools require large amounts of outdoor air, the system can still be used if the ventilation air is handled by a separate system (Goetzler, 2007). In fact, VRF provides an excellent method to prevent the mixing of air between zones, becoming a good candidate in spaces where this is a major concern, for instance in healthcare design; however, care must be taken to avoid routing refrigerant piping over patient rooms as Chapter 3.3 discusses (Cendon, 2009). Also, the designer must consider if the owner is going to be occupying the space; building owners who plan to sell or rent out the space may not be interested in the high up-front cost of the system.

VRF should not be considered in applications that are single zone or only have a few zones. For example “big box retail” applications or large warehouses would be best served by traditional DX rooftop units (Goetzler, 2007). The large open space and few zones associated with this type of building make the expense of VRF unjustifiable. Heat recovery systems are particularly useless in this application because no opportunity for self balancing exists.

VRF systems are better suited to some climates than others, especially climates with modest heating loads (Roth, et. al., 2002). In extremely cold climates, the heating capacity of the system is reduced, creating a need for electric resistance heating. Next, limitations on wet and dry bulb temperatures can cause trouble in hot humid climates by exceeding the capacity of the fan coils (Afify, 2008). Specifically, per the anaylsis performed in Chapter 6, areas with high cooling design temperatures and low wet bulb temperatures are less favorable for VRF application than other systems; in short, the system will use more energy than in other climates. These problems can be mitigated with energy recovery ventilators; however, energy use can still increase for VRF in such climates as the analysis in Chapter 6 demonstrates. the energy use of VRF in such climates. If guidance is needed as to what areas of the country are prone to these climatic conditions, Appendix B of ASHARE 90.1 can be referenced.

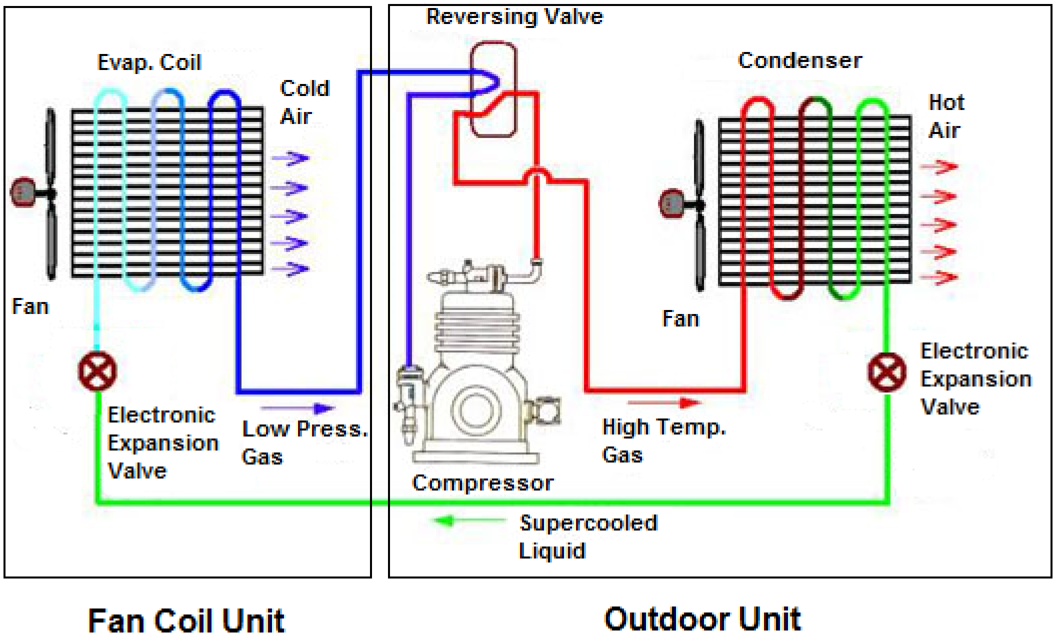
Chapter 2 - Operation

The VRF system consists of several components combined to heat or cool a conditioned space. For every system, at least one outdoor unit containing compressors, fans, condensers, and several valves must serve multiple indoor fan coil units; in some climates, electric heating coils must also be installed. The fan coil units contain a fan, a heat exchanger, and an expansion valve. Refrigerant piping connects the outdoor unit to the fan coil units, allowing refrigerant to transfer heat between units for space conditioning. With this system, heat can be transferred from indoors to outdoors or outdoors to indoors, similarly to how air-to-air heat pumps work. Because of the heat recovery capability, heat can also be transferred from one space to another as further explained in Section 2.1.2, Indoor Components.

The basic refrigeration cycle cools the space, and the heat pump cycle (the reverse refrigeration cycle), is used for heats it. In any system using refrigerant, the compression and expansion of the refrigerant is the basis of space conditioning; refrigerant must be allowed to expand to cool or must be compressed to heat, all while indoor and outdoor devices in the system serve a critical role in either cycle.

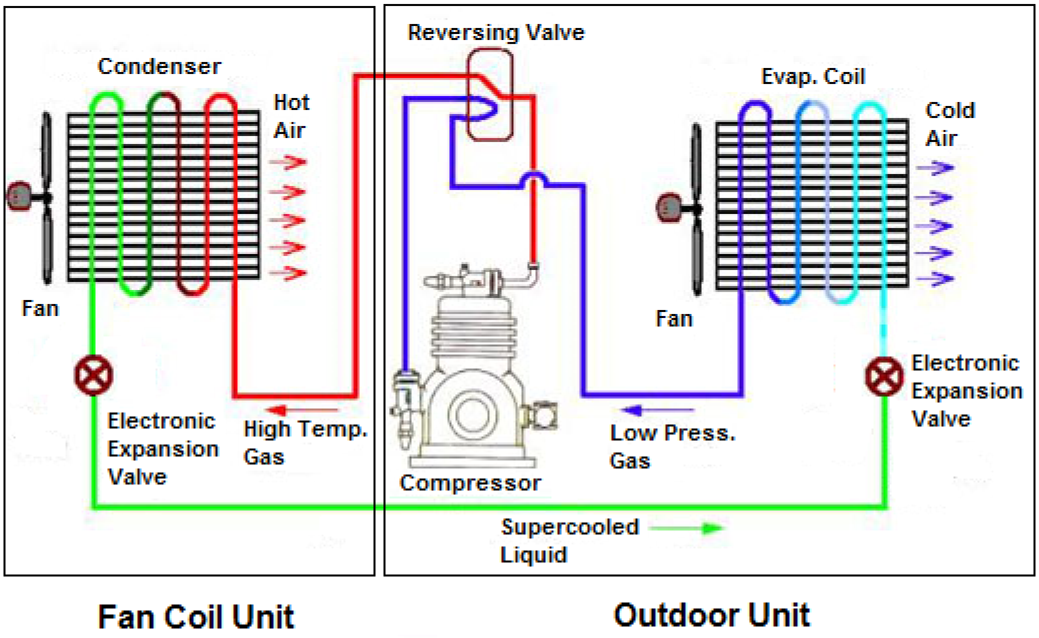
As illustrated in Figure 2.1, the VRF refrigeration cycle process begins with the compressor located in the outdoor unit converting low pressure refrigerant vapor into a high temperature refrigerant gas and delivering it at the vapor saturation point to the condenser in the outdoor unit. The condenser converts this refrigerant gas into liquid refrigerant by rejecting heat to the surrounding atmosphere with the aid of a fan; an electronic expansion valve allows the liquid refrigerant to expand, causing the refrigerant to reach a super cooled stage. The super cooled liquid is delivered from the outdoor unit to each indoor fan coil unit, where it passes through an evaporator coil and absorbs heat from the airstream. The refrigerant then becomes a low pressure gas and returns to the compressor in its initial low-pressure, gaseous state. Notably, this graphic shows only a single fan, condenser, and compressor in the outdoor unit for the sake of clarity; a unit actually contains two of these in parallel, as Section 2.1.1 explains.

Figure 2.1 The Refrigeration Cycle. (Modified with permission by the Singapore National Environment Agency).



In heating mode, the refrigeration cycle is reversed; the flow of refrigerant is reversed by a reversing valve within the outdoor unit, causing the outdoor condenser to become an evaporator and the evaporator within the fan coil unit to become a condenser. The refrigerant is delivered to each fan coil as hot gas; the coil in the unit acts as the condenser by rejecting heat to the conditioned space and causing the refrigerant phase to change from gas to liquid. The liquid refrigerant passes through the outdoor condensing unit, which acts as an evaporator by absorbing heat from the atmosphere. The refrigerant is then sent through the compressor and converted to a hot, high pressure gas. Figure 2.2 illustrates this process. As discussed in Section 2.1.1, Outdoor Components shows, the outdoor unit has two fans, two condensers, and two compressors, but these have not been shown for the sake of clarity.

Figure 2.2 The Heat Pump Cycle. (Modified with permission from the Singapore National Environment Agency).



2.1 Major Equipment and Components

The air-to-air variable refrigerant flow system needs several different components. Also, fan coil units must be selected for every thermal zone being conditioned, while an outdoor unit must be selected to handle the load of the multiple indoor fan coil units (Amarnath & Blatt, 2008). Thus, connected by one set of refrigerant pipes, both the outdoor unit and fan coil units are composed of components essential for proper heating and cooling.

2.1.1 Outdoor Components

An air-to-air VRF system consists of an outdoor unit serving up to 60 indoor fan coil units via one set of refrigerant pipes (Amarnath & Blatt, 2008). For heat pump systems, each outdoor unit can have a heating and cooling capacity anywhere between 12,000 and 300,000 BTUh. Heat recovery systems have a larger minimum capacity; 72,000 BTUh is currently the smallest unit available (Variable Refrigerant Flow, n.d.). These outdoor units contain two compressors, two condensers, two fans, and an accumulator (Xia, Winandy, Georges, & Lebrun, 2004). The outdoor unit is comparable to a traditional air cooled condensing unit, or ACCU, used in typical direct expansion cooling systems such as split systems. However, the big difference is that the outdoor VRF unit can act as an air-to-air heat pump by drawing heat from the atmosphere during heating mode because of the integral reversing valve.

The actual appearance and size of an outdoor unit varies depending on the manufacturer, the expected load, and whether it is serving a heat pump or a heat recovery VRF system. Figure 2.3 below provides an example of this variation. The manufacturer, model, type of system, unit capacity, and unit dimensions are given for each unit.

Figure 2.3 Outdoor Units. (Images printed courtesy of Daikin AC and Mitsubishi Electric).

Two hermetic scroll compressors are contained in the outdoor unit to compress the refrigerant, turning it into a high temperature gas; they both operate using the processes illustrated in Figures 2.1 and 2.2. One is variable speed and one is constant speed; each is sized for half the total capacity of the outdoor unit. The variable speed compressor is used most of the time and its motor modulates anywhere from the full capacity of the compressor to a minimum speed of 10% (Nye, 2002). If the load becomes higher than what the variable speed compressor can handle alone, the constant speed compressor turns on, and the frequency of the variable speed compressor is reduced to handle any excess load above 50%. Once loads are reduced to below 50%, the constant speed shuts off, and the variable speed compressor operates exclusively. If the load requires the variable speed compressor to operate below its minimum frequency, the compressor will shut off and cycle on if needed to avoid the inefficiency related to hot-gas bypass (R. Froebe, personal communication, September 16, 2010).

The variable speed compressor is the key to energy savings with the VRF system in that it utilizes an electronically commutated motor (ECM), also known as a brushless DC motor, and a variable speed inverter drive to gain an edge in efficiency over conventional compressors. The ECM, fitted with a magnetic rotor to prevent loss of efficiency at low speeds, boosts the overall efficiency of the condenser dramatically; some estimate that the motors are up to 30% more efficient than traditional induction motors (Amarnath, 2008). Meanwhile, the variable speed inverter drive is responsible for controlling the speed of the compressor ECM by allowing the motor to operate at a high speed initially to reach the conditioned space temperature set points; once these set points are reached, the variable speed drive will reduce the rpm of the ECM, keeping it running continuously to maintain the desired temperatures in served zones (Amarnath, 2008). These two technologies combine to create a variable speed compressor motor system that is generally 82-90% efficient; however, below 30% loading, this compressor system efficiency starts to drop dramatically (Amarnath, 2008). This is why the motor turns on and off completely below a predefined minimum capacity; occupant comfort will not be compromised because the system will still operate despite not running constantly.

Most manufacturers use two parallel condensers in the outdoor unit to handle the load of the building, one for each compressor, and each condenser is coupled with one variable speed fan. These condensers are really heat exchangers capable of serving also as evaporators. They serve as condensers when rejecting heat to the air during cooling mode and as evaporators when absorbing heat from the atmosphere during heating mode, depending on the direction of refrigerant flow. The two variable speed fans facilitate this heat transfer by forcing air to pass over each condenser; having a variable speed motor on these fans allows power consumption to be reduced at part load (Xia et. al., 2004).

An accumulator is needed in the outdoor unit because compressors can only handle vapor refrigerant; in fact, this device is needed for any DX system. Not shown in Figures 2.1 and 2.2 , the accumulator is located between the compressor and the reversing valve and captures liquid refrigerant and oil, preventing it from entering and damaging the compressor. The accumulator contains a metering device that converts the liquid refrigerant into a vapor and sends this refrigerant and oil to the compressor (The Role of the Suction Line Accumulator, 2001).

The outdoor unit needs several valves as well, including a reversing valve and expansion valves. A reversing valve is needed to control the direction of refrigerant flow for heating or cooling mode; this is controlled by the HVAC control system. The controls system will switch the direction of the valve depending on the load borne by the fan coil units. In heat pump systems, a majority of zones calling for cooling will send a signal to the valve, causing it to facilitate the refrigerant cycle; the opposite is true when a majority of zones call for heating. In heat pump systems, the controls system will reverse the direction of the valve depending on the total net load on the building. Also, the system has two expansion valves (Xia et. al., 2004); these are located downstream of each condenser to allow refrigerant to expand and reach a super-cooled temperature after passing through either condenser.

2.1.2 Indoor Components

After the refrigerant has passed through the outdoor unit, it goes into the building to provide conditioning. With the VRF system, several components must be located indoors including fan coil units, expansion valves, piping, and controls. These are all components essential for the efficient, local control associated with this system.

VRF fan coil units provide air distribution in each space being conditioned. Each unit can be paired with a thermostat to create a zone, or multiple units may be grouped together with one thermostat either to create fewer zones or to handle zones with loads exceeding the capacity of a single fan coil unit. Each unit is capable of providing anywhere from 5,000 to 120,000 BTUh of cooling or heating (Variable Refrigerant Flow, n.d.). The units can be in the space to directly supply conditioned air or can be ducted to provide air to rooms where a visible unit is not desired. Examples of both ducted and non-ducted units are in Figure 2.4. Direct-to-space units can be cassette ceiling mounted, ceiling suspended, wall mounted, floor consoles, or concealed floor consoles; however, with noise levels around 30 dBA, these could be excessively noisy in some applications. Designers should reference manufacturer data for exact noise levels of direct-to-space units. Ducted units, which would reduce the amount of mechanical noise in a space, can be horizontal or vertical units.

Figure 2.4 Indoor Units. (Images printed courtesy of Daikin AC).

The styles of units seen in Figure 2.4 are not the only styles available from manufacturers. Many variations in appearance exist; one manufacturer even offers a wall mounted unit that is disguised as artwork. In addition to different aesthetic options, many manufacturers offer both low profile units for shallow plenum depth applications and standard size units for situations with more generous plenum space. Capacities vary based on the style of unit selected; therefore, designers should reference manufacturer performance data to select fan coils.

Direct-to-space units are generally at or below the finished ceiling height; therefore, many manufacturers specify integral condensate drain pumps to avoid any drainage problems. Condensate will occur during cooling mode when the temperature of the coil is lower than the dew point of the air, with the amount dependent on the latent load treated by the coil. Accordingly, condensate piping must be designed and sized for the specific conditions of the building.

Electronic expansion valves are essential to the VRF system to modulate delivery of the correct volume of refrigerant to each unit (Xia et. al., 2004). The valve is usually installed by the manufacturer inside each indoor fan coil unit. However, some manufacturers locate the valves outdoors in a distribution box (Obella, 2009). An outdoor distribution box means the system requires more piping and a separate piping circuit for each indoor unit; this is not a common configuration. In both cooling and heating mode, the expansion valves modulate to deliver refrigerant at a manufacturer-specified temperature and pressure; to maximize efficiency. If an indoor unit is turned off, its expansion valve will remain slightly open to prevent refrigerant from becoming trapped in front of the unit. Because the expansion valves are located at each fan coil, balancing problems such as pressure drop and short circuiting are eliminated. As with any DX system, VRF must use oil return, the process by which oil used for internal lubrication of the compressor is returned to the compressor after escaping into refrigerant lines. During this process, the expansion valves open, and the compressor runs at high speeds set by the manufacturer; therefore, any oil resting in refrigerant lines is swept back to the compressor (Variable Refrigerant Flow, n.d.).

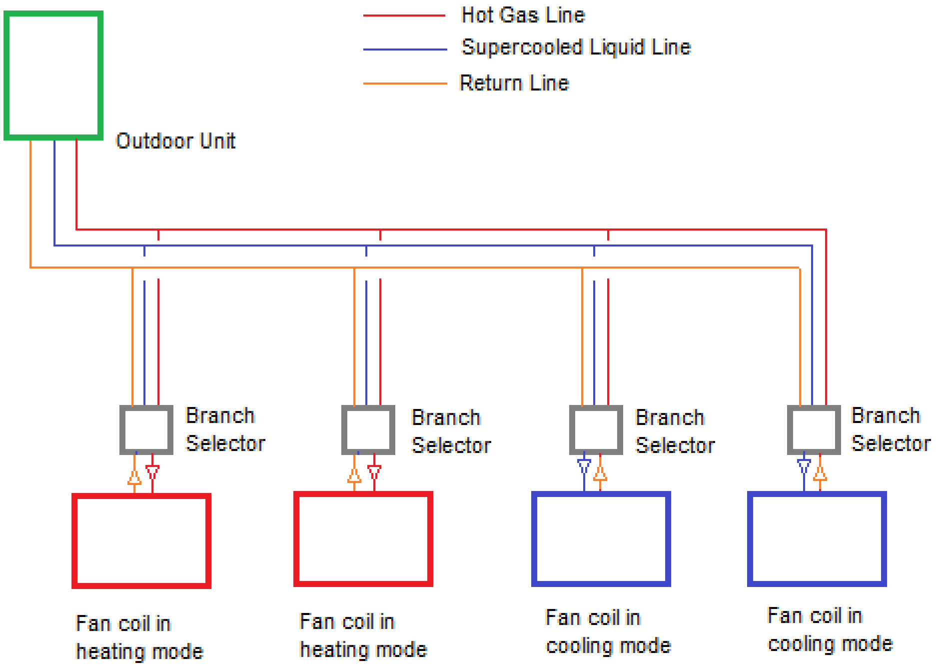
Refrigerant piping connects the outdoor unit and each indoor fan coil unit in the VRF system. Because the expansion valves will balance the system properly, the quantity of pipe can be minimized because the system can be laid out as a direct return system. As with any refrigerant piping, VRF piping must be insulated to prevent condensation from forming on the pipe surface and to minimize heat gain or loss from the refrigerant as it travels between the fan coil unit and the outdoor unit. Although heat pump and heat recovery piping systems differ slightly, one set of pipes connects the outdoor unit to the indoor units in both cases; however, the number of pipes in each set of pipes can vary.

For heat pump systems, the refrigerant can be delivered to the indoor units as either a cool liquid for cooling or a hot gas for heating, limiting the system to providing either heating or cooling. Alternatively, heat recovery VRF outdoor units, which allow for simultaneous heating and cooling, deliver refrigerant to the indoor units as a mixture of gas and liquid. The manufacturer determines the means by which this multi-phase mixture is delivered to the indoor units.

In a heat pump VRF system, the outdoor unit is connected to the indoor units by two pipes, a liquid line and a suction line. The liquid line acts as the refrigerant supply line. When selected with a small diameter, the pressure drop in this line will increase; conversely, pressure drop will decrease with a larger diameter. Because of the expansion valves located at each fan coil unit, pressure drop from the piping diameter off this line is not a major concern. Meanwhile, the second pipe, or the suction pipe, serves as the return piping from the fan coils to the outdoor unit. Here, pressure drop can have significant impact, as excessive drop can cause a 20% loss of cooling capacity in the compressor; to maintain a high coefficient of performance (COP) for the compressor, manufacturers must also try to keep the pressure constant. In addition, the diameter of the suction piping must be sized correctly or oil return will be compromised (Variable Refrigerant Flow, n.d.). Correct sizing can be accomplished with manufacturer design programs, which will provide pipe sizes after the system layout has been input.

For heat recovery systems, most manufacturers offer a three pipe system consisting of a liquid line, a suction line, and a discharge line; however, a two pipe system is available from one manufacturer. In three pipe systems, all three pipes are connected to the outdoor unit. The discharge line supplies hot gas, and the liquid line supplies liquid refrigerant to a device often referred to as a branch selector; a suction line is also connected to this device. A solenoid valve on each line and a heat exchanger are located at the branch selector. Each fan coil unit has a specified branch selector; the heat exchangers within the selector allow heat to be transferred from the return line to the hot gas line. When a zone requires cooling, the solenoid valves on the liquid line and the suction line open. When heating is required, the hot gas and suction line valves open (Goetzler, 2007). The refrigerant flows through the suction line back to the outdoor unit, which works to make up any difference between the net heating and cooling loads; when the net load is zero, the compressor will run at the minimum speed to keep refrigerant moving through the system. Figure 2.5 illustrates how this system is laid out. Although this three pipe system may seem inefficient on first inspection, this is what allows heating and cooling loads to cancel out, reducing the energy consumption of the outdoor unit.

Figure 2.5 Three-pipe Heat Recovery System.



For the two pipe heat recovery system, gas and liquid are mixed and travel over the same supply line; a branch controller, or BC controller, separates this gas-liquid mixture and sends the necessary refrigerant phase to each fan coil unit (Air Conditioning & Heat Pump Institute, 2010). A suction line and a liquid/gas line from the outdoor unit connect to one side of the controller. Multiple refrigerant supply and return connections are located on another face of the device; one of each is needed for every fan coil unit, requiring significant branch piping. Figure 2.6 illustrates a BC controller.

Figure 2.6 BC Controller. (Used with permission by Mitsubishi Electric).

Thus, refrigerant is delivered via the piping to each fan coil unit. Older VRF systems utilize R-22, which provides little to no savings in efficiency over traditional chillers (Goetzler, 2007). However, with the push for more environmentally friendly refrigerants, R-410a has become the more prevalent in VRF systems, with a 5% higher refrigeration efficiency than R-22

(Inman, 2007). Because of this higher refrigerant efficiency and the higher part-load efficiencies of the new systems that use R-410a, research by one VRF manufacturer has found that R-410a VRF systems can gain 30-40% energy savings over chiller systems (Goetzler, 2007); however, the refrigerant utilized by the chillers in this study is unknown. These savings may be inflated if the chillers in these studies utilized R-22. Additionally, these values may be skewed if the chillers and pumps in the comparison were single speed.

Controls must be provided for each zone served by a fan coil unit; these controls are then connected to a central control system. To create maximum levels of individual control and realize the full advantages of the VRF system, a thermostat is paired with each individual fan coil. If, however, the zone is large enough that multiple fan coils are needed to condition the space, it is possible to link one thermostat with multiple units. While no maximum number of fan coils can serve a single zone, each zone must have a minimum of one fan coil unit (Technology Spotlight: Heating and Cooling with VRF Systems, 2008). The temperature set point for each unit is monitored and set using direct digital control (DDC) technology; however, manual overrides could be provided at the space level to allow the occupants to modify the temperature as well. DDC allows for centralized control of these potentially complex, sprawling systems and permits for remote control of the system via the Internet (Variable Refrigerant Flow, n.d.). This function is useful for building owners and managers who want to be able to easily monitor and control temperatures in their buildings. BACnet or other traditional control systems can be paired with VRF, but some manufacturers have developed control systems to handle their specific systems. Using a manufacturer's HVAC control system ensures that the specific requirements are addressed for the complex VRF system, so if one is offered, it should be selected. If temperature setbacks are wanted when the space is unoccupied, the DDC software can be programmed with setback schedule. If this is not desired, a programmable thermostat linked to the control system can set temperature schedules in a space (Afify, 2008). Because the DDC control system could monitor occupancy allowing energy use to be minimized better than programmable thermostats could, this would be the preferred method for controlling the space temperature. In addition, with a ventilation air unit, the same DDC control system used for the fan coil units would need to control the ventilation system and any fans or dampers.

Chapter 3 - Design

Once a designer has determined that VRF is the best choice for a building, the design phase can begin; Appendix D contains a design flowchart to assist a designer at this stage. Ideally, a designer should select a VRF manufacturer before carrying out the design process; this could be based on quality of manufacturer regional support, company reputation, and designer familiarity with manufacturer products. Basing a design on a specific product is necessary because each manufacturer will have different system requirements. In the design process, the designer must make several design decisions based on the following questions addressed in this chapter:

- Will a heat pump or heat recovery system be used?
- What sustainable goals apply to the project?
- How will code compliance be achieved?
- Can the building latent load be met?
- How will piping be sized and routed?
- How will the building equipment be located for access?
- What controls will be provided for the system?
- What must be provided in the specifications, especially if a competitive bid is needed?
- What final steps should be taken to ensure correct system operation?

These considerations are laid out in the flowchart illustrated in Appendix A. A designer can reference this flowchart for the order of the design steps. Each step references a section of this chapter that explains further the considerations the designer must make. Although this reference can provide a designer with some insight into VRF, reading the paper in its entirety will make a designer most familiar with any potential limitations, advantages, or design requirements this system has.

3.1 Heat Pump or Heat Recovery System

When designing an air-to-air VRF system, designers should consider the number of zones needing cooling or heating at one time. When a building can be conditioned using all cooling or all heating, heat pump VRF systems are best; however, this situation is not often encountered in the commercial sector because most buildings have some sort of data or server room that requires year round cooling. However, heat pump systems require less piping, less equipment, and lower initial costs than heat recovery systems, which may make it a desirable option regardless of this need for year-round cooling. When some zones require year-round cooling, inline duct fans or exhaust fans could be installed in these areas to remove warm air and pull in conditioned air from surrounding space; of course, this applies only to rooms that do not contain highly sensitive electronic equipment. Alternatively, two separate heat pump systems could be installed; one could serve the year-round cooling areas, and one could provide conditioning for the other zones.

If the building being designed is expected to have some zones in cooling and some in heating simultaneously, the designer may want to consider a heat recovery VRF system. This type of system has higher initial costs than a heat pump system because VRF outdoor units are more expensive than the heating or cooling only units (Afify, 2008), in part because of the need for branch selectors and a third pipe. Even though this initial cost may be higher, heat recovery systems do use less energy than heat pump systems; therefore, long term costs will be lower. When deciding which option of VRF to choose, designers should remember that heat recovery VRF outdoor units are not available below 6 tons (Variable Refrigerant Flow, n.d.). Thus, buildings with block loads smaller than this may be better served by heat pump systems.

3.2 Sustainability

With more focus on sustainable design, a designer may need to create a VRF system that complies with one of several sustainable design guidelines, energy codes, or building rating systems. Meeting such requirements is possible with VRF. Because of the capability of the motors to operate at reduced speed, the VRF system reduces energy consumption when handling part loads. Heat recovery systems further reduce energy consumption by transferring heat from spaces needing cooling to spaces needing heating.

For sustainable design, a designer could reference one of the following codes: ASHRAE 90.1 - Energy Standard for Building Except Low-Rise Residential Buildings; ASHRAE 189.1 – Standard for the Design of High-Performance Green Buildings Except Low-Rise Residential Buildings; California Title 24; or local energy codes to achieve a rating with LEED or Energy

Star. ASHRAE 90.1 is quite commonly used, so learning its requirements for VRF is important for a designer who expects to use this technology. Although ASHRAE 189.1 has not been adopted as code yet, it is being considered by some jurisdictions and can provide a valuable sustainable design reference for any designer. California Title 24 requirements must be considered by designers who plan on working on VRF projects in that state. Finally, with the increasing popularity of LEED with building owners, likely, a designer will need to consider this certification.

In the summer of 2010, ASHRAE adopted the new AHRI Standard 1230 to provide minimum efficiency ratings for VRF under ASHRAE Standard 90.1-2007. These values are the minimum ratings for VRF equipment and are not difficult to attain. Indeed, the minimum efficiencies required are very low for what is achievable with VRF because ASHRAE did not want to provide an overly challenging requirement for such a new system (Ivanovich, 2010). Nonetheless, these efficiencies are higher than those required for other DX systems in ASHRAE 90.1-2007. The AHRI 1230 efficiency values required for air-to-air VRF systems per ASHRAE Standard 90.1, obtained from Mitsubishi Electric, are in Table 3.1. Currently, this is the only part of ASHRAE 90.1 that specifically addresses VRF systems; however, the requirements listed in Chapter 6 - Heating, Ventilating, and Air Conditioning, of ASHRAE 90.1 still apply to the system.

Table 3.1 VRF Minimum Efficiency Requirements.

Electrically Operated Variable Refrigerant Flow Air-to-Air Minimum Efficiency Requirements			
Size Category	Heating Section Type	Sub-Category or Rating Condition	Minimum Efficiency
<65,000 BTUh	All	VRF Multi-split System	13.0 SEER
≥65,000 BTUh and <135,000 BTUh	Electric Resistance (or none)	VRF Multi-split System	11.0 EER 12.3 IEER 12.9 IEER (as of 7/1/2012)
≥65,000 BTUh and <135,000 BTUh	Electric Resistance (or none)	VRF Multi-split System with Heat Recovery	10.8 EER 12.1 IEER 12.7 IEER (as of 7/1/2012)
≥135,000 BTUh and <240,000 BTUh	Electric Resistance (or none)	VRF Multi-split System	10.6 EER 11.8 IEER 12.3 IEER (as of 7/1/2012)
≥135,000 BTUh and <240,000 BTUh	Electric Resistance (or none)	VRF Multi-split System with Heat Recovery	10.4 EER 11.6 IEER 12.1 IEER (as of 7/1/2012)
≥240,000 BTUh	Electric Resistance (or none)	VRF Multi-split System	9.5 EER 10.6 IEER 11.0 IEER (as of 7/1/2012)
≥240,000 BTUh	Electric Resistance (or none)	VRF Multi-split System with Heat Recovery	9.3 EER 10.4 IEER 10.8 IEER (as of 7/1/2012)

ASHRAE 189.1 puts several constraints on a VRF system. VRF is not specifically addressed but applicable items that influence the system design are addressed. For example, Section 6.3.2.3 - HVAC Systems and Equipment, requires condensate to be recovered for reuse from units with capacities greater than 65,000 BTUh, and VRF fan coil units have capacities as high as 120,000 BTUh. If designing to 189.1, an appropriate recovery system will need to be designed for fan coil units that have capacities within this 65,000-120,000 BTUh range. Fortunately, this is not a significant obstacle as condensate collection and pipe routing is needed

for any system applying zone level cooling. To address the reuse aspect of this section, condensate piping can be routed to grey water retention vessels for recycling. Section 9.3.3 - Refrigerants, states that CFC-based refrigerants cannot be used; R-410a is not a CFC-based refrigerant, so specifying a VRF system using this refrigerant would meet the requirements of the standard. Because R-22 is a CFC-based refrigerant, VRF systems using this refrigerant cannot be specified to comply with ASHRAE 189.1. This does not pose a significant problem since most new VRF systems utilize R-410a in lieu of R-22.

The 2008 California Title 24 does not specifically address VRF. However, meeting the code requirements is possible by modeling the energy usage of the building according to the performance compliance path described in Chapter 9 - Performance Approach of this standard. Having lower consumption than the base energy usage defined by Title 24 can result in compliance; this should not be overly difficult to attain. Currently, EnergyPro and TRACE 700 are the only modeling programs that have options for modeling VRF and could be used to achieve compliance for Title 24 using the performance approach (Ivanovich, 2010).

VRF can be designed successfully to meet minimum requirements of the most popular building rating system, LEED 3.0 (administered by USGBC), and even generate up to 22 points toward the certification in the following categories:

- Energy and Atmosphere (EA) Credit 1, Optimize Energy Performance
- EA Credit 3, Enhanced Commissioning
- EA Credit 4, Enhanced Refrigerant Management
- Indoor Environmental Quality (IEQ) Credit 6.2, Controllability of Systems-Thermal Comfort
- IEQ Credit 7.1, Thermal Comfort-Design
- IEQ Credit 7.2, Thermal Comfort-Verification.

Each of these credits has aspects directly related to the design of a VRF system. Earning these credits is not guaranteed, however; a designer must be careful to comply with the requirements of each credit throughout system layout and design.

3.3 Code Compliance

As with any system, an extremely important step in the design of a VRF system is to consider its compliance with building health and safety codes, such as the International

Mechanical Code. Meeting these code requirements is relatively easy with this system; however, concerns can occur with the requirements of several standards referenced by the code, including ASHRAE 15 - Safety Standard for Refrigeration Systems, ASHRAE 34 - Designation and Safety Classification of Refrigerants, and ASHRAE 62.1 - Ventilation for Acceptable Indoor Quality. Although keeping refrigerant concentrations low and providing outdoor air may seem challenging, VRF systems installed in Europe have met requirements of similar European codes (Goetzler, 2007).

Designers can assure compliance with ASHRAE 15 - 2004 and ASHRAE 34 - 2004 simply by following the guidelines set forth in these standards. Since these standards deal with the concentration of leaked refrigerant in occupied space, it is important for designers to realize that the entire refrigerant charge can escape through one leak because one set of pipes serves all units. The following discussion shows how a VRF system utilizing R-410a determines compliance with these standards.

The first step in reaching compliance is to determine the maximum concentration allowed in a space. Table 1 in ASHRAE 15 lists common refrigerant blends and their respective allowable concentrations; a designer should be aware that for institutional occupancies, buildings in which occupants cannot leave because of health or security reasons, the values in this table are reduced by 50% per Section 7.2.1. These concentration values are all based on ASHRAE 34. In the case of a discrepancy between the two standards, ASHRAE 34 governs. Because R-410a has not been included on Table 1, ASHRAE 34 must be consulted for maximum allowable concentrations of this refrigerant. Per ASHRAE 34, Table 2, R-410a has a Refrigerant Concentration Limit (RCL) of 25 lb/1000 ft^3.

The second step is to determine the minimum allowable room volume for the VRF system being considered. The refrigerant charge of the system, often given in pounds, must be divided by the RCL. The resulting volume will be the minimum volume any room containing refrigerant piping or equipment can have without additional of mechanical ventilation. This example assumes a refrigerant charge of 40 lbs; a designer could find this value for a VRF system in the manufacturer's outdoor unit data. Dividing 40 lbs by 25 lb/Mcf renders a minimum room volume of 1600 ft^3.

Once the minimum allowable room volume has been determined, the designer must calculate volumes for all rooms that piping is routed within to ensure that none fall below this value, referring to ASHRAE 15, section 7.3. This section provides guidelines explaining what determines the boundaries of a completely enclosed room. If any completely enclosed room volumes are less than the minimum allowable volume, mechanical ventilation must be installed per Section 7.3.1. For example, a room with a volume of 1500 ft^3 would not meet the minimum room volume requirement of 1600 ft^3; thus, mechanical ventilation would be required.

Whenever the maximum allowable concentration of refrigerant of either code is exceeded for a space, mechanical ventilation can remove the refrigerant in the case of a leak. To minimize the need for mechanical ventilation, the piping could be strategically routed only through spaces exceeding the minimum volume; for instance, a ducted VRF unit could be located outside the space to prevent refrigerant piping from being routed in these small rooms. Also, the room volume could be increased to dilute the refrigerant by raising the ceiling, providing an undercut door to another space, or connecting the space to another room via a transfer grill (Variable Refrigerant Flow, n.d.). However, this solution may not always be possible for several reasons. First, architectural or construction limitations may exist preventing the implementation of this method. Second, additional expense can be incurred by undercutting a door or by supplying a transfer boot; in the case of a fire wall penetration, this cost would increase even more due to the need for a fire/smoke damper. Finally, creating openings between rooms can create an open path for noise; this could be a major concern with adjacent counseling or patient rooms.

ASHRAE 62.1 compliance can also be achieved using VRF. Some manufacturers will offer outdoor air kits to at least precondition ventilation air being sent via ducts to the fan coils. However, if the external static pressure capabilities of the outdoor units are exceeded, a separate DOAS can condition ventilation air (Afify, 2008). With a separate system, ductwork will need to be routed throughout the building to each space requiring ventilation air as described in Section 4.4. Designers should always check manufacturer data to ensure that an outdoor air ductwork connection can be made to the fan coil unit. If not, the outdoor air ductwork could be connected to return air ductwork on ducted units or supplied directly to the space via a diffuser.

3.4 Latent Heat

As Section 4.4 mentions, VRF, like many cooling systems, has limited capacity to neutralize latent heat, which can be introduced to a system externally via ventilation air or generated internally by people, equipment, or infiltration. When selecting VRF equipment, the designer must remember that latent heat removal only occurs in the cooling process when enough sensible cooling has occurred to allow latent cooling. This process is illustrated by the psychrometric chart in Appendix B. In most VRF applications, this is not a problem. If latent loads are found too high according to psychrometric analysis, the problem can be solved by selecting a ducted unit and setting it to cool the air low enough to remove moisture from the air stream. Because this air will be too cold to deliver to the space, an electric heating coil can be placed in the supply ductwork to reheat the air to an acceptable delivery temperature. This solution, illustrated on a psychrometric chart in Appendix B, should only be applied when absolutely necessary since the process is inefficient and may lead to problems with static pressure drop for the ducted fan coil unit. To reduce the negative effect reheating has on efficiency, a sequence could be programmed into the building controls system to reduce the amount of cooling done by the DX coil when relative humidity is low enough. As Section 4.4 states, energy recovery ventilators can quite effectively remove latent heat from ventilation air, although they may be initially more expensive.

3.5 Building Load Profile and Unit Selection

Selecting equipment that can handle the load is essential to proper building conditioning and system performance. Ventilation loads, envelope loads, and internal loads must be determined in order to size indoor and outdoor equipment. Designers should first select the indoor fan coils followed by selecting an outdoor unit to accommodate those indoor units.

Each individual fan coil is sized based on the peak load, or worst case load, of the zone being served. Over-sizing the units must be avoided to avoid problems with humidity control, to permit the electronic expansion valve to modulate, and to avoid problems with pressure drop and fan coils receiving the wrong amount of refrigerant (Variable Refrigerant Flow, n.d.). For this reason, the engineers should calculate accurate loads for the space, including both sensible and latent loads.

The total capacity of all the fan coils can be greater than, equal to, or less than the capacity of the outdoor unit, which must be able to handle 70-130% of the total fan coil cooling or heating capacity. Manufacturers have various limitations on sizing outdoor units any higher than this range; among manufacturers, the most a unit can be over-sized is 200% of the total indoor unit capacity. Because outdoor units only need to handle the block load, or overall peak load, of the building, the capacity will likely be smaller than the total fan coil unit heating and cooling capacity. An instance where the outdoor unit selection would be equal to or greater than the block load is if the owner plans to expand the facility and HVAC system in the future. A higher capacity unit is acceptable as it will still run effectively, although this situation should be avoided unless an expansion is planned for the building (Afify, 2008). Efficiency will drop with an oversized unit because although the variable speed compressor can accommodate the smaller load, it will have to shut off sooner than a smaller compressor would. As a result, more time will be spent cycling the compressor on and off. To avoid this, many manufacturers offer system selection software to help the designer select appropriate units.

3.6 Piping

Designers must consider building characteristics before designing piping for a VRF system. Moreover, a large building requiring long runs of refrigerant piping will result in higher costs, both initial and long term. First, more piping must be purchased and installed initially. Also, longer pipe runs require higher operating costs because more energy can be lost from the refrigerant while being routed to the fan coil units (Afify, 2008). One way to minimize pipe length is to consider building configuration. For example, buildings with multiple roof levels allow the designer to place the outdoor units on different roof spaces, reducing the distance to the fan coils each outdoor unit is serving and therefore minimizing pipe lengths (Afify, 2008). For other buildings, placing the outdoor units on one roof of the structure or at ground level will be the only options for the designer, which may prohibit significant piping length reductions. As with any DX system, piping must be insulated to meet local code requirements; thus, the entire length of pipe must be insulated in this system regardless of allowable pipe length.

Notably, a VRF system cannot be installed in a building if the maximum allowable lengths of refrigerant piping defined by the manufacturer are exceeded. Currently, the maximum vertical distance between the outdoor unit and the most remote indoor unit for any manufacturer is 295 feet, and the maximum overall length is 540 feet. Also, only 49 feet of vertical piping, supply or return, can be installed between two indoor fan coil units (Variable Refrigerant Flow, n.d.). Additionally, a few manufacturers limit the maximum allowable distance from the first piping tee to the furthest indoor unit; obviously, designers should consult manufacturer technical information for specific requirements and allowable pipe lengths. In addition, R-410a systems utilize smaller pipe diameters than R-22 systems; based on manufacturer data, this difference is approximately one pipe size. Designers should contact the manufacturer or utilize manufacturer computer software for pipe sizes since they will vary depending on the manufacturer selected because the basic system layouts are different for each.

3.7 Power and Maintenance Accessibility

Designers need to consider the location of equipment when designing a VRF system to account for power and maintenance. Compressors, branch selectors, fans, and condensate drain pumps are all components of the VRF system that require power. Because the outdoor units and fan coil units house this equipment, designers do not need to worry about providing a power connection for each piece of equipment individually. Instead, only one power connection is required for each outdoor unit and fan coil unit. Most manufacturers require a 208 volt/3 phase or a 240 volt/1 phase connection for fan coil units, while outdoor units can be specified as 208, 240, or 480 volt, 3 phase. Manufacturer data will clearly specify what power is required for a given unit. As with any piece of mechanical equipment, a disconnect should be provided for every unit to comply with the National Electric Code. Additionally, water-proof GFCI receptacles should be located outdoors or on the roof per the NEC for power to service the outdoor units.

Ensuring efficient operation of a VRF system requires proper maintenance. The units contain mechanical components that will eventually fail, and as such must be readily accessible for repairs or replacement (Afify, 2008). As with any system, operation and maintenance manuals should always be provided to the owner after the installation. Educating the owner and any facility managers about the maintenance needs of the system will ensure operation beyond the time of commissioning and minimize efficiency loss over time. Specifically, VRF compressors and condensers in outdoor units will need routine maintenance; indoor unit contain filters that must be cleaned. If refrigerant piping develops a leak, this must quickly be repaired as well.

3.8 Controls

Because VRF is a complicated technology, a controls system is necessary for correct operation. As Section 2.1.2 mentions, VRF manufacturers usually offer DDC systems for their product. This enables each system to be controlled by a single PC or over the internet. Each zone must have a thermostat setpoint for the corresponding fan coil units; this can be set via the central control point. However, manual overrides for each space can allow occupants to modify temperatures. Additionally, multiple units can operate off one thermostat, or each unit can be controlled individually, depending on the zoning requirements.

When determining the controls sequence for a VRF system, a designer must consider several main issues. First, if ventilation air is to be reduced during unoccupied mode, more return air must be allowed to enter the fan coils. With ducted units, this requires a motorized damper. Second, temperature setbacks should be programmed into the system during unoccupied mode to allow VRF to reduce its energy consumption. However, buildings that do not have strict occupancy schedules should have manual overrides in each space to allow occupants to change the temperature setpoint during off-peak hours. Third, with so many units throughout the system, a designer should set the controls system to provide information about where a disabled unit is located. This will help building maintenance to correct the issue quickly with minimal down-time. Finally, the designer must make sure that the expansion valves of the indoor units are properly interlocked with the motor controller of the outdoor unit. This is essential to permit the variable speed compressor to change speeds and adapt to the part loads in the building.

3.9 Specifications

When writing the specification for a VRF system, it is important in many projects for the designer to allow for approved equipment substitutions to allow for a competitive bid. With the wide discrepancies that exist between manufacturer system requirements, it can be difficult for a designer to list acceptable system substitute systems. The solution is to write a performance specification requiring the contractor to install the system based on manufacturer requirements. It is relatively easy to substitute one manufacturer’s outdoor or indoor unit for another based on unit capacity; the real problem is encountered when dealing with the piping sizes and lengths required by the different systems. Fortunately, if a contractor proposes a different manufacturer than scheduled, that manufacturer will provide all necessary pipe diameters and allowable lengths. The specifications should require that this piping be installed in the same locations indicated in the original design. Consequently, refrigerant concentrations may be a problem because of changes in piping diameter and length; a designer should double check the refrigerant concentration calculations to ensure that the proposed substitutions meet the requirements of the initial design. Unfortunately, this forces the designer to spend more time on a VRF project, increasing the engineering fees.

Ultimately, substituting systems could increase the number of outdoor units, affect piping lengths, require different controls systems, and influence overall system costs; however, this does not mean that the substituted product would perform less efficiently. Since added system costs should be avoided, the designer can require contractors to pay for any additional costs incurred by substituting equipment; but often contractors have no way of knowing exactly what this cost difference is. However, the contractor could base the bid price on the system originally selected by the designer; any cost above this bid would be handled by the contractor. Appendix C contains an example VRF specification provided by TME, Inc. Three acceptable manufacturers are listed at the beginning of the specification, chosen because of similarity of product; this is one way to provide for a competitive bid without significantly affecting the design. All operation and installation requirements for the system are given. Note that while this specification solely addresses VRF equipment and piping, it does reference a separate controls specification section; a controls specification should be done to clarify what controls are preferred for the system.

3.10 System Completion and Commissioning

After the design has been completed, the contractor must install the system according to the manufacturer's requirements. Most manufacturers will offer training classes for contractors to learn about the installation requirements for their specific system (Afify, 2008). Contractors must consider several issues especially when installing refrigerant piping. All refrigerant piping must be leak-free copper that has been kept clean and dry. To accomplish this, all piping must be sealed closed when stored on site to prevent debris or moisture from entering. Brazing, the use of a filler material to connect two pieces of metal, combined with the use of flared fittings is the most common method of installing the piping (Goetzler, 2007). If the piping is welded together, which involves melting the ends of the pipe together, nitrogen gas must be used to prevent oxidation reactions from occurring on the inside of the pipes (Afify, 2008). Because refrigerant leaks are dangerous and costly, contractors must install piping correctly to decrease the chance of these occurring (Amarnath & Blatt, 2008). As with any refrigerant-based system, the piping should be tested for leaks prior to receiving the refrigerant charge.

After installation, the designer must ensure that the system will operate correctly to achieve maximum efficiency. Site visits and final punch lists can be used to inspect the system for correct installation, which has a direct effect on the immediate operation of the system. However, the designer should also ensure the correct operation of the system over its life by informing the owner about the maintenance needs. Moreover, training on the system requirements could be called for in the mechanical spec.

Commissioning is essential for ensuring that the VRF system is operating properly. Also, if LEED certification is a concern, commissioning is part of Prerequisite 1 for Energy and Atmosphere. With a VRF system, all equipment must be operating correctly, the controls system must be modulating the frequency of motors based on thermostat readings, the outdoor unit must be providing heating and cooling at peak loads and ambient extremes, and self diagnostics must be operating properly to alert the occupants to any problems (Afify, 2008). Performance should be monitored for a year to ensure that the system operates properly in all seasons.

Chapter 4 - System Selection

VRF has great potential; however, like any other system, it has functional limitations. To avoid unnecessary expense of design time and money, a designer must understand when the system is appropriate; knowing this will help give owners the best possible HVAC system for their building. Several aspects critical to selection are zoning, climate, refrigerant, ventilation, costs, and building space allocation for mechanical equipment.

The specific considerations a designer must address are laid out in the flowchart in Appendix D. Each box in the chart presents questions and considerations for the designer. A corresponding section of this paper is listed in each section of the flowchart so the designer can locate more information quickly.

4.1 Zoning

Zoning needs must be understood when considering a VRF system; for instance, as with any system, zones should group only those rooms that have similar use and exterior envelope exposure. Also, zones will vary in size and peak heating and cooling load, therefore influencing the required capacity of the indoor fan coil units. When many zones are needed, VRF systems can control each individual indoor fan coil unit with its own thermostat. In addition, if expansion or remodeling is expected, VRF provides great flexibility since existing controls can be easily modified and fan coil units can be added to the system.

To verify if one or multiple fan coil units are needed to meet the zone load requirements, manufacturer data should be referenced, which will list the minimum and maximum capacities of the units. Each zone must have at least one unit, although large zones may require multiple fan coil units to meet the heating/cooling demand. Outdoor units have a maximum capacity of 60 tons, but multiple units can be used if the building load exceeds this value; therefore, no upper limit to the number of zones the system can handle exists as long as the system complies with maximum piping length limitations as discussed in Section 3.6. However, VRF systems do have a minimum number of zones; buildings with fewer than four zones are better served by other systems since the initial cost of a VRF system outweighs the energy savings with so few zones (Inman, 2007).

4.2 Refrigerant

VRF systems route large amounts of refrigerant throughout the building, making the safety of the occupant a consideration with this system; refrigerant volumes tend to be higher than for ducted conventional split systems because the length of piping through the space is much higher for VRF systems. Specifically, refrigerant is heavier than oxygen, meaning a potential leak can displace air in the breathing zone, suffocating occupants. To address this issue, room sensors could activate an alarm when oxygen levels are too low, or a sensor could measure the pressure in the piping and notify occupants if the refrigerant pressure in the system becomes too low; however, this does increase the initial costs of the system and is not required by code. ASHRAE 15- Safety Standard for Refrigeration only requires detection in machinery rooms for equipment exceeding limits in Section 7.2. – Restrictions on Refrigerant Use. This does not pertain to VRF since the outdoor units are the only pieces of equipment in the system that exceed these limits, and they are located outdoors rather than in a machinery room. A designer might still want to supply this alarm system since a leak is otherwise undetectable by the occupant since refrigerant has no odor. In most applications, a warning system is sufficient because occupants will know to leave the building when the alarm goes off. However, incapacitated, young, or constrained occupants might not be able to leave a building; this situation might be encountered in hospitals, day-care facilities, or penitentiaries. Some designers may choose to avoid refrigerant-based systems altogether in these cases; however, installing mechanical ventilation or ducted units could decrease the risk associated with a leak in these cases.

If refrigerant concentrations exceed the allowable limit for a space, the solutions discussed in Section 3.3 can be implemented. However, if these changes cannot be made, VRF would not be appropriate for such a building. Either a forced air system or a hydronic system would be a better solution.

4.3 Climate

Climate is important when considering whether or not to design an air-to-air VRF system. This can be related to lower efficiency than for other technologies or to the possibility of reduced capacity at temperature extremes. Such conditions can be encountered in dry, hot areas and in cold, northern areas.

In dry, hot regions of the United States, primarily in the southwest, identified by ASHRAE 90.1 as climate zones 2-5B, evaporatively cooled systems are usually the best choice for buildings because of the low wet bulb temperatures in these regions. Here, the effectiveness of air-cooled systems such as VRF should be carefully compared to that of traditional evaporative technologies. An evaporative cooling system could possibly challenge VRF in terms of effectiveness. A closed looped water-to-air VRF systems could be considered in lieu of a standard air-to-air outdoor unit to capture the benefits of both technologies; more research on the efficiency of this type of VRF system should be done to investigate this option.

In very cold winter climates, including Midwestern, northern areas of the U.S. defined by ASHRAE 90.1 as zones 6-7, the heating capacity of air-to-air VRF is compromised (Variable Refrigerant Flow, n.d.). Reduced heating capacity has obvious negative implications. To address this issue, many manufacturers have developed low-ambient kits that boost the output of the system at low temperatures; if outdoor temperatures are so low that the kit cannot raise the fan coil entering air temperature high enough, supplemental electric heating coils must be supplied for each fan coil (Afify, 2008). An energy recovery ventilator could also be used to temper the outdoor air entering the space. Although VRF can still be used in harsh, cold climates, designers must check manufacturer data to confirm that the VRF equipment selected can handle the building loads in worst case design temperatures; if not, additional measures must be taken.

4.4 Ventilation

VRF has several limitations concerning outdoor air for the designer to consider before selecting the system. For instance, VRF fan coil units have maximum and minimum entering dry bulb and wet bulb temperatures. These values vary based on manufacturer, but values are generally near 80°F db/ 65°F for cooling and 70°F for heating. Temperatures outside the specified ranges require oversizing of indoor units. However, oversizing is limited to around 20% extra capacity by most manufacturers to avoid problems caused by freezing the indoor unit coils. If outdoor air temperatures are too low for the system to heat ventilation air to design temperatures, supplemental electric heating coils can boost heating output (Afify, 2008). Also, high temperatures and humidity cause problems for cooling (Afify, 2008). In VRF systems, as is the case with all systems, latent cooling occurs only as sensible cooling occurs; then, once the temperature set point is reached, no more latent heat is removed from the space. This process is explained further in Appendix B. If latent loads are expected to be an issue, as is the case in very humid climates, the supply air can be subcooled by the indoor unit and reheated to the design supply temperature with electric heating coils before being discharged into the space, although this will greatly compromise system efficiency. In addition to limited load capacity, the fan coil units have low external static pressure capabilities, which may cause problems in routing ventilation air ductwork to spaces to meet the requirements of ASHRAE 62.1 - Ventilation for Acceptable Indoor Air Quality. Several options to address these issues are available allowing a designer to choose between a system that is integrated with VRF and a system that is completely separate from this technology. These options are listed as follows:

1. Manufacturer outdoor air kits conditioning/preconditioning ventilation air with designated VRF heat pump outdoor units
2. Manufacturer outdoor air units utilizing energy recovery in conjunction with a designated VRF heat pump outdoor unit
3. DOAS units handling ventilation load
4. Energy recovery ventilators (ERVs) conditioning/preconditioning ventilation air.

First, manufacturers have developed outdoor air kits that are integrated with the basic VRF system (Afify, 2008). These outdoor air kits consist of a VRF unit usually served by a designated heat pump outdoor unit; a heat recovery unit is not necessary because ventilation air through the unit can only be cooled or heated at one time. These 100% outdoor air units can route conditioned air directly to the space as supply; but in some climates, they cannot provide the supply air at ideal temperatures, limiting their use to preconditioning of outdoor air sent to fan coils (R. Froebe, personal communication, September 16, 2010).

Another option is manufacturer developed outdoor air units that utilize energy recovery to moderate the load. If the selected manufacturer has designed outdoor air ductwork connections on the fan coil units, as is common, the ventilation ductwork connected to an outdoor air unit can be ducted directly to the indoor units. If not, the ventilation ductwork must be routed to a diffuser in the space or connected to return air ductwork connected to the fan coil. Figures 4.1 and 4.2 illustrate how a 100% outdoor air unit utilizing energy recovery might be configured within the space; no refrigerant piping is shown for the sake of clarity, but piping is needed to connect each outdoor unit to its designated indoor equipment.

Figure 4.1 100% Outdoor Air Unit Independent of VRF Fan Coils

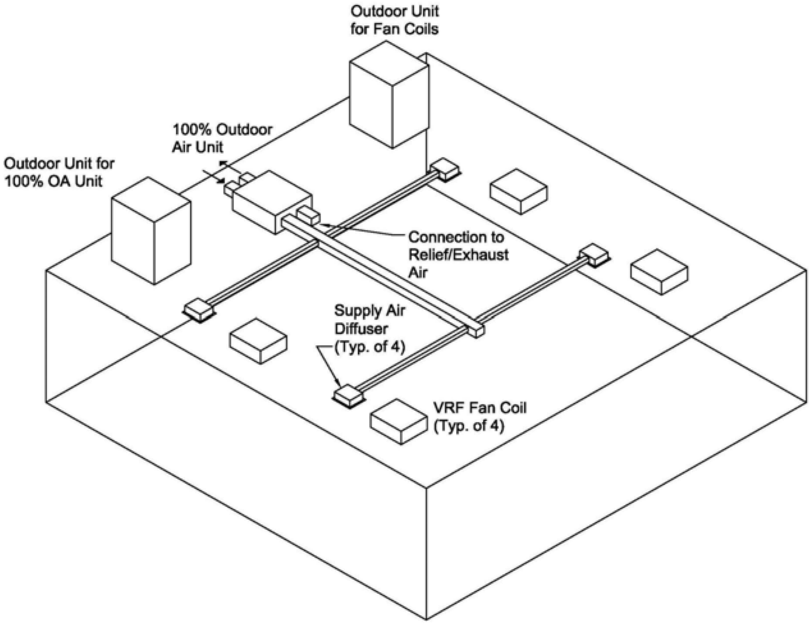
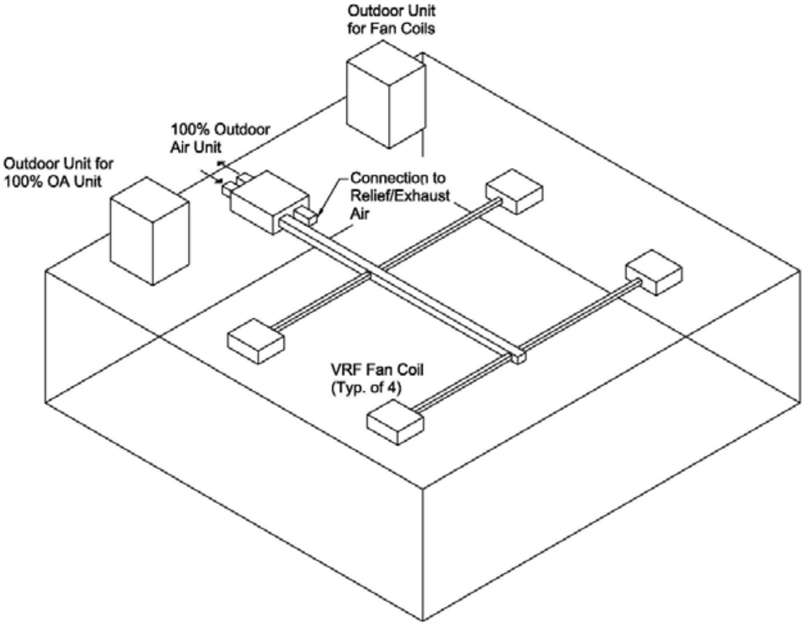


Figure 4.2 100% Outdoor Air Unit Connected to VRF Fan Coils.



External static pressure capabilities are limited for the fan coil units and manufacturer outdoor units; as a result, a large building with significant ventilation duct lengths may require a separate designated outdoor air system, referred to as a DOAS (Afify, 2008). A separate air handling unit (AHU) could be specified to serve as the DOAS, whose fan could handle the high external static pressure caused by long duct lengths. An extra diffuser would be required in each space needing outside air, or ventilation air could be ducted to the indoor unit return ductwork.

Alternatively, an energy recovery ventilator (ERV) could be specified with heating and cooling coils to provide conditioned ventilation air at the space level. However, the cost associated with these coils could be eliminated by using the unit to handle the static pressure drop from the outside air intake to the fan coil and to moderate the heating and cooling loads via the unit's energy wheel. This preconditioned air could be sent to the fan coil units and heated or cooled further as needed.

4.5 Costs

As with any system, the costs associated with a VRF system are an important consideration. As discussed further in Chapter 5, the initial cost of the VRF system can be higher than that of more conventional systems, primarily because of installation. This is expected to decrease as contractors become more familiar with the product (Goetzler, 2007).

When replacing existing systems, owners and designers must consider the cost of demolishing the installed system and designing and installing a new VRF system. In some cases, existing components can be reused for the new VRF system, reducing installation costs. For example, ductwork and diffusers previously connected to VAV boxes or hydronic fan coil units could be reused for ducted VRF units. This would depend on the new VRF system utilizing a supply air temperature low enough to prevent airflow rates from being higher than in the previous system; this should be checked since VRF fan coils run constantly, and the temperature of the supply air is increased or decreased as needed. Another example could be ductwork mains connecting VAV boxes to air handling units being reused as outdoor air only ductwork for a VRF system. This ductwork would probably be much larger than necessary but would not have a negative impact on the performance of the system. Thus, while the AHU would need to be replaced with a new unit sized to handle the outdoor air, the cost of new ventilation ductwork could be avoided.

Before final system selection, the life cycle costs of VRF must be compared to those of other systems being considered. A comprehensive analysis will show the expected long-term costs for the owner including energy, maintenance, and equipment replacement costs. If the life cycle costs of VRF are too high, the owner may prefer the use of another system.

Furthermore, the cost of replacing an existing system must be compared to the cost of repairing the system. For example, replacing a chiller that already has hydronic piping installed in the building could be a much less expensive option than installing a new VRF system, which would require all new indoor and outdoor units with refrigerant piping (Goetzler, 2007). The replacement cost of the chiller would likely be lower unless the new chiller had to be installed in an inaccessible space. This situation might be encountered when a chiller is located within a building rather than a central energy plant. In this situation, building walls may need to be demolished and rebuilt in order to access the chiller space. Additionally, getting the new chiller to the designated area could prove very difficult because of its weight and size.

An analysis of the life cycle should help determine the most appropriate solution for all applications. This analysis should include all costs associated with the potential systems: first cost, utility costs, routine maintenance costs, and system and equipment replacement costs. Life-cycle cost analyses are important to accurately inform an owner about the most effective system for a building from a cost standpoint. Owners investing in VRF want to know exactly how long the pay-back period is when compared to that of other systems being considered; this can be less than five years when VRF is compared to some systems but may be much too long to be considered competitive against other systems.

4.6 Building Space Allocation

VRF components require space throughout the building. The amount and location of space occupied by the equipment varies depending on the size of the building and the style of units selected for the system. However, a significant advantage to a VRF system is that it does not require a central mechanical room.

Outdoor units can be located on the ground adjacent to a building or on the roof of the building. Depending on the size of the building, multiple outdoor units may have to be installed to meet maximum piping length requirements or to accommodate building load. This may cause aesthetic issues, so placement of these units should be carefully considered. In some situations, visible exterior equipment may be a serious drawback; a designer must coordinate with the architect when this is a concern. If a good location for the outdoor unit cannot be found, the designer may have to consider another system such as a ground source heat pump. This is really the only main space allocation concern that may confront a designer; the unit sizes and noise levels are very reasonable compared to those of exterior equipment associated with other systems.

Space required by indoor units varies widely depending on what type of unit the designer has selected and the size of the zone. For instance, ceiling cassette and concealed ducted ceiling units require plenum space; both standard sizes and low profile units are available for designers to meet plenum space limitations or to conserve floor space when a ceiling plenum is not available. Alternatively, ceiling suspended units can be hung from the ceiling, which may not be desirable for aesthetic reasons. Finally, cabinet style units require space within the actual occupied area, either on the floor or mounted on a wall. Ultimately, this aspect of the system should not be a significant drawback since any system will have indoor units that require some form of plenum or room space.

Chapter 5 - Comparison of VRF to Conventional Systems

Although VRF was developed as an alternative to chiller systems, it can be used in place of several different conventional systems. Comparing VRF with some more familiar systems will help illustrate the potential advantages and a few limitations of this technology. This can also aid a designer in deciding whether a VRF system is the appropriate choice for a specific application.

5.1 VRF vs. Chiller/Boiler Hydronic Systems

VRF can be used in many situations in which central chiller and boiler systems are applied (Bergman, 2008). Both types of system utilize a central unit rejecting or adding heat to the system and fan coil units located in the space. Both must likely utilize some form of DOAS as well to provide ventilation air at the space level; VRF units have low capacities and hydronic fan coil units have limited static pressure capabilities. New chillers have efficiencies higher than VRF at loads greater than 90%; however, because most chillers will spend the majority of the time operating at a load below this, VRF can gain an advantage in overall efficiency (Goetzler, 2007).

In a manufacturer study, VRF systems were generally more expensive to install than chiller/boiler systems, ranging from 8% higher than water cooled chiller systems to 16% higher than air cooled chiller systems. However, according to the manufacturer, the maximum payback period for VRF compared to either of these systems is 1.5 years (Amarnath & Blatt, 2008). These values do not hold true everywhere; for example, a case study by a manufacturer of a 43,000 ft^2 German hotel showed identical costs for both systems. This case study showed that the VRF units were more expensive than the units for a hydronic system utilizing an air cooled scroll chiller, but the VRF insulation, valves, and installation were much less expensive than the hydronic system counterparts. It is unlikely that initial costs for the VRF system will be less than for a hydronic chiller-based system in the United States because currently, installer unfamiliarity will drive VRF costs 5-20% higher than conventional system costs (Roth, et. al., 2002). However, the energy consumption for VRF in some climates has been found to be much lower than for a chiller-based hydronic system. For example, a manufacturer study performed in Italy showed 35% savings in energy use over traditional chiller and boiler systems (Amarnath & Blatt, 2008). However, the life cycle of VRF components is shorter than that of chillers. VRF components can last from 10-20 years (Roth, et. al., 2002), while chillers can perform for as long as 30 years (Goetzler, 2007). Despite this, the Italian study showed 40% lower VRF maintenance costs than for chiller/boiler systems (Amarnath & Blatt, 2008). The exact setup of the study and the systems are unknown for these studies, but the results illustrate the necessity to perform a complete life cycle analysis during the system selection phase. Clearly, the designer must carefully consider initial cost, energy use, maintenance cost, and equipment life-span of the technologies being compared. In addition, this study shows the need for independent parties to research VRF systems in comparison to other systems to generate truly independent data.

5.2 VRF vs. Conventional DX Systems

The substitution of VRF systems for typical DX cooling systems can be beneficial for two reasons. The modulation of the expansion valve in the VRF fan coil unit minimizes the frequent on/off cycling common with more conventional DX systems. As a result, the cooling demand is more accurately met since the space temperature fluctuation is much smaller; temperature drift can be as low as ±1°F. In addition to creating a higher level of comfort in the space, VRF allows longer lengths of refrigerant piping than some common DX systems, enabling designers to reach more remote sections of buildings with the same refrigerant system (Xia et. al., 2004).

Two DX systems that are most similar to VRF systems are multi-split and conventional split systems. These two systems operate on the same principle as VRF systems; refrigerant is cooled in outdoor units and piped to some style of indoor unit. All three technologies require a DOAS since the indoor units have low capacity to treat air at temperature extremes. Neither conventional technology has an electronic expansion valve at each coil within the fan coil unit modulating the flow of the refrigerant or the variable speed compressor in the outdoor unit (Amarnath, 2008). Additionally, the outdoor units of both conventional systems have less capacity than do VRF outdoor units.

Multi-split systems are DX systems that can provide a high degree of zoning, just like VRF; these could likely be used in small to medium sized commercial buildings. However, while VRF units only need one set of piping connected to the outdoor unit, multi-split systems require multiple sets of refrigerant piping to be connected to the outdoor unit; each individual unit has a separate piping circuit, which creates the need for more piping and more building penetrations. Because this simplifies the actual piping run, multi-split systems can have longer allowable piping lengths than VRF; piping lengths can be over 250' for some multi-split manufacturers. However, multi-split systems do not utilize ducted indoor units and have a smaller load capacity than VRF systems (Amarnath, 2008). Additionally, these systems can only function as all cooling or all heating; they offer no option for heat recovery.

A conventional split system only has one set of piping connected to the outdoor unit from a single indoor unit, resulting in an outdoor unit for each zone. In a facility requiring many zones, this can become prohibitive due to lack of outdoor space or aesthetic issues. In addition, conventional split systems require a furnace to provide heating. Also, conventional split systems typically have significant amounts of ductwork compared to VRF since the zones are typically larger; this is done to minimize the number of outdoor units. This results in larger duct losses, decreasing the efficiency of the system. VRF needs only one outdoor unit to serve multiple zones allows for fewer outdoor units, smaller and more effective zoning, less ductwork, and less individual refrigerant piping runs, leading to significant equipment and energy savings in many applications.

In a study by the Electric Power Research Institute (EPRI), VRF used less energy than existing ducted split systems. In an office facility in Knoxville, Tennessee owned by EPRI, an installed VRF system had up to a 48% energy use advantage over the old split systems previously installed. In addition, occupants in this study indicated that the VRF system provided excellent thermal comfort (Amarnath, 2008). Several reasons exist for this significant advantage. VRF systems have higher part load efficiencies than split systems because of lower duct losses within the system (Goetzler, 2007). This is a result of the smaller zones in a VRF system; conditioning is localized rather than centralized. As a result of these small zones and localized conditioning, VRF energy use can be reduced since temperatures can be set back in unoccupied rooms without compromising comfort in occupied spaces. One other aspect that favored VRF over split systems in that particular study is that the VRF system was new and the split system was existing. An existing system might run at a lower efficiency than could be attained with new equipment; this would allow VRF to gain a bigger advantage.

One area in which more conventional forms of DX cooling outperform VRF is in large spaces with minimal or no zoning need. Rooftop systems are still the best choice for these applications (Goetzler, 2007). Where large volumes of air need to be conditioned, there is no need for the individual control and flexibility that zoning provides in a VRF system. In addition, rooftop units eliminate the need for a DOAS since ventilation air is treated by the unit. Ultimately, rooftop units are much less expensive and have fewer components requiring maintenance than do VRF systems.

5.3 VRF vs. Heat Pump Systems

VRF and air-to-air heat pump technology are similar systems because heating and cooling is possible with one outdoor unit; additionally, both can provide high levels of individual control and energy savings. While VRF systems are generally more expensive than heat pump systems, ton for ton, in an air-to-air heat pump system, each zone must have its own outdoor unit. This results in multiple electrical connections for outdoor units and prevents the system from being able to utilize heat recovery. A connection is required for every VRF outdoor unit as well, but the higher capacity capability of VRF outdoor units reduces the total number of units required. Additionally, air-to-air heat pumps require more refrigerant line exterior envelope penetrations than VRF outdoor units; this can add to installation expenses. With these considerations in mind, VRF technology should be considered over air-to-air heat pumps when the design requires more than three zones. In applications requiring three or fewer zones, air-to-air heat pumps are a better application (Inman, 2007).

VRF can also be compared to geothermal and water source heat pump systems. In an energy modeling study utilizing DOE-2, geothermal heat pump systems offered up to 36% higher energy efficiencies than VRF systems (Liu & Hong, 2010). However, the upfront cost of VRF systems is lower than the cost of geothermal systems because VRF systems do not require the pumps and bore fields associated with geothermal technology. Also, VRF systems have an advantage over geothermal systems in that they need very limited space; geothermal systems require significant land and mechanical room space. Water source heat pumps are often less expensive to install than VRF systems (Goetzler, 2007). However, these would have difficulty competing in terms of efficiency because they are dependent on the efficiencies of the cooling towers used in conjunction with the heat pumps. Geothermal and water source heat pump systems both require a pump to move water throughout the system; conversely,VRF is not dependent upon pump efficiency because it has no pumps. Instead, the compressor provides adequate pressure to move refrigerant throughout the system. Nonetheless, a building owner who has the space and money for a geothermal system may want to consider using this technology in lieu of VRF.

5.4 VRF vs. VAV Systems

VRF and variable air volume, or VAV, systems are both useful in larger buildings requiring multiple zones with different conditioning requirements. Both have desirable attributes. VRF reduces electricity consumption with a variable speed compressor at the outdoor unit and multiple small variable speed fans at the fan coil units. In comparison, a VAV system requires a large fan to force the air through long lengths of ductwork, which can cause significant duct loss. Also, a VAV system cannot turn off the central unit when at partial load. These factors contribute to VRF systems having a higher efficiency than VAV systems. In one study, a VAV system was 38% less efficient than VRF. In this study, the Los Angeles Department of Water and Power compared a 6-ton VRF system to a 7-ton VAV system on the west wing of one of their buildings. The different systems were alternated weekly to provide conditioning for the space (Nye, 2002). Although significant savings were recorded, it is most likely that savings around 5-15% would actually be realized because this study compared a new VRF system to an existing VAV system (Amarnath, 2008).

In addition to the energy use reduction, VRF also provides higher levels of occupancy comfort than a VAV system. The ±1^{o}F temperature drift in a VRF system is much smaller than the 6 to 10^{o}F drift experienced in a VAV system (Nye, 2002). Equipment cost for the VRF system may be higher, but this is offset by the relatively minimal length and size of ductwork; ductwork is needed only to route ventilation air to each space and to provide supply air from ducted units. As a result, ductwork costs are reduced and plenum space is also conserved. VAV systems require larger, more extensive ductwork because the entire supply air must be routed from the central air handling unit, including ventilation air.

Although VRF has several very useful characteristics, VAV systems still have applications. One main advantage of VAV systems is that refrigerant piping is not routed to all spaces. This gives VAV an advantage over VRF in certain applications, particularly healthcare facilities; routing refrigerant piping in patient rooms is less than desirable.

Chapter 6 - Climate Study

Although studies exist indicating that VRF systems have potential for increased efficiencies over other technologies, little information shows how the energy use and operational cost of the system is affected by climate. As a result, a basic energy modeling study was performed using TRACE 700 Version 6.2.0.0 to generate understanding of the possible performance and energy costs in different regions of the United States for a VRF heat pump system. Five locations around the country were give a representation of VRF operation in five different climates. Table 6.1 shows each of these climates and the corresponding city used to represent that climate.

Table 6.1 Climates and Corresponding Locations

Climates and Corresponding Locations		
Climate	**Location**	**ASHRAE 90.1 Climate Zone**
Hot, Dry Summer, Mild Winter	Phoenix, AZ	2B
Mild Year-Round	San Diego, CA	3B
Warm, Humid Year-Round	Orlando, FL	2A
Hot Summer, Cold Winter	Topeka, KS	4A
Warm Summer, Very Cold Winter	Minneapolis, MN	6A

Table 6.2 lists the design outdoor temperatures in the energy model for each location. All values are based on 2001 ASHRAE weather data for the 1% average dry bulb and corresponding mean coincident cooling wet bulb for cooling calculations and the 99.6% average dry bulb for heating.

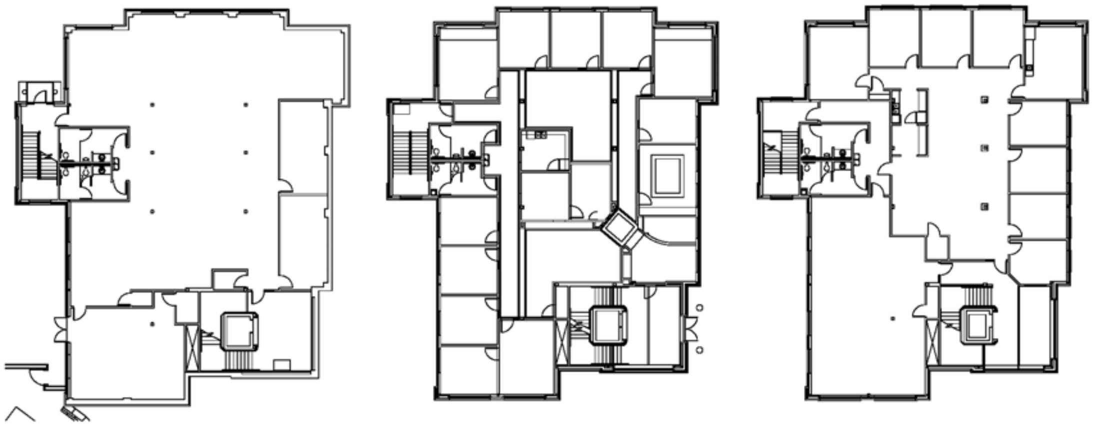
Table 6.2 Design Temperatures for Studied Locations

Design Outdoor Temperatures				
Location	1% Cooling DB/MWB		99.6% Heating DB (^oF)	Heating Degree Days (HDD65)
	DB (^oF)	MWB (^oF)		
Phoenix, AZ	108.1	69.9	37.2	1765
San Diego, CA	81.5	67.7	44.8	1458
Orlando, FL	92.5	76.3	37.4	766
Topeka, KS	93.6	75.6	-1.6	5182
Minneapolis, MN	87.8	71.8	-14.9	8382

Note: Values taken from 2001 ASHRAE Handbook - Fundamentals.

A two story, 16,707 ft^2, office building with a walk-out basement was used for the study, with a total of 58 zones. Figure 6.1 shows the floorplan for this structure. An enlarged and more detailed floor plan can be found in Appendix E.

Figure 6.1 Building Floorplan



The envelope construction and room areas of the building were modeled in TRACE 700, while the inputs used for each component can be referenced in Appendix F. Ventilation rates for each space based on people and area were assigned per minimums of ASHRAE 62.1-2004. Lighting loads were set at 1 watt/square foot with 80% load to space. Next, various miscellaneous loads were also added; with all values taken from commonly specified manufacturer cutsheets. The number of people occupying the space was based on the furniture plan for the building. The interior occupancy schedule, including people, lights, and miscellaneous loads, was selected from the TRACE 700 library as “Low-Rise Office.” This allows the interior loads to contribute to heating of the space and reduce heating energy consumption during building occupancy. Example inputs for Topeka, Kansas are in Appendix F. TRACE 700 outputs for this study were based on these input values and a 20 year life cycle; again, outputs for Topeka, Kansas have also been provided as part of Appendix F.

Several parameters were set to model the VRF system. For example, each room was defined as a separate zone. Next, the default VRF model titled “VRF Heat Pump” was utilized, leaving all default values assigned by TRACE except for the minimum outdoor heating operating temperature; this value was changed from 40°F to 10°F to match common manufacturer minimum temperatures. To supplement the system during heating mode, electric resistance heating was added as backup to handle any load the VRF system could not. Additionally, to remove any potential problems with latent heat from ventilation air, an energy recovery wheel was included in the system to transfer heat to or from the building exhaust/relief air to temper the ventilation air. This unit was modeled as tying into each indoor unit requiring ventilation air; no cooling or heating coils were utilized. All parameters, except for the total capacity of the VRF system, remained the same for all locations to provide the most accurate comparison of systems. TRACE automatically sized the system based on the loads of the building.

For each location, energy rates were defined based on the 2010 electric rates per state taken from the 2010 U.S. Energy Information Administration (EIA) website. These values were used in order to eliminate discrepancies in charges among power companies within each region. Table 6.3 below shows the electricity consumption charges used in the study.

Table 6.3 Electricity Rates per Location.

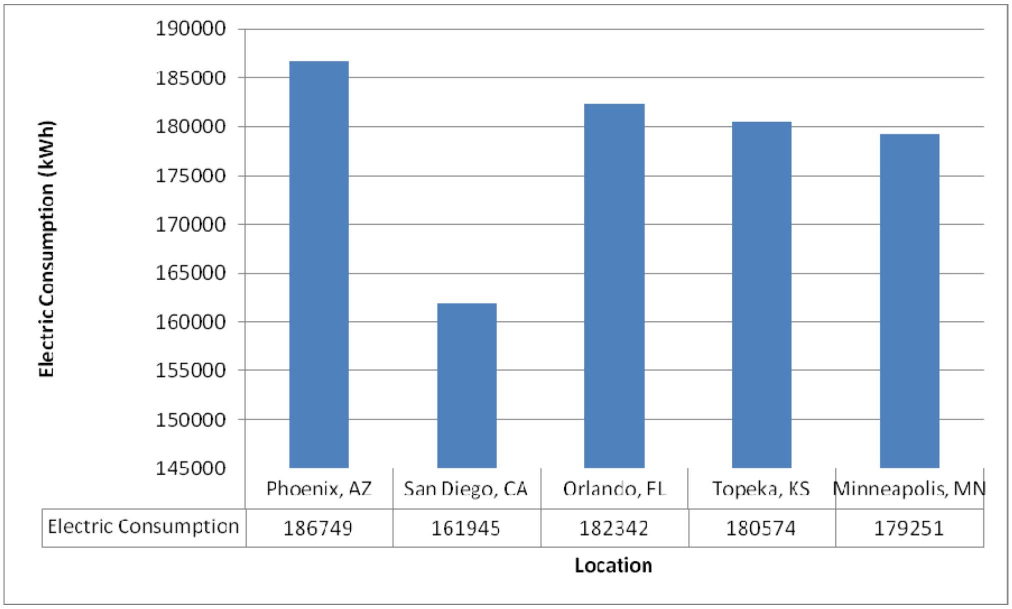
Electricity Rates per Location	
Location	**Charge (Cents/kWh)**
Phoenix, AZ	10.02
San Diego, CA	14.98
Orlando, FL	9.80
Topeka, KS	8.40
Minneapolis, MN	8.80

Note: All values based on U.S. EIA 2010 Average Rates.

After TRACE was run, cooling and heating loads, energy usage, and economic cost results were compared for each location. Figures 6.2 through 6.5 illustrate the loads for the building at each location and the comparison in energy usage at each site.

Figure 6.2 Building Peak Cooling and Heating Loads.

Figure 6.3 Total Yearly Building Electric Consumption.



Note: These total values include lighting and miscellaneous load consumption.

Figure 6.4 Total Cooling Equipment Electric Consumption.

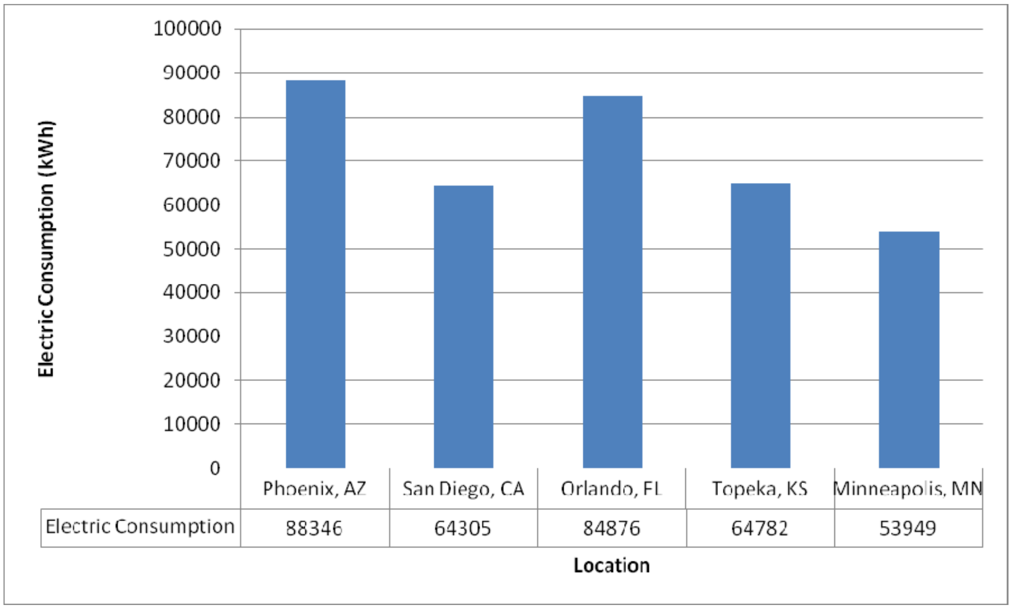
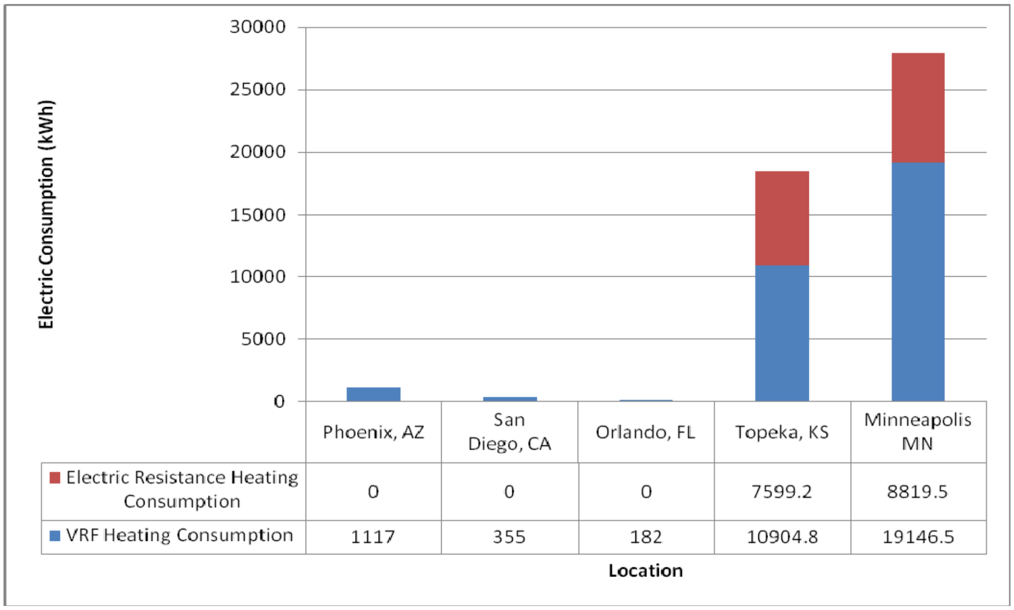


Figure 6.5 Total Heating Equipment Electric Consumption.

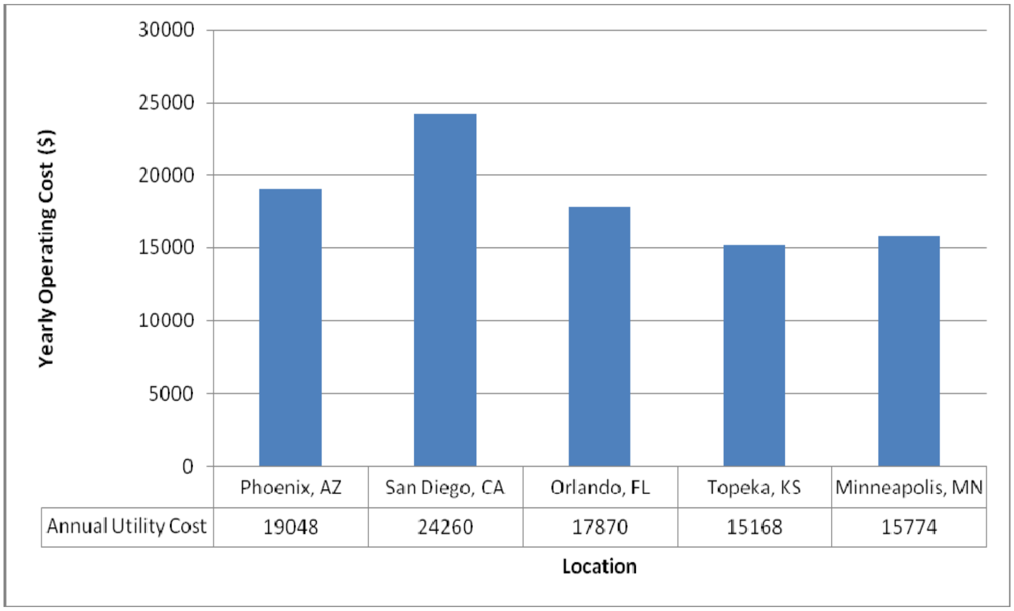


As the graphs show, higher summer design temperatures resulted in higher cooling electric consumption while lower winter design temperatures resulted in higher heating electric consumption in all cases but Orlando. This is because TRACE accounts for year-long weather data when performing an energy calculation. As a result, one location may have a lower winter design temperature than another but require less energy for heating over the course of a year; Orlando, which had a lower winter dry bulb temperature than San Diego, required less annual heating energy because Orlando has fewer heating degree days than San Diego. Also apparent is the correlation between electric resistance heating use and design temperature; Topeka and Minneapolis, the only two locations with winter design temperatures below 10°F, were the only two locations that required the use of electric resistance heat.

The fact that the total peak load did not always result in higher annual energy consumption can be confusing. For example, Minneapolis, Minnesota has a total cooling and heating peak load of 651 MBh with an annual electric consumption of 179,251 kWh; Phoenix, Arizona has a total load of 517 MBh and an annual electric consumption of 186,749 kWh. This is caused by the VRF system in Minneapolis spending more time operating at a lower speed than the VRF system in Phoenix.

Even though the building had significantly higher electric consumption in some locations, this did not directly relate to annual operating costs, demonstrating how greatly economics can differ for building owners in different areas of the country. Figure 6.6 below illustrates the operational cost comparison of the locations.

Figure 6.6 Total Yearly Operating Costs.



Clearly, San Diego, California had the highest yearly operating cost. This location had the most moderate design temperatures, the smallest peak, and the lowest annual energy consumption. However, because of the significantly higher electricity rate in this city, the annual expense to operate the building was much higher than in any other location.

This study yields several important results. First, VRF might not be the most energy efficient technology in specific climates; for example, VRF paired with an ERV consumed a significant amount of electricity in Phoenix. Because of the high number of days requiring cooling and low wet bulb temperature at this location, a designer may wish to install an evaporatively cooled system for hot, dry climates to potentially reduce the electric consumption of the building. However, VRF does offer the advantage of reducing potable water consumption, which must be used in evaporatively cooled systems and is a valuable resource in dry climates. Second, building peak load is not directly related to annual building consumption; because the system can effectively operate at part load conditions, energy consumption can be surprisingly small for a building with a high peak load. Third, a supplemental heating system must be considered in climates with design heating temperatures below VRF manufacturer minimum operating temperatures. These systems can result in significant yearly energy consumption. Although a heat pump system was used in the study, heat recovery VRF systems can still be used in the case supplemental heating is needed; however, the heat recovery aspect of the system will be lost when the supplemental heating is operating. Fourth, energy consumption, the primary concern of this study, is not directly related to operating costs. Operating costs vary based on regional utility rates, making a system with high electric consumption potentially economical to operate. Finally, this study illustrates the all-electric operation of VRF. In northern areas where natural gas is a more affordable source of energy, designers should compare VRF to systems using natural gas heating. For example, the operating costs in Minneapolis, Minnesota may be reduced by installing a heating and cooling system utilizing natural gas heating; such a system would need to be modeled year round to ensure that cooling costs did not offset any savings in heating costs.

Chapter 7 - Recommendations and Conclusion

Air-to-air VRF is a strong competitor with more conventional systems and should be considered by designers. This chapter provides several important considerations and recommendations based on research presented in this paper as well as an energy modeling study specific to climate and energy use. The reader should realize that much of the research included system comparison studies that, for various reasons, could be slightly skewed. More independent research is needed to provide more accurate and credible results that support the use of VRF.

Designers should consider VRF as a viable option for providing efficient means of space conditioning. The efficiency of this system is best in low ventilation air situations, making it particularly useful in commercial offices and hotels. It can also be utilized in schools, hospitals, and many other buildings with high ventilation requirements, although some of the efficiency advantages could be lost if a secondary ventilation system is also necessary.

VRF has a very high refrigerant charge, which may limit its use in some applications. Routing such high amounts of refrigerant through some buildings, particularly hospitals, daycares, and penitentiaries, should likely be avoided. Although designers could reduce risks associated with such a high refrigerant charge, many designers, contractors, and building owners are rightfully inclined to install hydronic or forced air systems in these situations to limit risks to the health of occupants.

In addition to considering efficiency and building occupancy type, considering a VRF system requires a full life cycle cost analysis. In some areas, providing an all-electric system may be economically unwise; the cost of electricity for heating could be much higher than that of natural gas. A life cycle cost analysis will help determine if the effect of this utility cost is negated by lower energy consumption and by system maintenance costs.

The design process for VRF is relatively straightforward. First, each room can be its own zone; however, grouping rooms together into one zone based on exposure and occupancy is also possible. Second, indoor units must be selected based on the peak load in each space they serve; outdoor units must be selected based on building block load. Third, working with manufacturers is essential to producing a functioning system; many manufacturers offer software packages to aid designers with piping sizing and outdoor unit sizing. Also, requirements vary based on manufacturer, so choosing a manufacturer early in the design phase is recommended.

More comparisons between VRF and other systems need to be made before exact efficiency advantages can be determined. Several of the comparative studies presented in this paper are manufacturer studies; more independent studies would remove any potential bias that VRF manufacturers have favoring this technology. Additionally, other studies presented in this paper compared new VRF systems to systems utilizing existing equipment. Having studies that compare new VRF systems to newly installed conventional systems will provide better comparisons of actual differences among systems.

Based on the energy modeling study presented in this paper, VRF systems can be used in a wide range of climates when coupled with an energy recovery ventilator; however, hot, dry climates may create conditions where energy consumption is higher for VRF than for other systems. Nonetheless, VRF can still be used in this region if it proves to be more economical to operate than alternate systems. More in-depth studies illustrating the energy consumption and economic costs in different climates should be undertaken, including comparing systems in different climates based on the way they would most likely be installed. For example, the comparison done in this paper involved all systems utilizing an ERV to precondition outside air as needed. A designer in San Diego, California might not see the need to actually specify this unit, while a designer in Orlando, Florida might use one to handle latent load; however, it was beyond the scope of this paper to make this complicated of a comparison. In a study taking this into account, areas where ERV's were unnecessary might become even better candidates for VRF systems. Eliminating the electric consumption of the ERV could potentially decrease overall system consumption; however, this would not be the case in all areas because the outdoor unit compressor would have to operate at higher speeds.

Finally, water-to-air VRF systems should be further investigated. This paper focuses solely on installation and efficiencies of air-to-air systems. An in depth study of water-to-air systems would offer a better understanding of the advantages of these systems, not only in comparison to conventional systems, but also in comparison to air-to-air VRF systems.

In conclusion, this paper serves to provide designers with a basic insight into the operation, efficiency, selection, and design of air-to-air VRF as the technology exists today; a study also illustrates the impact climate and utility rates can have on the system. In appropriate applications, VRF has the ability to reduce the overall energy usage of commercial buildings in the United States. This system has been successfully used in other countries for the better part of 30 years. With what appears to be higher efficiencies than conventional systems, VRF systems will likely be useful in the U.S. Within the commercial building sector, the system offers most advantages when used in office buildings and hotels because these occupancies require lower ventilation air rates; however, this does not limit it from being used in other types of construction. Air-to-air VRF is a promising technology, and designers should strive to stay abreast of new developments in the field over the coming years.

Bibliography

Afify, R. (2008). Designing VRF systems. *ASHRAE Journal, 50*(6), 52-55.

Amarnath, A. (2008). Variable refrigerant flow: Demonstration of efficient space conditioning technology using variable speed drives. Electric Power Research Institute.

Amarnath, A. & Blatt, M. (2008). Variable refrigerant flow: How, why, and how. *Engineered systems, 25*(2), 54-57.

American Society of Heating, Refrigeration, and Air-Conditioning Engineers. (2001). *2001 - ASHRAE Handbook - Fundamentals.* Atlanta, GA.

American Society of Heating, Refrigeration, and Air-Conditioning Engineers. (2004). *ASHRAE Standard 15 - 2007 – Safety standard for refrigeration systems.* Atlanta, GA.

American Society of Heating, Refrigeration, and Air-Conditioning Engineers. (2007). *ASHRAE Standard 34 - 2007 – Designation and safety classification of refrigerants.* Atlanta, GA.

American Society of Heating, Refrigeration, and Air-Conditioning Engineers. (2007). *ASHRAE Standard 90.1 - 2007 – Energy standard for buildings except low-rise residential buildings.* Atlanta, GA.

American Society of Heating, Refrigeration, and Air-Conditioning Engineers. (2009). *ASHRAE Standard 189.1 - 2009 – Standard for the design of high performance green buildings.* Atlanta, GA.

Bergman, J. (2008). Radiant and VRV pressing U.S. industry. *Air Conditioning, Heating & Refrigeration News, 233*(2), 16.

California Energy Commission. (2008). *2008 Building energy efficiency standards nonresidential compliance manual.* Sacramento, CA.

Cendon, S.F. (2009). New and cool: variable refrigerant flow system: superior control and efficiency are bringing VRF systems to America. *AIArchitect, 16*, 16-21.

Center for Sustainable Systems, University of Michigan. (2009). *Commercial buildings factsheet.* Pub. No. CSS05-05.

Daikin AC. www.daikinac.com.

Dickens, K. V. (2003). Prepared for takeoff. *Engineered Systems, 20*(5), 34.

Goetzler, W. (2007). Variable refrigerant flow systems. *ASHRAE Journal*, *49*, 24-31.

Inman, J. (2007). Going with the (variable) flow. *Engineered Systems, 24*(9), 43-48.

Ivanovich, M. (2010). VRF foundation laid for growth: Codes, standards, and modeling software developments for variable refrigerant flow technology makes life easier for engineers. Retrieved from http://www.csemag.com/industry-news/industry-news/single-article/vrf-foundation-laid-for-growth/374bf2cfbf.html.

Mitsubishi Electric. http://www.mitsubishipro.com/media/226453/cmcatalog.pdf.

Nye, H. (2002). Digital variable multi A/C technology passes test. Retrieved from http://www.achrnews.com/Articles/Feature_Article/d2a18e4ae5a5a010VgnVCM100000f932a8c0.

Obella, C. C. (2009). Electronic controls for VRF systems. Retrieved from http://www.refrige.com/index2.php?option=com_content&do_pdf=1&id=4783.

Roth, K. W., Westphalen, D., Dieckmann, J., Hamilton, S.D., & Goetzler, W. (2002). Energy consumption characteristics of commercial building HVAC systems volume III: Energy savings potential. 4-100 – 4-104.

Ryan, William. (2008). Electric system in a crisis: More power plants or a lower-cost, greener solution? Retrieved from http://www.gasairconditioning.org/ PDF_files_new_site/Electric_System_in_Crisis.pdf.

Siddens, Scott. (2007). Variable refrigerant systems make for zones HVAC control. Retrieved from http://old.csemag.com/article/179058-Variable_Refrigerant_Systems_Make_for_Zoned_HVAC_Control.php.html.

Singapore National Climate Change Committee. http://www.nccc.gov.sg/building/ heat_pump.shtm.

Technology spotlight: heating and cooling with VRF systems. (2008). Retrieved from http://www.wapa.gov/es/pubs/esb/2008/apr/apr084.htm.

The role of the suction line accumulator. (2001). *Air Conditioning, Heating, and Refrigeration News*. Retrieved from http://www.achrnews.com/Articles/Feature_ Article/84a8241c4f95a010VgnVCM100000f932a8c0____.

U.S. Energy Information Administration. (2010). *Average retail price of electricity to ultimate customers by end-use sector, by state.* Retrieved from http://www.eia.doe.gov/ electricity/ epm/table5_6_a.html.

United State Green Building Council. (2008). *LEED2009 for new construction and major renovations.* Washington, DC.

Variable refrigerant flow. (n.d.). Retrieved from http://tg8vrf.ashraetcs.org/pdf/6-14-09%20Variable%20Refrigerant%20Flow%20Handbook.doc.

Variable refrigerant flow zoning research profile. (2009). Retrieved from http://www.mnpower.com/powerofone/one_business/powergrant/profiles/VRFZweb.pdf.

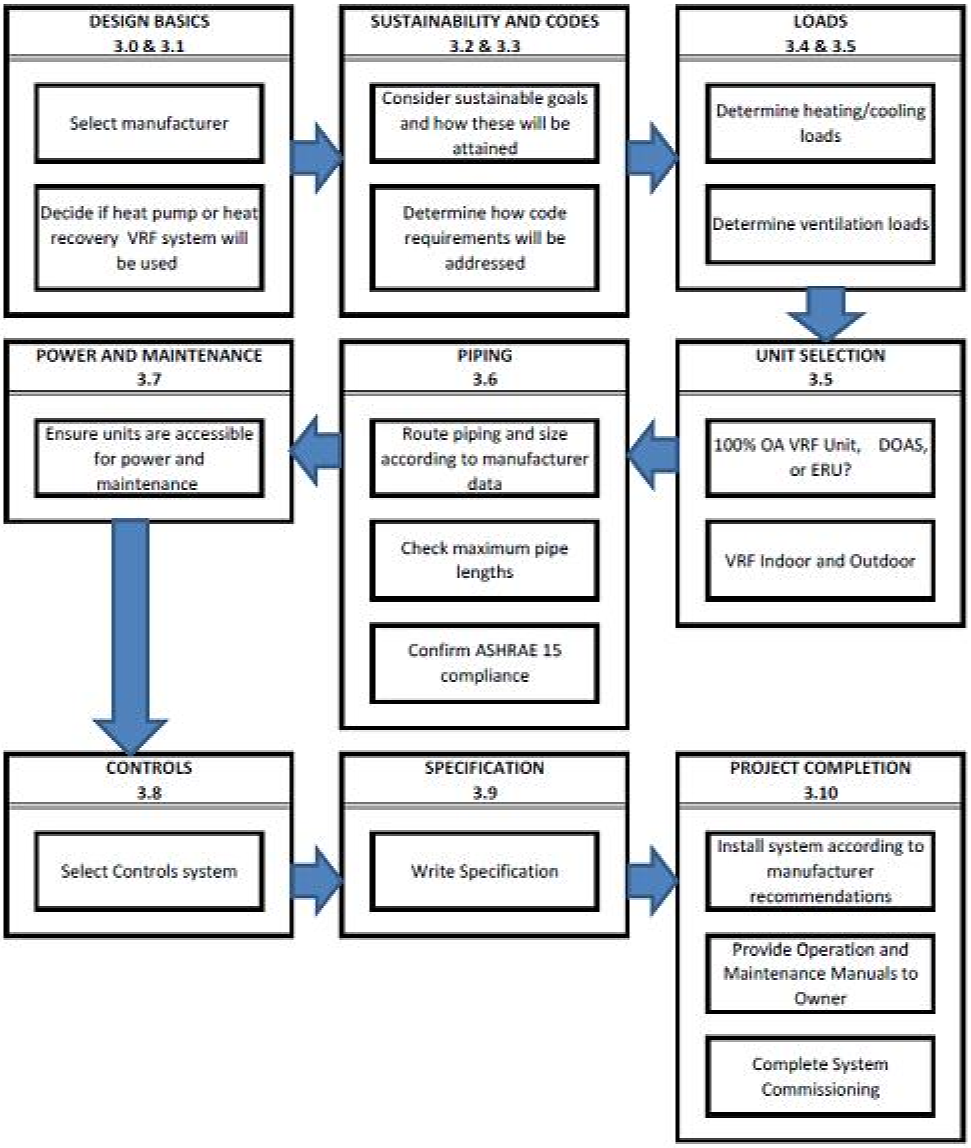
Xia, J., Winandy, E., Georges, B., & Lebrun, J. (2004). Experimental analysis of the performances of variable refrigerant flow systems. *Building Services Engineering Research and Technology, 25*(1), 17-23.

Xiaobing, L. & Tianzhen, H. (2010). Comparison of energy efficiency between variable refrigerant flow systems and ground source heat pump systems. *Energy and Buildings*, *42*(5), 584-58.

Appendix A - Design Flowchart

This section includes the design flowchart developed as part of this paper.

Figure A.1 Design Flowchart



Appendix B - Latent Heat

The psychrometric chart illustrated in Figure B.1 illustrates four points, including outdoor air, return air, mixed air at 50% return air and 50% outdoor air, and supply air. The chart also identifies the cooling curve for this air conditioning process. Sensible and latent energy is extracted along these lines, with (1) mostly sensible cooling occurring first followed by (2) part latent and part sensible cooling. In this example, a coil could reach the design supply air temperature specified for the 50/50 mixed air with a ΔG (change in grains of moisture per pound of dry air) of only 27.5. However, if the percentage of outdoor air were increased to 100%, a total ΔG of 54 would exist; this is twice the latent load presented by the mixed air condition.

Meanwhile, the psychrometric chart illustrated in Figure B.2 illustrates the process of supercooling air to remove latent heat and reheating it to a desirable design supply temperature. In this illustration, a supply air temperature dry bulb temperature of 55°F is needed. The coil cools the mixed air to a temperature of 40°F dry bulb to remove more latent heat than if it cooled the air to only 55°F dry bulb; an additional 20 grains of moisture were removed this way. Because the supply temperature needed is 55°F, an electric coil must provide 15°F of sensible only heating. This is pointed out for clarity in the figure.

Figure B.1 The Cooling Curve.

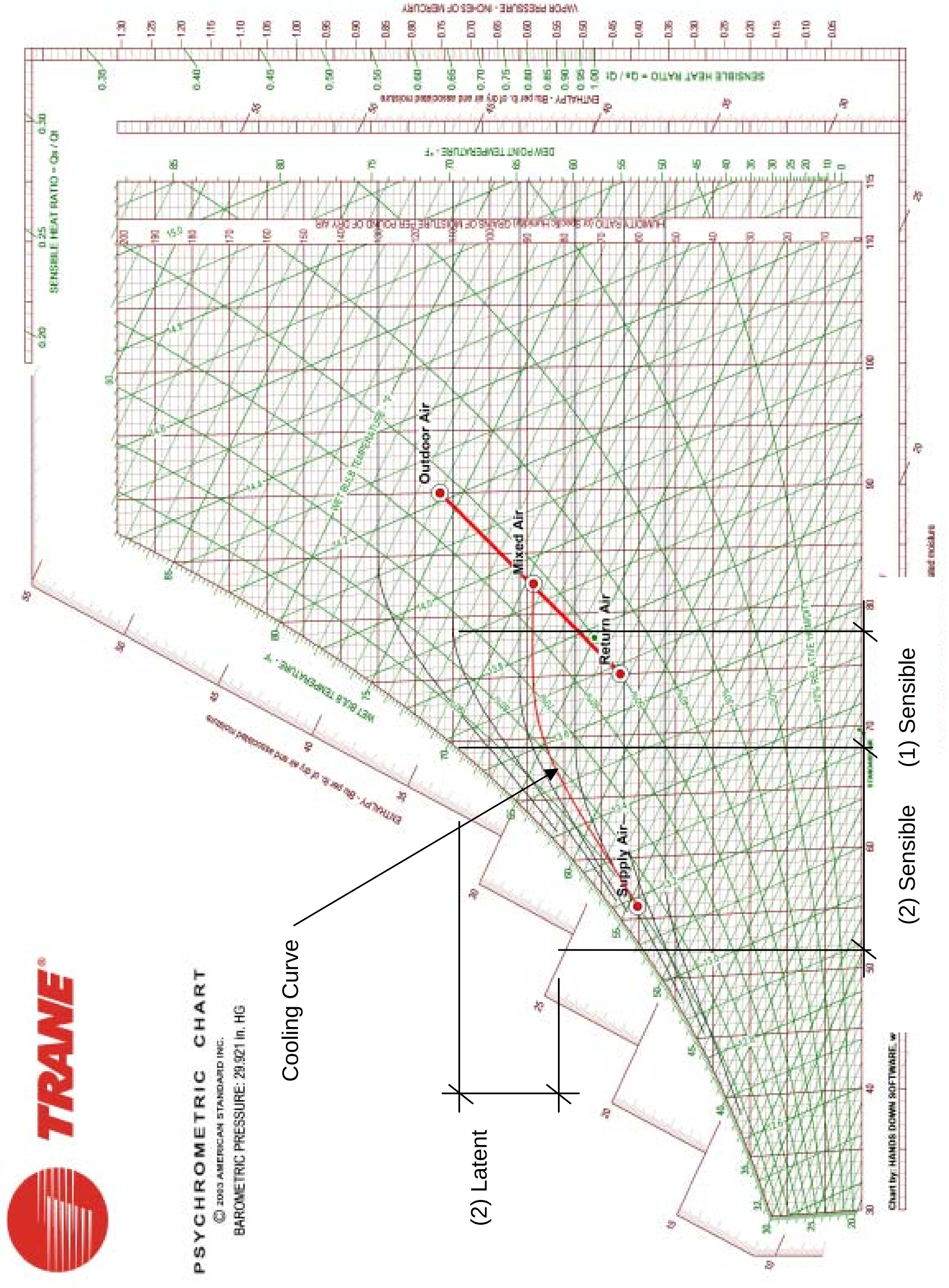
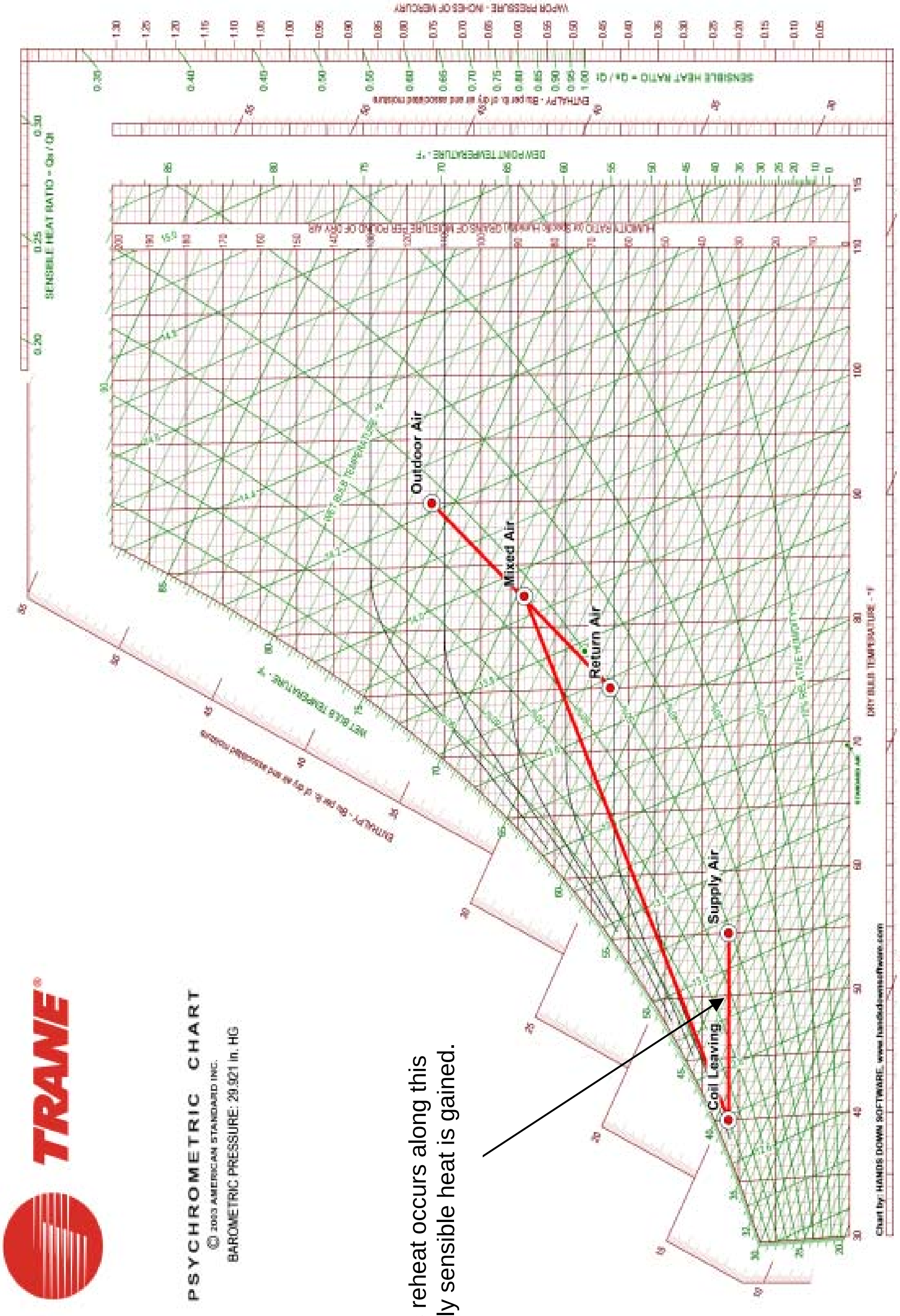


Figure B.2 The Supercooling and Reheat Process.



Electric reheat occurs along this line; only sensible heat is gained.

Appendix C - Example Specification

This section includes an example specification provided courtesy of TME, Inc.

VARIABLE REFRIGERANT VOLUME HEAT PUMP SYSTEM

PART 1 GENERAL

1.1 SYSTEM DESCRIPTION

A. The variable capacity, heat pump air conditioning system shall be a Variable Refrigerant Volume Series (heat or cool model) split system as specified. The system shall consist of multiple evaporators, headers, a 2-pipe refrigeration distribution system using PID control, and VRV outdoor unit. The outdoor unit is a direct expansion (DX), air-cooled heat pump, multi-zone air-conditioning system with variable speed driven compressors using R-410A refrigerant. The outdoor unit may connect an indoor evaporator capacity up to 200% of the outdoor condensing unit capacity. All zones are each capable of operating separately with individual temperature control.

B. The outdoor unit shall be interconnected to indoor unit models and shall range in capacity from 7,500 Btu/h to 48,000 Btu/h in accordance with schedule data for each available indoor unit. The indoor units shall be connected to the outdoor utilizing piping joints and headers.

C. Operation of the system shall permit either cooling or heating of all of the fan coil units. Each fan coil or group of fan coils shall be able to provide set temperature independently via a local remote controller. System shall have a touch screen icon-based centralized controller. Controller shall be capable of time of day scheduling by zone or group, start/stop operation, temperature setting, switching between temperature control modes and operation status. Controller shall be capable of controlling/monitoring up to 64 indoor units. Controller shall be equipped with internal battery back up to run the clock during blackouts.

1.2 RELATED WORK SPECIFIED IN OTHER SECTIONS

A. Insulation of HVAC equipment; Section 23 07 13.

B. Air distribution equipment; Section 23 31 00.

C. Control work required in conjunction with HVAC equipment; Section 23 09 33.

D. Air side balancing; Section 23 05 93.

E. Motor starters, disconnects, power wiring of HVAC equipment; Division 26.

1.3 QUALITY ASSURANCE

A. AFBMA Standards: Load Ratings and Fatigue Life for Ball Bearings. Bearings must have an L10 life of not less than 200,000 hours for air handling units and utility vent sets, and an L50 life of not less than 200,000 hours for in-line and rooftop exhaust fans.

B. AGA Standards: Boilers shall be tested, listed and certified in accordance with the requirements of the American Gas Association Laboratories and ANSI Standard Z31.13.

C. AMCA Standards: Comply with Air Movement and Control Association (AMCA) standards as applicable to testing and rating fans (AMCA 300), air moving devices (AMCA 301) and testing of louvers, dampers and shutters.

D. ANSI/ASHRAE Standards: Comply with ANSI/ASHRAE Standard 15 - Safety Code for Mechanical Refrigeration. Provide central station packaged air handling units which comply with ANSI Standard 430.

E. ARI Certification: Provide central station packaged air handling units which comply with Air-Conditioning and Refrigeration Institute (ARI) Standard 410. Provide chillers which comply with ARI Standard 550.

F. ARI/ASHRAE Standards: Heat Recovery Wheels shall be tested in accordance with the requirements of ASHRAE Standard 84/ARI Standard 1060.

G. ASHRAE Standards: Comply with ASHRAE Standard 52-76 as applicable to air filter efficiencies.

H. ASME Code Compliance: Construct boilers in accordance with ASME Boiler and Pressure Vessel Code, "Heating Boilers", Section IV. Construct heat transfer units in accordance with ASME Boiler and Pressure Vessel Code, Section VIII "Unfired Pressure Vessels", bearing the National Board stamp.

I. ASTM (& UL) Compliance: Thermal insulation for equipment shall have the following maximum UL Fire Resistance Ratings, per ASTM Standard E84, unless otherwise specified:

1. Flame Spread: 25
2. Fuel Contributed: 50
3. Smoke Developed: 0

J. NEMA (& IEEE) Compliance: Provide electric motors which meet the scheduled full load efficiency, as indicated on the drawings, per NEMA Standard MG1-12.53a, based on dynamometer testing per IEEE 12 Method B.

K. NEMA (& UL) Compliance: Provide electric motors and products which have been listed and labeled by Underwriters Laboratories (UL) and comply with NEMA standards.

L. NFPA Compliance: Comply with applicable provisions of ANSI/NFPA 70 "National Electric Code", pertaining to construction and installation of electrically operated components of packaged air handling units. Install fuel gas piping in accordance with NFPA 54 "National Fuel Gas Code".

M. TEMA Standard Compliance: Construct and install heat transfer units in accordance with “Standards of the Tubular Exchanger Manufacturers Association”.

N. UL Compliance: Comply with UL Standard 900 as applicable to listing of air filters. Comply with UL 984 - Safety Standards for Hermetic Motor Compressors.

O. All electronic equipment shall conform to the requirements of FCC Regulations, Part 15, Section 15, governing radio frequency and electromagnetic interference and shall be so labeled.

P. The units shall be listed by Electrical Laboratories (ETL) and bear the c etl label.

Q. All wiring shall be in accordance with the National Electric Code (NEC).

R. The system will be produced in an ISO 9001 and ISO 14001 facility, which are standards set by the International Standard Organization (ISO). The system shall be factory tested for safety and function.

S. The outdoor unit will be factory charged with R-410A.

1.4 SUBMITTALS: Comply with Sections 01 33 00 and 23 00 00.

A. General: Submit manufacturer's technical product data, including rated capacities of selected model clearly indicated (including fan and pump curves, tabular data etc.), weights and loadings (shipping, installed, and operating where applicable), construction materials furnished (including gages and finishes), required clearances, location and size of field connections, motor electrical characteristics and accessories. Clearly differentiate between portions of wiring that are factory installed and portions to be field installed. Indicate equipment, piping and connections, valves, strainers, piping accessories required for a complete system. Submit suggested structural steel support, including dimensions, sizes and locations for mounting bolt holes. Include weight distribution drawings showing point loadings.

B. Provide data as described above for the following equipment:

1. Variable Refrigerant Volume Heat Pump System and components

C. Any substitutions of specified HVAC equipment to be installed in a mechanical room will require revised plan and elevation drawings of each mechanical room where substitutions occur, in a scale not less than 1/4” = 1' 0”. Drawings shall indicate equipment sizes, clearances and elevations, of ALL equipment (substituted and non-substituted). Failure to furnish revised mechanical room drawings with the substituted equipment submittal will result in rejection of the submission and no additional contract time will be allowed for delay of this cause.

D. Shop Drawings: Submit manufacturer's assembly-type shop drawings indicating dimensions, weight loadings, required clearances, methods of assembly of components, construction details, and field connection details.

E. Wiring Diagrams: Submit ladder-type wiring diagrams for electrically operated accessories. Clearly differentiate between portions of wiring that are factory-installed and portions to be field-installed.

F. Electric Motors: All equipment specified with premium efficiency motors shall include data sheets, with equipment submittals, on motors. Data sheets shall include manufacture/model number of motor including statement motor complies with NEMA standard MG1 part 31.40.4.2.

G. Maintenance Data: Submit maintenance data, including lubrication instructions, filter replacement, motor and drive replacement, and spare parts lists for each equipment item, including "trouble - shooting" maintenance guide. Include this data and product data in maintenance manual; in accordance with requirements of Division 1

1.5 DELIVERY, STORAGE AND HANDLING

A. Unit shall be stored and handled according to the manufacturer's recommendations.

B. Deliver HVAC equipment with factory-installed shipping skids and lifting lugs; pack components in factory-fabricated protective containers. Factory assemble entire unit, where practical. For shipping, disassemble into as large as practical sub-assemblies so that minimum amount of field work is required for re-assembly.

C. Store HVAC equipment in clean dry place and protect from weather and construction traffic. Handle HVAC equipment carefully to avoid damage to components, enclosures, and finish. Leave factory shipping covers in place until installation. Do not install damaged components; replace and return damaged components to equipment manufacturer.

D. Comply with manufacturer's installation instructions for rigging, unloading and transporting units.

PART 2 PRODUCTS

2.1 ACCEPTABLE MANUFACTURERS

A. Daikin

B. Mitsubishi

C. Sanyo

2.2 CONCEALED CEILING DUCTED UNIT

A. General: Indoor unit model shall be a built-in ceiling concealed fan coil unit, operable with R-410A refrigerant, equipped with an electronic expansion valve, for installation into the ceiling cavity. The unit shall be constructed of a galvanized steel casing. It shall be available from 9,500 Btu/h to 48,000 Btu/h capacities to be connected to outdoor unit model heat pump or heat recovery model. It shall be a horizontal discharge air with horizontal return air or bottom return air configuration. All models feature a low height (11-7/8") making them applicable to ceiling pockets that tend to be shallow. Computerized PID control shall be used to control superheat to deliver a comfortable room temperature condition. The unit shall be equipped with a programmed drying mechanism that dehumidifies while inhibiting changes in room temperature when used with remote control. Included as standard equipment, a long-life filter that is mold resistant and a condensate drain pan and drain pump kit that pumps to 9-13/16" from the drain pipe opening. The indoor units sound pressure shall range from 35 dB(A) to 43 dB(A) at low speed 5 feet below the suction grille.

B. Performance: Each unit's performance is based on scheduled operating conditions.

C. Indoor Unit:

1. The indoor unit shall be completely factory assembled and tested. Included in the unit is factory wiring, piping, electronic proportional expansion valve, control circuit board, fan motor thermal protector, flare connections, condensate drain pan, condensate drain pump, self-diagnostics, auto-restart function, 3-minute fused time delay, and test run switch. The unit shall have an adjustable external static pressure switch.

2. Indoor unit and refrigerant pipes will be charged with dehydrated air prior to shipment from the factory.

3. Both refrigerant lines shall be insulated from the outdoor unit.

4. Return air shall be through a net mold resistant filter.

5. The indoor units shall be equipped with a condensate pan and condensate pump. The condensate pump provides up to 9-13/16" of lift.

6. The indoor units shall be equipped with a return air thermistor.

7. The indoor unit will be separately powered with 208~230V/1-phase/60Hz.

8. The voltage range will be 253 volts maximum and 187 volts minimum.

9. Switch box shall be reached from the side or bottom for ease of service and maintenance.

D. Unit Cabinet:

  1. The cabinet shall be located into the ceiling and ducted to the supply and return openings.

  2. The cabinet shall be constructed with sound absorbing foamed polystyrene and polyethylene insulation.

  3. Optional high efficiency air filters are available for each model unit.

E. Fan:

  1. The fan shall be direct-drive Sirocco type fan, statically and dynamically balanced impeller with high and low fan speeds available.

  2. The fan motor shall operate on 208/230 volts, 1 phase, 60 hertz with a motor output range from 0.07 to 0.30 HP.

  3. The airflow rate shall be available in high and low settings.

  4. The fan motor shall be thermally protected.

F. Filter: The return air shall be filtered by means of a washable long-life filter with mildew proof resin.

G. Coil:

  1. Coils shall be of the direct expansion type constructed from copper tubes expanded into aluminum fins to form a mechanical bond.

  2. The coil shall be of a waffle louver fin and high heat exchange, rifled bore tube design to ensure highly efficient performance.

  3. The coil shall be a 3-row cross fin copper evaporator coil with 14 FPI design completely factory tested.

  4. The refrigerant connections shall be flare connections and the condensate will be 1 -1/4 inch outside diameter PVC.

  5. A condensate pan shall be located under the coil.

  6. A condensate pump with a 9-13/16” lift shall be located below the coil in the condensate pan with a built in safety alarm.

  7. A thermistor will be located on the liquid and gas line.

H. Electrical:

1. A separate power supply will be required of 208/230 volts, 1 phase, 60 hertz. The acceptable voltage range shall be 187 to 253 volts.

2. Transmission (control) wiring between the indoor and outdoor unit shall be a maximum of 3,280 feet (total 6,560 feet).

3. Transmission (control) wiring between the indoor unit and remote controller shall be a maximum distance of 1,640 feet.

I. Control:

1. The unit shall have controls to perform input functions necessary to operate the system.

2. The unit shall be compatible with interfacing with connection to LonWorks networks or interfacing with connection to BMS system.

J. Accessories Available:

1. Remote "in-room" sensor kit.

   a. The wall mounted, hard wired remote sensor kit is recommended for ceiling-embedded type fan coils, which often result in a difference between set temperature and actual temperature. The sensor for detecting the temperature can be placed away from the indoor unit (branch wiring is included in the kit).

2.3 CONCEALED CEILING DUCTED UNIT (Med. Static)

A. General: Indoor unit shall be a built-in ceiling concealed fan coil unit, operable with refrigerant R-410A, equipped with an electronic expansion valve, for installation into the ceiling cavity. It is constructed of a galvanized steel casing. It shall be available from 30,000 Btu/h to 48,000 Btu/h capacities to be connected to outdoor unit model heat pump or heat recovery model. It shall be a horizontal discharge air with horizontal return air configuration. All models feature a low height (15-3/8") cabinet making them applicable to ceiling pockets that tend to be shallow. Computerized PID control shall be used to control superheat to deliver a comfortable room temperature condition. The unit shall be equipped with a programmed drying mechanism that dehumidifies while inhibiting changes in room temperature when used with remote control. The indoor units sound pressure shall range from 41 dB(A) to 45 dB(A) at low speed measured 5 feet below the ducted unit.

B. Performance: Each unit's performance is based on scheduled operating conditions.

C. Indoor Unit:

1. The indoor unit shall be completely factory assembled and tested. Included in the unit is factory wiring, piping, electronic proportional expansion valve, control circuit board, fan motor thermal protector, flare connections, self-diagnostics, auto-restart function, 3-minute fused time delay, and test run switch. The unit shall have an adjustable external static pressure switch.

2. Indoor unit and refrigerant pipes will be charged with dehydrated air prior to shipment from the factory.

3. Both refrigerant lines shall be insulated from the outdoor unit.

4. The indoor units shall be equipped with a return air thermistor.

5. The indoor unit will be separately powered with 208~230V/1-phase/60Hz.

6. The voltage range will be 253 volts maximum and 187 volts minimum.

D. Unit Cabinet:

1. The cabinet shall be located into the ceiling and ducted to the supply and return openings.

2. The cabinet shall be constructed with sound absorbing foamed polystyrene and polyethylene insulation.

E. Fan:

1. The fan shall be direct-drive Sirocco type fan, statically and dynamically balanced impeller with high and low fan speeds available.

2. The fan motor shall operate on 208/230 volts, 1 phase, 60 hertz with a motor output range 0.21, 0.36 and 0.58 HP respectively.

3. The airflow rate shall be available in high and low settings.

4. The fan motor shall be thermally protected.

F. Coil:

1. Coils shall be of the direct expansion type constructed from copper tubes expanded into aluminum fins to form a mechanical bond.

2. The coil shall be of a waffle louver fin and high heat exchange, rifled bore tube design to ensure highly efficient performance.

3. The coil shall be a 3 row cross fin copper evaporator coil with 13 fpi design completely factory tested.

4. The refrigerant connections shall be flare connections and the condensate will be 1-1/4 inch outside diameter PVC.

5. A thermistor will be located on the liquid and gas line.

G. Electrical:

1. A separate power supply will be required of 208/230 volts, 1 phase, 60 hertz. The acceptable voltage range shall be 187 to 253 volts.
2. Transmission (control) wiring between the indoor and outdoor unit shall be a maximum of 3,280 feet (total 6,560 feet).
3. Transmission (control) wiring between the indoor unit and remote controller shall be a maximum distance of 1,640 feet.

H. Control:

1. The unit shall have controls provided to perform input functions necessary to operate the system.
2. The unit shall be compatible with interfacing with connection to LonWorks networks or interfacing with connection to BMS system.

I. Accessories Available:

1. Remote "in-room" sensor kit KRCS01-1A (recommended).
    a. The wall mounted, hard wired remote sensor kit is recommended for ceiling-embedded type fan coils, which often result in a difference between set temperature and actual temperature. The sensor for detecting the temperature can be placed away from the indoor unit (branch wiring is included in the kit.).
    b. A condensate pump.

2.4 WALL MOUNTED UNIT

A. General: Indoor unit shall be a wall mounted fan coil unit, operable with refrigerant R-410A, equipped with an electronic expansion valve, for installation onto a wall within a conditioned space. This compact design with finished white casing shall be available from 7,500 Btu/h to 24,000 Btu/h capacities. Computerized PID control shall be used to control superheat to deliver a comfortable room temperature condition. The unit shall be equipped with a programmed drying mechanism that dehumidifies while inhibiting changes in room temperature when used with remote control. A mildew-proof, polystyrene air filter and condensate drain pan shall be included as standard equipment. The indoor units sound pressure shall range from 32 dB(A) to 35 dB(A) at low speed measured at 3.3 feet below and from the unit.

B. Performance: Each unit's performance is based on scheduled operating conditions.

C. Indoor Unit:

1. The indoor unit shall be completely factory assembled and tested. Included in the unit is factory wiring, piping, electronic proportional expansion valve, control circuit board, fan motor thermal protector, flare connections, condensate drain pan, self-diagnostics, auto-restart function, 3-minute fused time delay, and test run switch. The unit shall have an auto-swing louver which ensures efficient air distribution, which closes automatically when the unit stops. The remote controller shall be able to set 5 steps of discharge angle. The front grille shall be easily removed for washing. The discharge angle shall automatically set at the same angle as the previous operation upon restart. The drain pipe can be fitted to from either left or right sides.

2. Indoor unit and refrigerant pipes will be charged with dehydrated air prior to shipment from the factory.

3. Both refrigerant lines shall be insulated from the outdoor unit.

4. Return air shall be through a resin net mold resistant filter.

5. The indoor units shall be equipped with a condensate pan.

6. The indoor units shall be equipped with a return air thermistor.

7. The indoor unit will be separately powered with 208~230V/1-phase/60Hz.

8. The voltage range will be 253 volts maximum and 187 volts minimum.

D. Unit Cabinet:

1. The cabinet shall be affixed to a factory supplied wall mounting template and located in the conditioned space.

2. The cabinet shall be constructed with sound absorbing foamed polystyrene and polyethylene insulation.

E. Fan:

1. The fan shall be a direct-drive cross-flow fan, statically and dynamically balanced impeller with high and low fan speeds available.

2. The fan motor shall operate on 208/230 volts, 1 phase, 60 hertz with a motor output range 0.054 to 0.058 HP.

F. The airflow rate shall be available in high and low settings.

G. The fan motor shall be thermally protected.

H. Coil:

1. Coils shall be of the direct expansion type constructed from copper tubes expanded into aluminum fins to form a mechanical bond.

2. The coil shall be of a waffle louver fin and high heat exchange, rifled bore tube design to ensure highly efficient performance.

3. The coil shall be a 2-row cross fin copper evaporator coil with 14 fpi design completely factory tested.

4. The refrigerant connections shall be flare connections and the condensate will be 11/16 inch outside diameter PVC.

5. A thermistor will be located on the liquid and gas line.

6. A condensate pan shall be located in the unit.

I. Electrical:

1. A separate power supply will be required of 208/230 volts, 1 phase, 60 hertz. The acceptable voltage range shall be 187 to 253 volts.

2. Transmission (control) wiring between the indoor and outdoor unit shall be a maximum of 3,280 feet (total 6,560 feet).

3. Transmission (control) wiring between the indoor unit and remote controller shall be a maximum distance of 1,640 feet.

J. Control:

1. The unit shall have controls provided to perform input functions necessary to operate the system.

2. The unit shall be compatible with interfacing with connection to LonWorks networks or interfacing with connection to BMS system.

K. Accessories Available:

1. Remote “in-room” sensor kit KRCS01-1A.

2. A condensate pump.

2.5 CEILING SUSPENDED CASSETTE UNIT

A. General: Daikin indoor unit shall be a ceiling suspended fan coil unit, operable with refrigerant R-410A, equipped with an electronic expansion valve, for installation onto a wall or ceiling within a conditioned space. This compact design with finished white casing shall be available from 12,000 Btu/h to 36,000 Btu/h capacities. Computerized PID control shall be used to control superheat to deliver a comfortable room temperature condition. The unit shall be equipped with a programmed drying mechanism that dehumidifies while inhibiting changes in room temperature when used with remote control. A mildew-proof, polystyrene air filter and condensate drain pan shall be included as standard equipment. The indoor units sound pressure shall range from 32 dB(A) to 38 dB(A) at low speed measured at 3.3 feet below and from the unit.

B. Performance: Each unit's performance is based on nominal operating conditions:

C. Indoor Unit:

1. The Daikin indoor unit shall be completely factory assembled and tested. Included in the unit is factory wiring, piping, electronic proportional expansion valve, control circuit board, fan motor thermal protector, flare connections, condensate drain pan, self-diagnostics, auto-restart function, 3-minute fused time delay, and test run switch. The unit shall have an auto-swing louver which ensures efficient air distribution, which closes automatically when the unit stops. The remote controller shall be able to set 5 steps of discharge angle. The front grille shall be easily removed for washing. The discharge angle shall automatically set at the same angle as the previous operation upon restart. The drain pipe can be fitted to from either left or right sides.
2. Indoor unit and refrigerant pipes will be charged with dehydrated air prior to shipment from the factory.
3. Both refrigerant lines shall be insulated from the outdoor unit.
4. Return air shall be through a resin net mold resistant filter.
5. The indoor units shall be equipped with a condensate pan.
6. The indoor units shall be equipped with a return air thermistor.
7. The indoor unit will be separately powered with 208~230V/1-phase/60Hz.
8. The voltage range will be 253 volts maximum and 187 volts minimum.

D. Unit Cabinet:

1. The cabinet shall be affixed to a factory supplied wall/ceiling hanging brackets and located in the conditioned space.
2. The cabinet shall be constructed with sound absorbing foamed polystyrene and polyethylene insulation.

E. Fan:

1. The fan shall be a direct-drive cross-flow fan, statically and dynamically balanced impeller with high and low fan speeds available.
2. The fan motor shall operate on 208/230 volts, 1 phase, 60 hertz with a motor output range 62W to 130W.
3. The airflow rate shall be available in high and low settings.
4. The fan motor shall be thermally protected.

F. Coil:

1. Coils shall be of the direct expansion type constructed from copper tubes expanded into aluminum fins to form a mechanical bond.

2. The coil shall be of a waffle louver fin and high heat exchange, rifled bore tube design to ensure highly efficient performance.

3. The coil shall be a 2-row cross fin copper evaporator coil with 15 fpi design completely factory tested.

4. The refrigerant connections shall be flare connections and the condensate will be 1 inch outside diameter PVC.

5. A thermistor will be located on the liquid and gas line.

6. A condensate pan shall be located in the unit.

G. Electrical:

1. A separate power supply will be required of 208/230 volts, 1 phase, 60 hertz. The acceptable voltage range shall be 187 to 253 volts.

2. Transmission (control) wiring between the indoor and outdoor unit shall be a maximum of 3,280 feet (total 6,560 feet).

3. Transmission (control) wiring between the indoor unit and remote controller shall be a maximum distance of 1,640 feet.

H. Control:

1. The unit shall have controls provided by Daikin to perform input functions necessary to operate the system.

2. The unit shall be compatible with interfacing with connection to LonWorks networks or interfacing with connection to BMS system. Consult with Daikin prior to applying controls.

2.6 OUTDOOR UNIT

A. General: The outdoor unit is designed specifically for use with VRV series components.

1. The outdoor unit shall be factory assembled and pre-wired with all necessary electronic and refrigerant controls. The refrigeration circuit of the condensing unit shall consist of scroll compressors, motors, fans, condenser coil, electronic expansion valves, solenoid valves, 4-way valve, distribution headers, capillaries, filters, shut off valves, oil separators, service ports and refrigerant regulator. High/low pressure gas line, liquid and suction lines must be individually insulated between the outdoor and indoor units.

2. The outdoor unit can be wired and piped with outdoor unit access from the left, right, rear or bottom.

3. The connection ratio of indoor units to outdoor unit shall be permitted up to 200%.

4. Each outdoor system shall be able to support the connection of up to 41 indoor units dependant on the model of the outdoor unit.

5. The sound pressure level standard shall be that value as listed in the engineering manual for the specified models at 3 feet from the front of the unit. The outdoor unit shall be capable of operating automatically at further reduced noise during night time.

6. The system will automatically restart operation after a power failure and will not cause any settings to be lost, thus eliminating the need for reprogramming.

7. The unit shall incorporate an auto-charging feature and a refrigerant charge check function.

8. The outdoor unit shall be modular in design and should allow for side-by-side installation with minimum spacing.

9. The following safety devices shall be included on the condensing unit; high pressure switch, control circuit fuses, crankcase heaters, fusible plug, high pressure switch, overload relay, inverter overload protector, thermal protectors for compressor and fan motors, over current protection for the inverter and anti-recycling timers.

10. To ensure the liquid refrigerant does not flash when supplying to the various fan coil units, the circuit shall be provided with a sub-cooling feature.

11. Oil recovery cycle shall be automatic occurring 2 hours after start of operation and then every 8 hours of operation.

12. The outdoor unit shall be capable of heating operation at 0 F dry bulb ambient temperature without additional low ambient controls.

13. The system shall continue to provide heat to the indoor units while in the defrost mode.

B. Unit Cabinet: The outdoor unit shall be completely weatherproof and corrosion resistant. The unit shall be constructed from rust-proofed mild steel panels coated with a baked enamel finish.

C. Fan:

1. The condensing unit shall consist of one or more propeller type, direct-drive 750 W fan motors that have multiple speed operation via a DC (digitally commutating) inverter.

2. The condensing unit fan motor shall have multiple speed operation of the DC (digitally commutating) inverter type, and be of high external static pressure and shall be factory set as standard at 0.12 in. WG. A field setting switch to a maximum 0.32 in. WG pressure is available to accommodate field applied duct for indoor mounting of condensing units.

3. The fan shall be a vertical discharge configuration with a nominal airflow maximum range of 6,530 CFM to 14,120 CFM dependant on model specified.

4. Nominal sound pressure levels shall be a maximum of 60 dB(A).

5. The fan motor shall have inherent protection and permanently lubricated bearings and be mounted.

6. The fan motor shall be provided with a fan guard to prevent contact with moving parts.

7. Night setback control of the fan motor for low noise operation by way of automatically limiting the maximum speed shall be a standard feature. Operation sound level shall be selectable from 3 steps as shown below.

Operation Sound (dB)	Night Mode Sound Pressure Level (dB)
Step 1 Max.	55
Step 2 Max.	50
Step 4 Max.	45

D. Condenser Coil:

1. The condenser coil shall be manufactured from copper tubes expanded into aluminum fins to form a mechanical bond.

2. The heat exchanger coil shall be of a waffle louver fin and rifled bore tube design to ensure high efficiency performance.

3. The heat exchanger on the condensing units shall be manufactured from Hi-X seamless copper tube with N-shape internal grooves mechanically bonded on to aluminum fins to an e-Pass Design.

4. The fins are to be covered with an anti-corrosion acrylic resin and hydrophilic film type E1.

5. The pipe plates shall be treated with powdered polyester resin for corrosion prevention. The thickness of the coating must be between 2.0 to 3.0 microns.

E. Compressor:

1. The inverter scroll compressors shall be variable speed (PAM inverter) controlled which is capable of changing the speed to follow the variations in total cooling and heating load as determined by the suction gas pressure as measured in the condensing unit. In addition, samplings of evaporator and condenser temperatures shall be made so that the high/low pressures detected are read every 20 seconds and calculated. With each reading, the compressor capacity (INV frequency or STD ON/OFF) shall be controlled to eliminate deviation from target value.
2. The inverter driven compressor in each condensing unit shall be of highly efficient reluctance DC (digitally commutating), hermetically sealed scroll "G-type" with a maximum speed of 7,980 rpm.
3. Neodymium magnets shall be adopted in the rotor construction to yield a higher torque and efficiency in the compressor instead of the normal ferrite magnet type. At complete stop of the compressor, the neodymium magnets will position the rotor into the optimum position for a low torque start.
4. The capacity control range shall be as low as 6% to 100%.
5. Each non-inverter compressor shall also be of the hermetically sealed scroll type.
6. Each compressor shall be equipped with a crankcase heater, high pressure safety switch, and internal thermal overload protector.
7. Oil separators shall be standard with the equipment together with an intelligent oil management system.
8. The compressor shall be spring mounted to avoid the transmission of vibration.
9. Units sized 8-10 ton shall contain a minimum of 2 compressors, 12-20 ton units shall contain a minimum of 4 compressors. In the event of compressor failure the remaining compressors shall continue to operate and provide heating or cooling as required at a proportionally reduced capacity. The microprocessor and associated controls shall be designed to specifically address this condition.
10. In the case of multiple condenser modules, conjoined operation hours of the compressors shall be balanced by means of the Duty Cycling Function, ensuring sequential starting of each module at each start/stop cycle, completion of oil return, completion of defrost or every 8 hours.

F. Electrical:

1. The power supply to the outdoor unit shall be 240 volts, 3 phase, 60 hertz +/- 10%.

    a. The control voltage between the indoor and outdoor unit shall be 16VDC non-shielded, stranded 2 conductor cable.

    b. The control wiring shall be a 2-wire multiplex transmission system, making it possible to connect multiple indoor units to one outdoor unit with one 2-cable wire, thus simplifying the wiring operation.

    c. The control wiring lengths shall be as shown below.

	Outdoor to Indoor Unit	Outdoor to Central Controller	Indoor Unit to Remote Control
Control Wiring Length	6,665 ft	3,330 ft	1,665 ft
Wire Type	16 AWG, 2 wire, non-polarity, non-shielded, stranded		

PART 3 INSTALLATION

3.1 WARRANTY: The units shall have a manufacturer's warranty for a period of 1 year from date of installation. The units shall have a limited labor warranty for a period of 1 year from date of installation. The compressors shall have a warranty of 6 years from date of installation. During the stated period, should any part fail due to defects in material and workmanship, it shall be repaired or replaced at the discretion of according to standard terms and conditions. All warranty service work shall be preformed by a factory trained service professional.

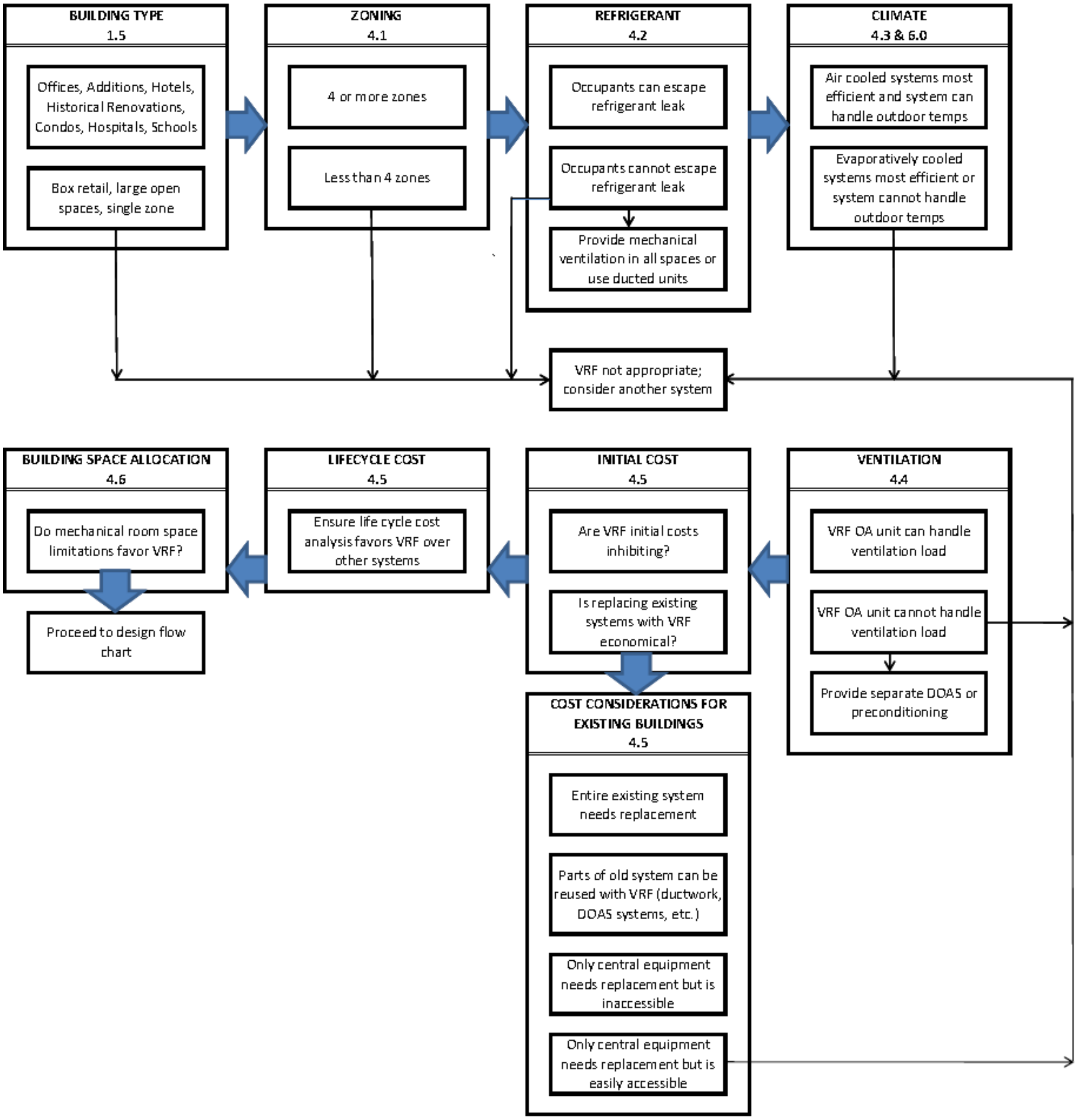
3.2 INSTALLATION REQUIREMENTS: The system must be installed by a factory trained contractor/dealer. The mechanical contractor's installation price shall be based on the systems installation requirements. The mechanical contractor bids with complete knowledge of the HVAC system requirements. System startup shall be provided by factory trained personnel.

3.3 DESIGN BASIS: All bidders shall furnish the minimum system standards as defined by the base bid model numbers, model families or as otherwise specified herein (see Key General Specifications Alternate Supplier Checklist). In any event the contractor shall be responsible for all specified items and intents of this document without further compensation.

Appendix D - System Selection Flowchart

This section includes the system selection flowchart developed as part of this paper.

Figure D.1 System Selection Flowchart.



Appendix E - Climate Study Building Floorplans

This appendix includes enlarged floorplans of the building used for the climate study performed as part of this report.

Figure E.1 Basement Floorplan.

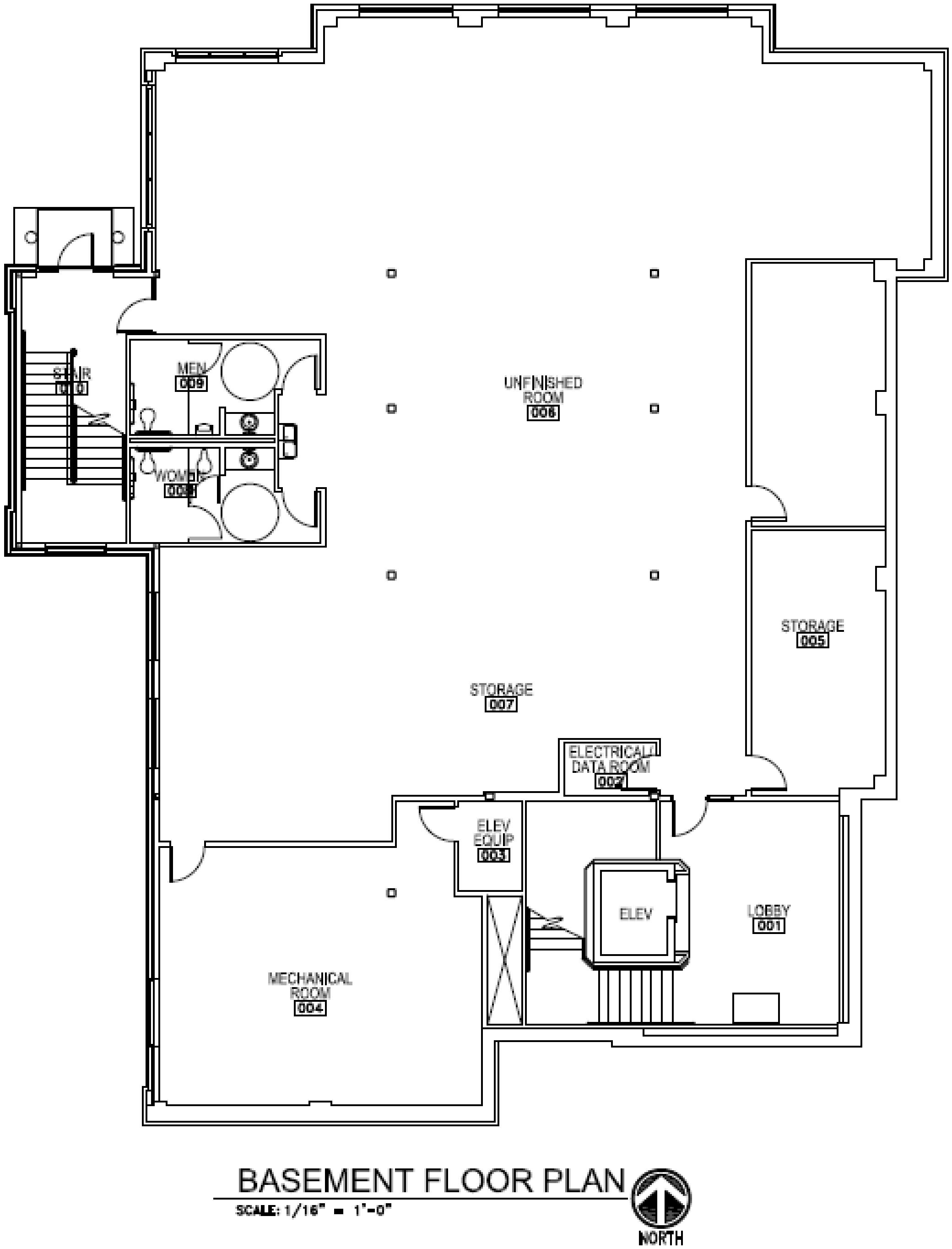


Figure E.2 First Floor Plan

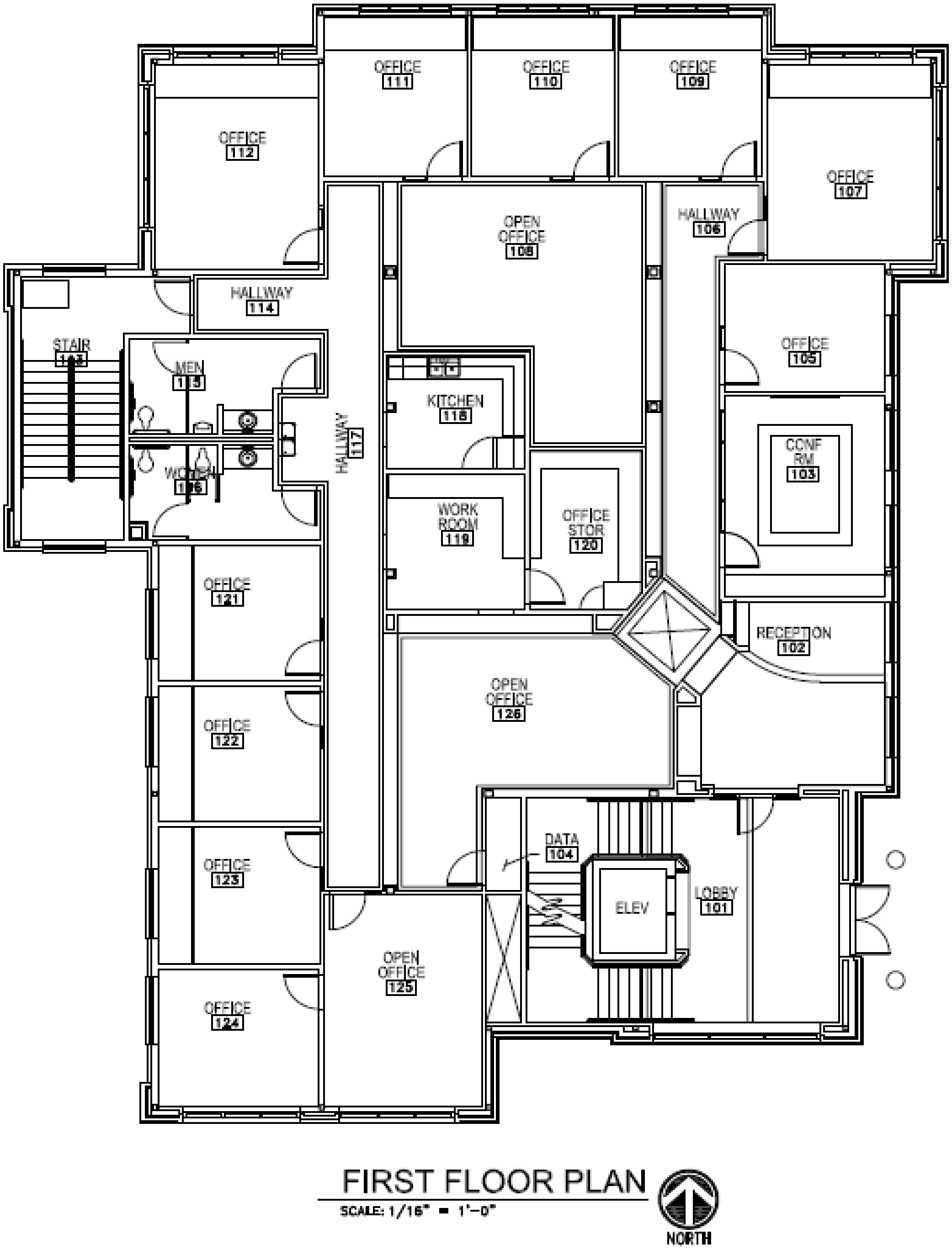
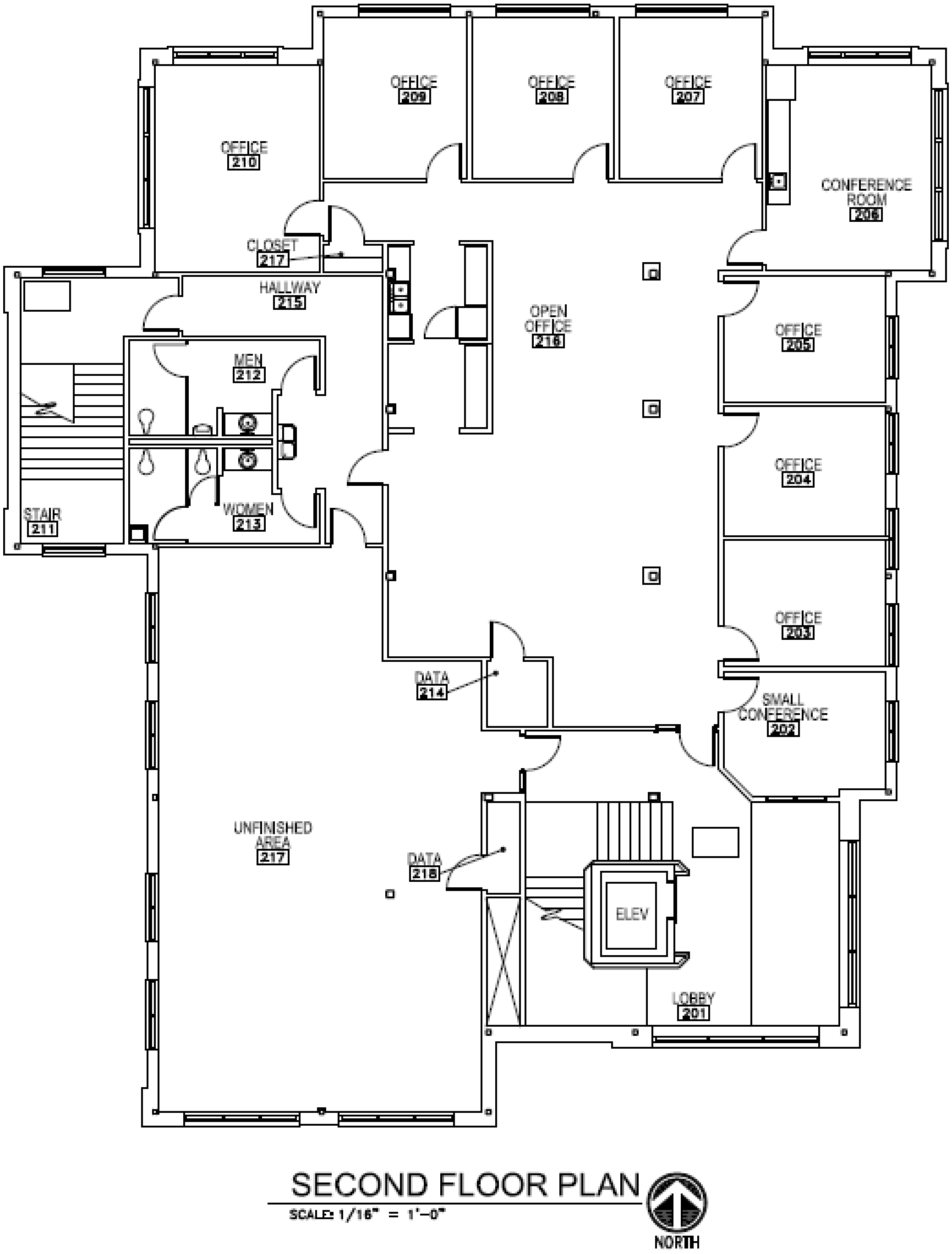


Figure E.3 Second Floor Plan



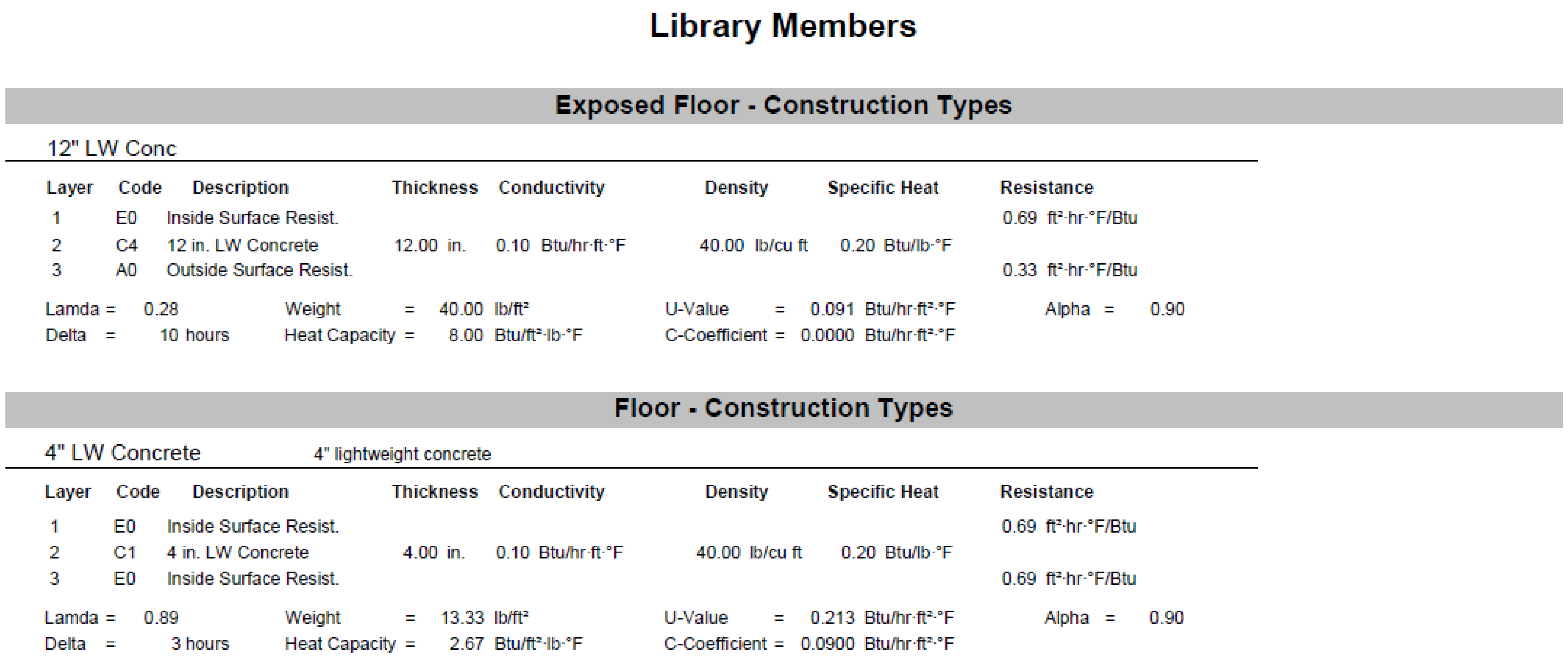
Appendix F - Climate Study TRACE 700 Values

This appendix includes TRACE 700 inputs in Section F.1 and outputs in Section F.2.

F.1 TRACE 700 Inputs

All TRACE 700 input values for Topeka, Kansas are included in this section as Figures F.1 through F.13.

Figure F.1 Floor Construction.



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Exposed Floor - Construction Types

12" LW Conc

Layer	Code	Description	Thickness	Conductivity	Density	Specific Heat	Resistance
1	E0	Inside Surface Resist.					0.69 ft²·hr·°F/Btu
2	C4	12 in. LW Concrete	12.00 in.	0.10 Btu/hr·ft·°F	40.00 lb/cu ft	0.20 Btu/lb·°F	
3	A0	Outside Surface Resist.					0.33 ft²·hr·°F/Btu

Lamda = 0.28   Weight = 40.00 lb/ft²   U-Value = 0.091 Btu/hr·ft²·°F   Alpha = 0.90

Delta = 10 hours   Heat Capacity = 8.00 Btu/ft²·lb·°F   C-Coefficient = 0.0000 Btu/hr·ft²·°F

Floor - Construction Types

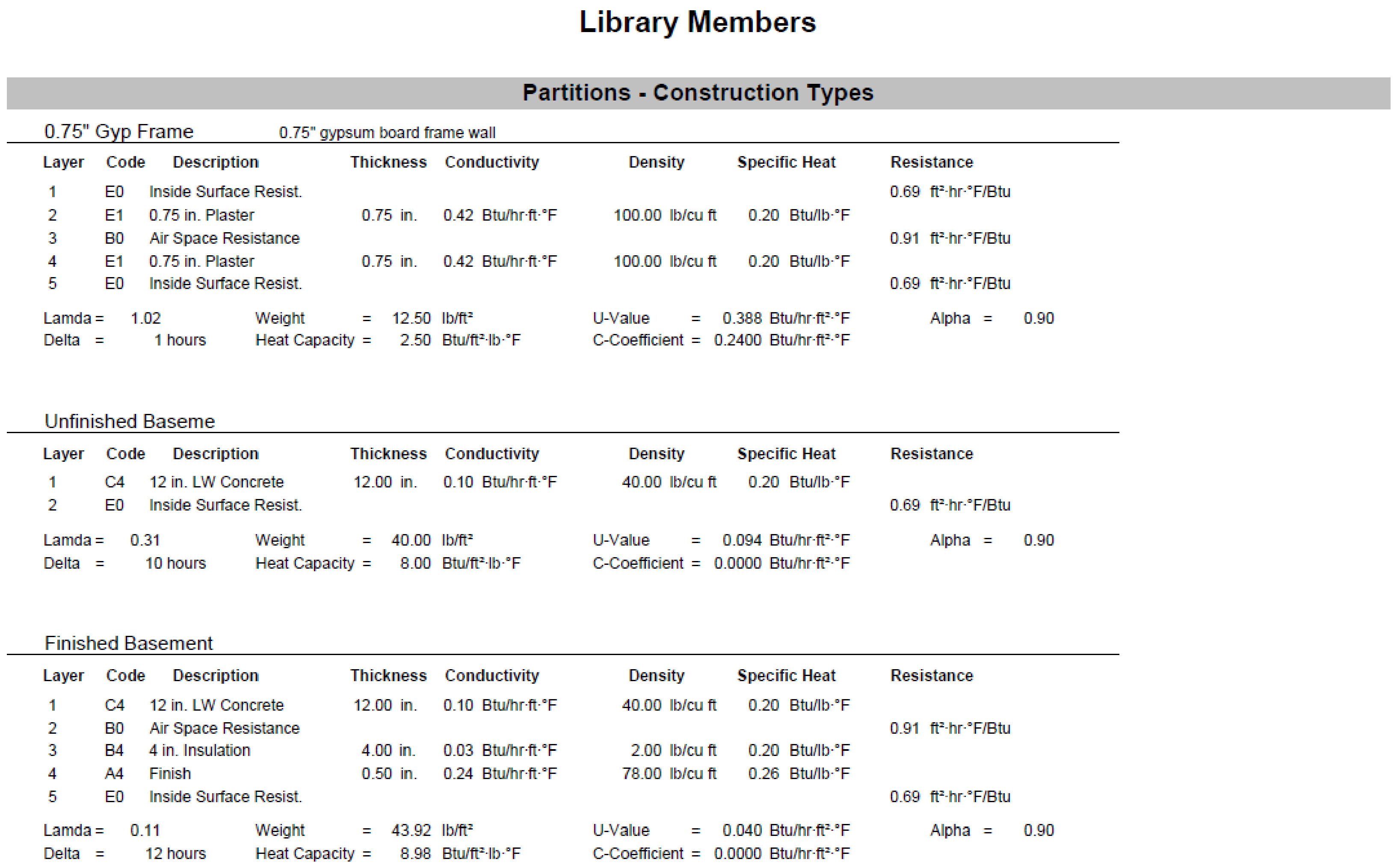
4" LW Concrete   4" lightweight concrete

Layer	Code	Description	Thickness	Conductivity	Density	Specific Heat	Resistance
1	E0	Inside Surface Resist.					0.69 ft²·hr·°F/Btu
2	C1	4 in. LW Concrete	4.00 in.	0.10 Btu/hr·ft·°F	40.00 lb/cu ft	0.20 Btu/lb·°F	
3	E0	Inside Surface Resist.					0.69 ft²·hr·°F/Btu

Lamda = 0.89   Weight = 13.33 lb/ft²   U-Value = 0.213 Btu/hr·ft²·°F   Alpha = 0.90

Delta = 3 hours   Heat Capacity = 2.67 Btu/ft²·lb·°F   C-Coefficient = 0.0900 Btu/hr·ft²·°F

Figure F.2 Partition Construction.



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Partitions - Construction Types

0.75" Gyp Frame   0.75" gypsum board frame wall

Layer	Code	Description	Thickness	Conductivity	Density	Specific Heat	Resistance
1	E0	Inside Surface Resist.					0.69 ft²·hr·°F/Btu
2	E1	0.75 in. Plaster	0.75 in.	0.42 Btu/hr·ft·°F	100.00 lb/cu ft	0.20 Btu/lb·°F	
3	B0	Air Space Resistance					0.91 ft²·hr·°F/Btu
4	E1	0.75 in. Plaster	0.75 in.	0.42 Btu/hr·ft·°F	100.00 lb/cu ft	0.20 Btu/lb·°F	
5	E0	Inside Surface Resist.					0.69 ft²·hr·°F/Btu

Lamda = 1.02   Weight = 12.50 lb/ft²   U-Value = 0.388 Btu/hr·ft²·°F   Alpha = 0.90

Delta = 1 hours   Heat Capacity = 2.50 Btu/ft²·lb·°F   C-Coefficient = 0.2400 Btu/hr·ft²·°F

Unfinished Baseme

Layer	Code	Description	Thickness	Conductivity	Density	Specific Heat	Resistance
1	C4	12 in. LW Concrete	12.00 in.	0.10 Btu/hr·ft·°F	40.00 lb/cu ft	0.20 Btu/lb·°F	
2	E0	Inside Surface Resist.					0.69 ft²·hr·°F/Btu

Lamda = 0.31   Weight = 40.00 lb/ft²   U-Value = 0.094 Btu/hr·ft²·°F   Alpha = 0.90

Delta = 10 hours   Heat Capacity = 8.00 Btu/ft²·lb·°F   C-Coefficient = 0.0000 Btu/hr·ft²·°F

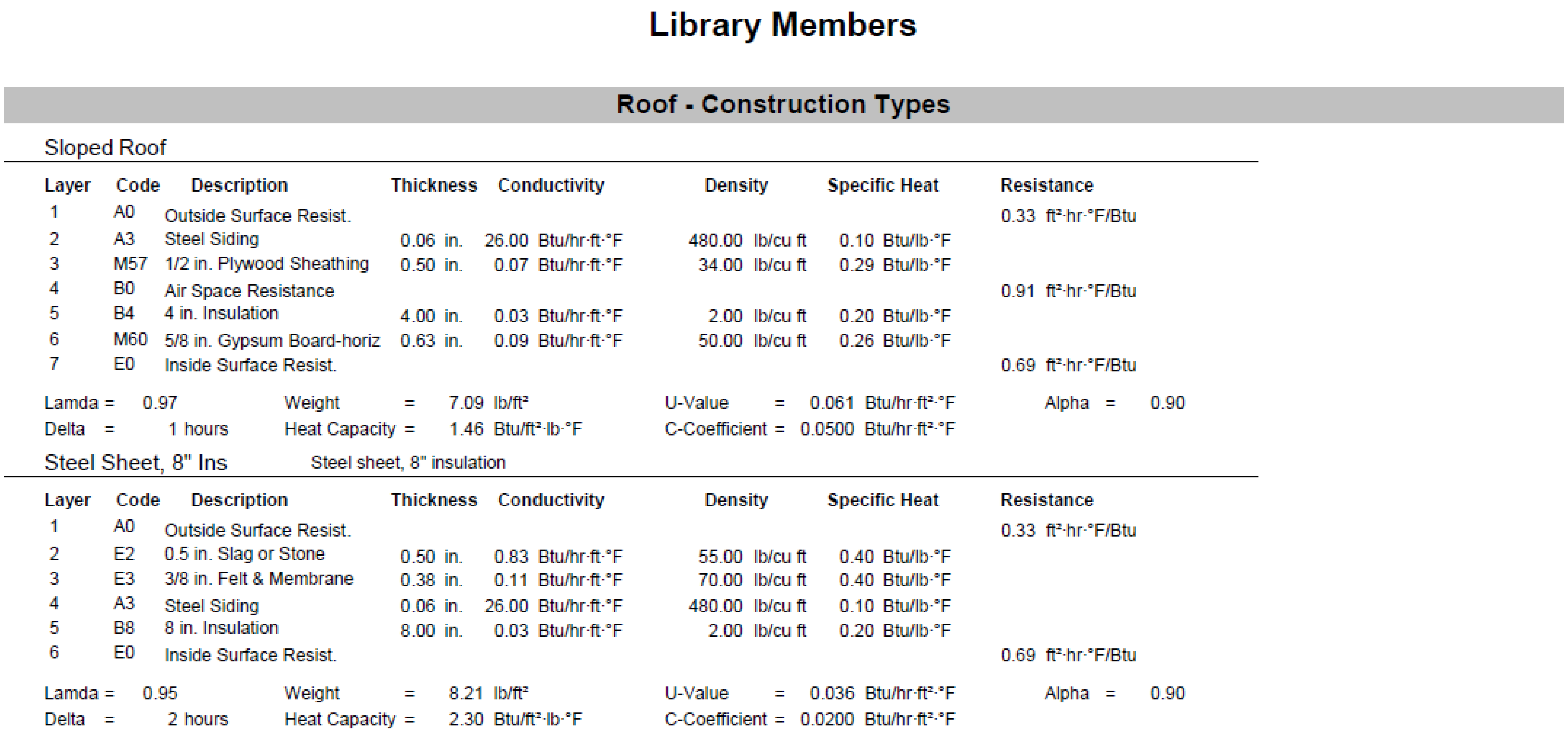
Finished Basement

Layer	Code	Description	Thickness	Conductivity	Density	Specific Heat	Resistance
1	C4	12 in. LW Concrete	12.00 in.	0.10 Btu/hr·ft·°F	40.00 lb/cu ft	0.20 Btu/lb·°F	
2	B0	Air Space Resistance					0.91 ft²·hr·°F/Btu
3	B4	4 in. Insulation	4.00 in.	0.03 Btu/hr·ft·°F	2.00 lb/cu ft	0.20 Btu/lb·°F	
4	A4	Finish	0.50 in.	0.24 Btu/hr·ft·°F	78.00 lb/cu ft	0.26 Btu/lb·°F	
5	E0	Inside Surface Resist.					0.69 ft²·hr·°F/Btu

Lamda = 0.11   Weight = 43.92 lb/ft²   U-Value = 0.040 Btu/hr·ft²·°F   Alpha = 0.90

Delta = 12 hours   Heat Capacity = 8.98 Btu/ft²·lb·°F   C-Coefficient = 0.0000 Btu/hr·ft²·°F

Figure F.3 Roof Construction.



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Roof - Construction Types

Sloped Roof

Layer	Code	Description	Thickness	Conductivity	Density	Specific Heat	Resistance
1	A0	Outside Surface Resist.					0.33 ft²·hr·°F/Btu
2	A3	Steel Siding	0.06 in.	26.00 Btu/hr·ft·°F	480.00 lb/cu ft	0.10 Btu/lb·°F	
3	M57	1/2 in. Plywood Sheathing	0.50 in.	0.07 Btu/hr·ft·°F	34.00 lb/cu ft	0.29 Btu/lb·°F	
4	B0	Air Space Resistance					0.91 ft²·hr·°F/Btu
5	B4	4 in. Insulation	4.00 in.	0.03 Btu/hr·ft·°F	2.00 lb/cu ft	0.20 Btu/lb·°F	
6	M60	5/8 in. Gypsum Board-horiz	0.63 in.	0.09 Btu/hr·ft·°F	50.00 lb/cu ft	0.26 Btu/lb·°F	
7	E0	Inside Surface Resist.					0.69 ft²·hr·°F/Btu

Lamda = 0.97 Weight = 7.09 lb/ft² U-Value = 0.061 Btu/hr·ft²·°F Alpha = 0.90

Delta = 1 hours Heat Capacity = 1.46 Btu/ft²·lb·°F C-Coefficient = 0.0500 Btu/hr·ft²·°F

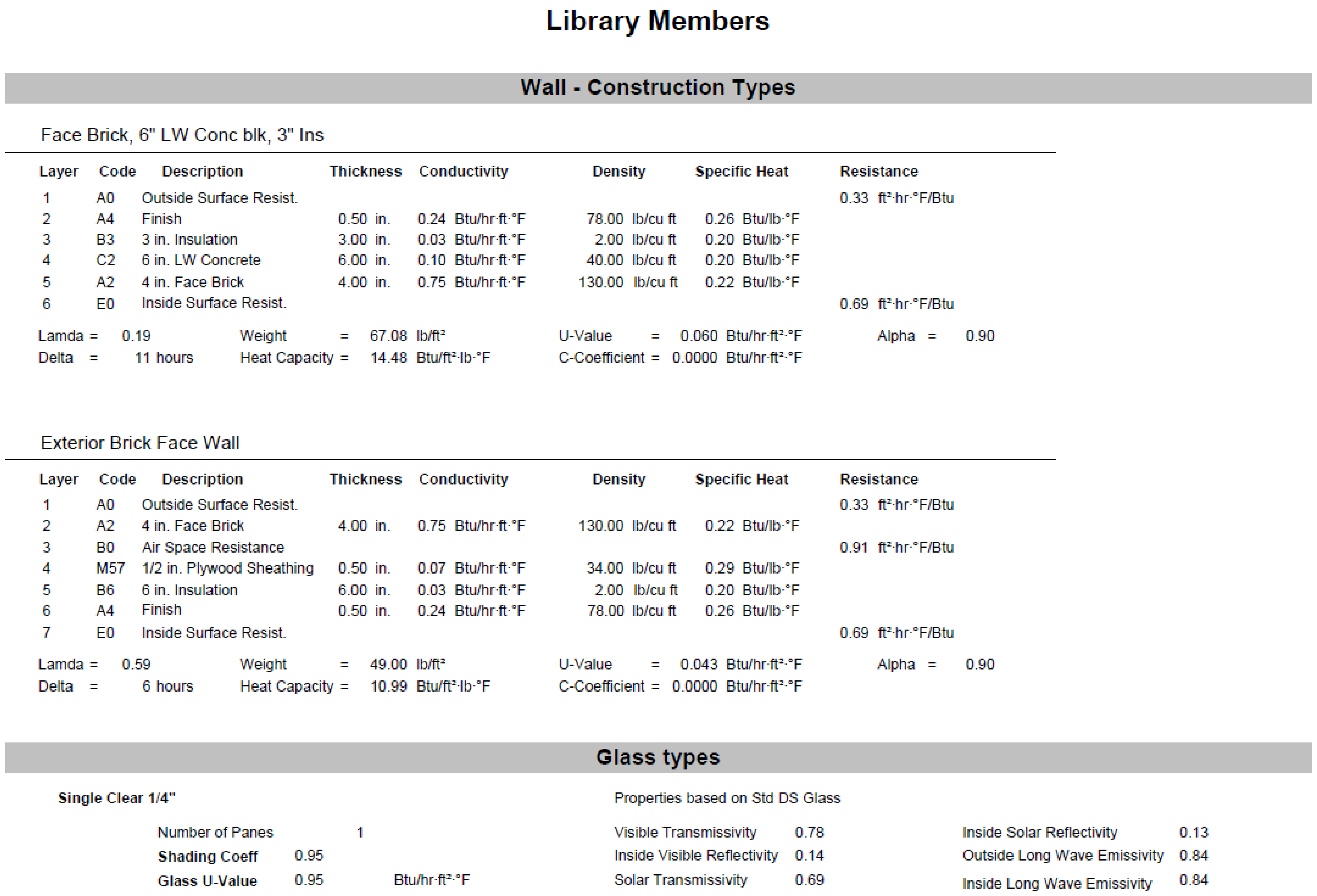
Steel Sheet, 8" Ins Steel sheet, 8" insulation

Layer	Code	Description	Thickness	Conductivity	Density	Specific Heat	Resistance
1	A0	Outside Surface Resist.					0.33 ft²·hr·°F/Btu
2	E2	0.5 in. Slag or Stone	0.50 in.	0.83 Btu/hr·ft·°F	55.00 lb/cu ft	0.40 Btu/lb·°F	
3	E3	3/8 in. Felt & Membrane	0.38 in.	0.11 Btu/hr·ft·°F	70.00 lb/cu ft	0.40 Btu/lb·°F	
4	A3	Steel Siding	0.06 in.	26.00 Btu/hr·ft·°F	480.00 lb/cu ft	0.10 Btu/lb·°F	
5	B8	8 in. Insulation	8.00 in.	0.03 Btu/hr·ft·°F	2.00 lb/cu ft	0.20 Btu/lb·°F	
6	E0	Inside Surface Resist.					0.69 ft²·hr·°F/Btu

Lamda = 0.95 Weight = 8.21 lb/ft² U-Value = 0.036 Btu/hr·ft²·°F Alpha = 0.90

Delta = 2 hours Heat Capacity = 2.30 Btu/ft²·lb·°F C-Coefficient = 0.0200 Btu/hr·ft²·°F

Figure F.4 Wall Construction and Glass Types.



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Wall - Construction Types

Face Brick, 6" LW Conc blk, 3" Ins

Layer	Code	Description	Thickness	Conductivity	Density	Specific Heat	Resistance
1	A0	Outside Surface Resist.					0.33 ft^2·hr·°F/Btu
2	A4	Finish	0.50 in.	0.24 Btu/hr·ft·°F	78.00 lb/cu ft	0.26 Btu/lb·°F	
3	B3	3 in. Insulation	3.00 in.	0.03 Btu/hr·ft·°F	2.00 lb/cu ft	0.20 Btu/lb·°F	
4	C2	6 in. LW Concrete	6.00 in.	0.10 Btu/hr·ft·°F	40.00 lb/cu ft	0.20 Btu/lb·°F	
5	A2	4 in. Face Brick	4.00 in.	0.75 Btu/hr·ft·°F	130.00 lb/cu ft	0.22 Btu/lb·°F	
6	E0	Inside Surface Resist.					0.69 ft^2·hr·°F/Btu

Lamda =	0.19	Weight =	67.08 lb/ft^2	U-Value =	0.060 Btu/hr·ft^2·°F	Alpha =	0.90
Delta =	11 hours	Heat Capacity =	14.48 Btu/ft^2·lb·°F	C-Coefficient =	0.0000 Btu/hr·ft^2·°F		

Exterior Brick Face Wall

Layer	Code	Description	Thickness	Conductivity	Density	Specific Heat	Resistance
1	A0	Outside Surface Resist.					0.33 ft^2·hr·°F/Btu
2	A2	4 in. Face Brick	4.00 in.	0.75 Btu/hr·ft·°F	130.00 lb/cu ft	0.22 Btu/lb·°F	
3	B0	Air Space Resistance					0.91 ft^2·hr·°F/Btu
4	M57	1/2 in. Plywood Sheathing	0.50 in.	0.07 Btu/hr·ft·°F	34.00 lb/cu ft	0.29 Btu/lb·°F	
5	B6	6 in. Insulation	6.00 in.	0.03 Btu/hr·ft·°F	2.00 lb/cu ft	0.20 Btu/lb·°F	
6	A4	Finish	0.50 in.	0.24 Btu/hr·ft·°F	78.00 lb/cu ft	0.26 Btu/lb·°F	
7	E0	Inside Surface Resist.					0.69 ft^2·hr·°F/Btu

Lamda =	0.59	Weight =	49.00 lb/ft^2	U-Value =	0.043 Btu/hr·ft^2·°F	Alpha =	0.90
Delta =	6 hours	Heat Capacity =	10.99 Btu/ft^2·lb·°F	C-Coefficient =	0.0000 Btu/hr·ft^2·°F		

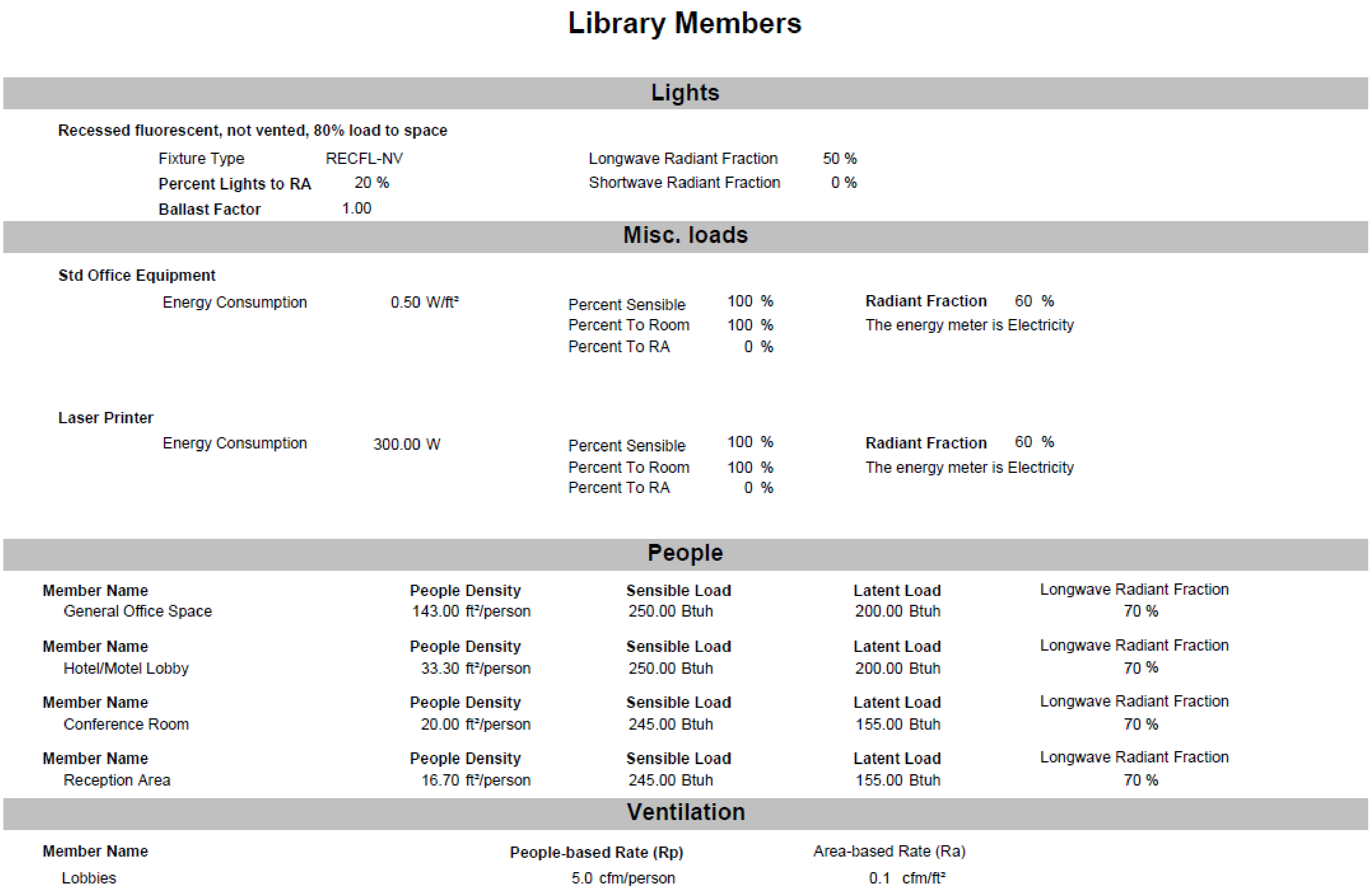
Glass types

Single Clear 1/4"

Properties based on Std DS Glass

Number of Panes	1		Visible Transmissivity	0.78	Inside Solar Reflectivity	0.13
Shading Coeff	0.95		Inside Visible Reflectivity	0.14	Outside Long Wave Emissivity	0.84
Glass U-Value	0.95	Btu/hr·ft^2·°F	Solar Transmissivity	0.69	Inside Long Wave Emissivity	0.84

Figure F.5 Lights, Miscellaneous Loads, People, and Ventilation.



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Lights

Recessed fluorescent, not vented, 80% load to space

Fixture Type	RECFL-NV	Longwave Radiant Fraction	50 %
Percent Lights to RA	20 %	Shortwave Radiant Fraction	0 %
Ballast Factor	1.00		

Misc. loads

Std Office Equipment

Energy Consumption	0.50 W/ft²	Percent Sensible	100 %	**Radiant Fraction**	60 %
		Percent To Room	100 %	The energy meter is Electricity	
		Percent To RA	0 %		

Laser Printer

Energy Consumption	300.00 W	Percent Sensible	100 %	**Radiant Fraction**	60 %
		Percent To Room	100 %	The energy meter is Electricity	
		Percent To RA	0 %		

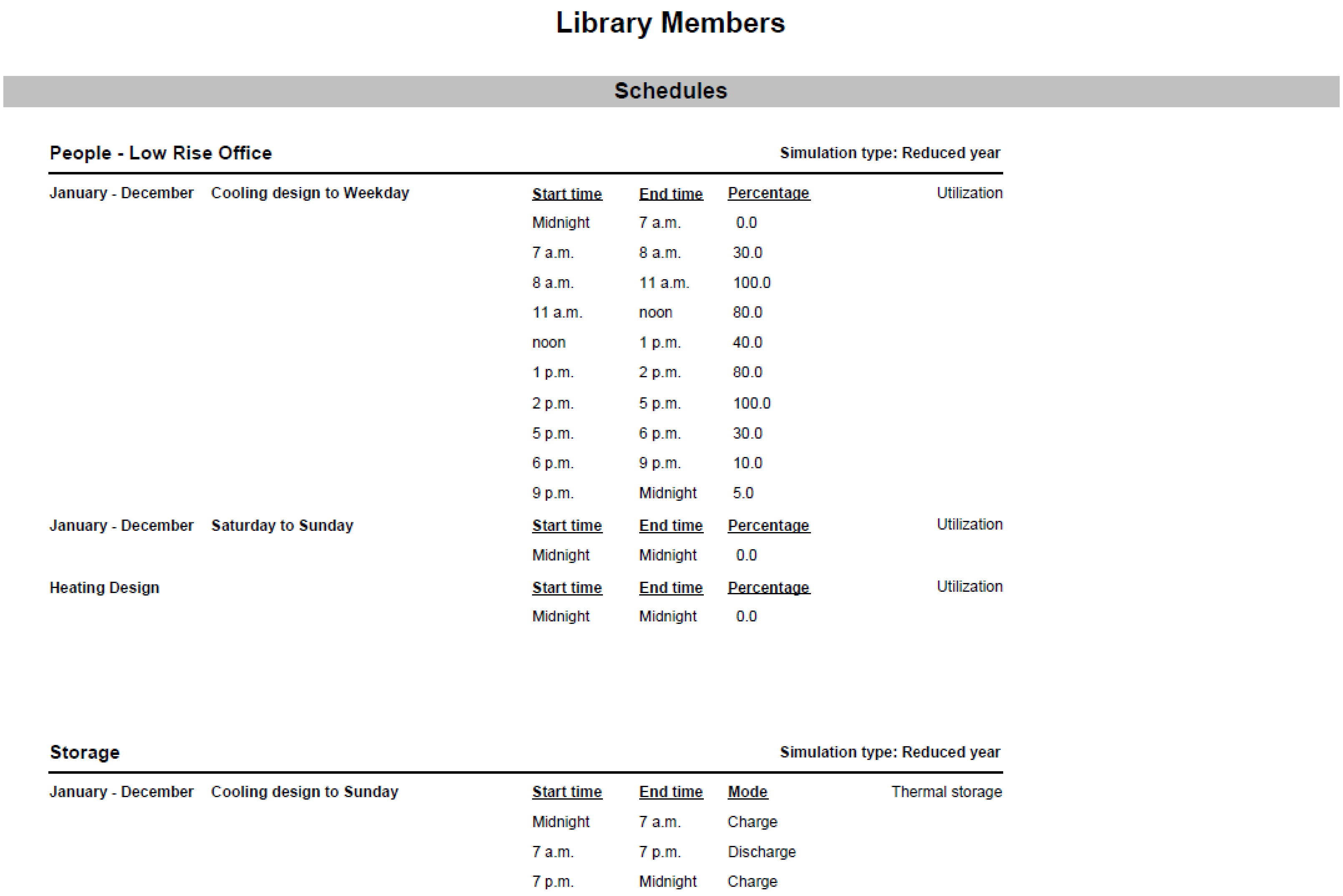
People

Member Name	**People Density**	**Sensible Load**	**Latent Load**	Longwave Radiant Fraction
General Office Space	143.00 ft²/person	250.00 Btuh	200.00 Btuh	70 %
Member Name	**People Density**	**Sensible Load**	**Latent Load**	Longwave Radiant Fraction
Hotel/Motel Lobby	33.30 ft²/person	250.00 Btuh	200.00 Btuh	70 %
Member Name	**People Density**	**Sensible Load**	**Latent Load**	Longwave Radiant Fraction
Conference Room	20.00 ft²/person	245.00 Btuh	155.00 Btuh	70 %
Member Name	**People Density**	**Sensible Load**	**Latent Load**	Longwave Radiant Fraction
Reception Area	16.70 ft²/person	245.00 Btuh	155.00 Btuh	70 %

Ventilation

Member Name	**People-based Rate (Rp)**	Area-based Rate (Ra)
Lobbies	5.0 cfm/person	0.1 cfm/ft²

Figure F.6 Occupancy Schedules.



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Schedules

People - Low Rise Office Simulation type: Reduced year

		Start time	End time	Percentage	Utilization
January - December	Cooling design to Weekday	Midnight	7 a.m.	0.0	
		7 a.m.	8 a.m.	30.0	
		8 a.m.	11 a.m.	100.0	
		11 a.m.	noon	80.0	
		noon	1 p.m.	40.0	
		1 p.m.	2 p.m.	80.0	
		2 p.m.	5 p.m.	100.0	
		5 p.m.	6 p.m.	30.0	
		6 p.m.	9 p.m.	10.0	
		9 p.m.	Midnight	5.0	

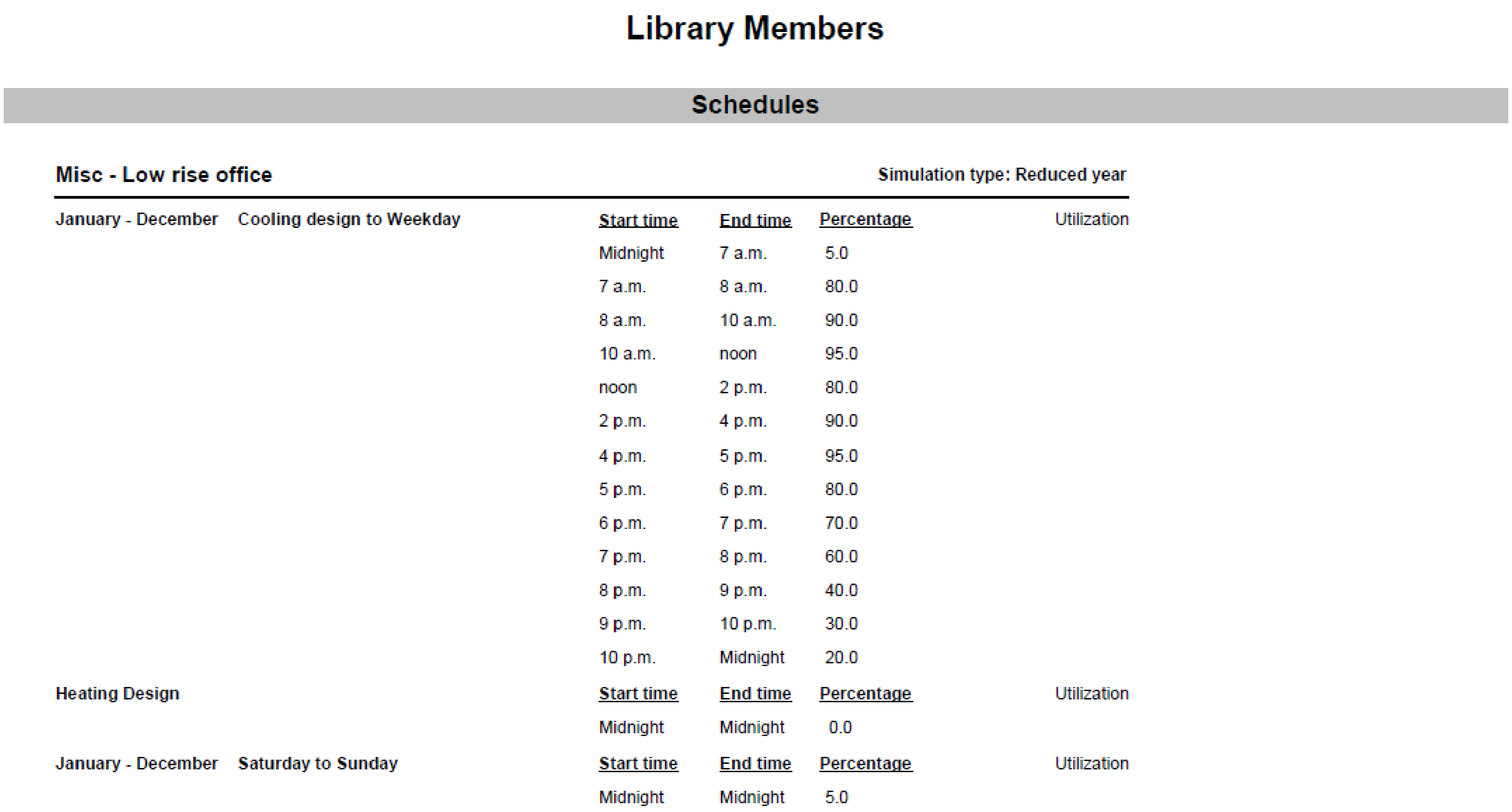
		Start time	End time	Percentage	Utilization
January - December	Saturday to Sunday	Midnight	Midnight	0.0	

	Start time	End time	Percentage	Utilization
Heating Design	Midnight	Midnight	0.0	

Storage Simulation type: Reduced year

		Start time	End time	Mode	Thermal storage
January - December	Cooling design to Sunday	Midnight	7 a.m.	Charge	
		7 a.m.	7 p.m.	Discharge	
		7 p.m.	Midnight	Charge	

Figure F.7 Miscellaneous Load Schedules.



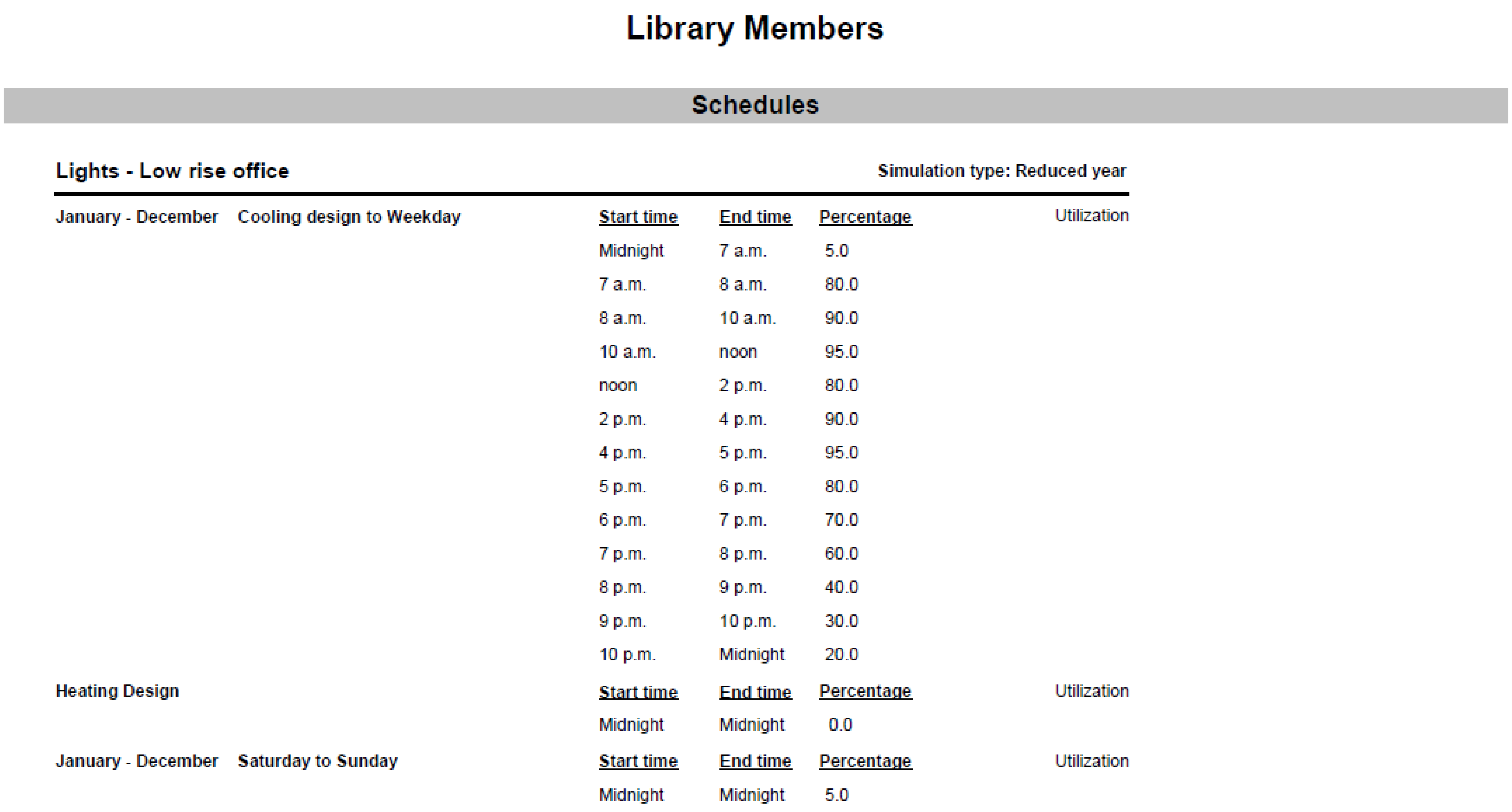
Library Members

Schedules

Misc - Low rise office — **Simulation type: Reduced year**

January - December	Cooling design to Weekday	Start time	End time	Percentage	Utilization
		Midnight	7 a.m.	5.0	
		7 a.m.	8 a.m.	80.0	
		8 a.m.	10 a.m.	90.0	
		10 a.m.	noon	95.0	
		noon	2 p.m.	80.0	
		2 p.m.	4 p.m.	90.0	
		4 p.m.	5 p.m.	95.0	
		5 p.m.	6 p.m.	80.0	
		6 p.m.	7 p.m.	70.0	
		7 p.m.	8 p.m.	60.0	
		8 p.m.	9 p.m.	40.0	
		9 p.m.	10 p.m.	30.0	
		10 p.m.	Midnight	20.0	
Heating Design		Start time	End time	Percentage	Utilization
		Midnight	Midnight	0.0	
January - December	Saturday to Sunday	Start time	End time	Percentage	Utilization
		Midnight	Midnight	5.0	

Figure F.8 Lighting Schedules.



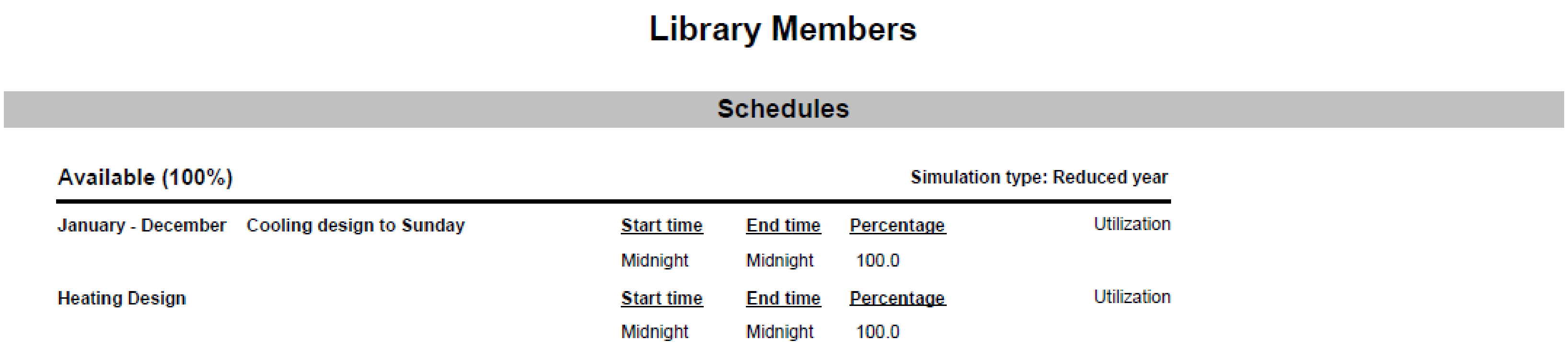
Library Members

Schedules

Lights - Low rise office — **Simulation type: Reduced year**

		Start time	End time	Percentage	Utilization
January - December	**Cooling design to Weekday**				
		Midnight	7 a.m.	5.0	
		7 a.m.	8 a.m.	80.0	
		8 a.m.	10 a.m.	90.0	
		10 a.m.	noon	95.0	
		noon	2 p.m.	80.0	
		2 p.m.	4 p.m.	90.0	
		4 p.m.	5 p.m.	95.0	
		5 p.m.	6 p.m.	80.0	
		6 p.m.	7 p.m.	70.0	
		7 p.m.	8 p.m.	60.0	
		8 p.m.	9 p.m.	40.0	
		9 p.m.	10 p.m.	30.0	
		10 p.m.	Midnight	20.0	
Heading Design		**Start time**	**End time**	**Percentage**	Utilization
		Midnight	Midnight	0.0	
January - December	**Saturday to Sunday**	**Start time**	**End time**	**Percentage**	Utilization
		Midnight	Midnight	5.0	

Figure F.9 Availability Schedule.



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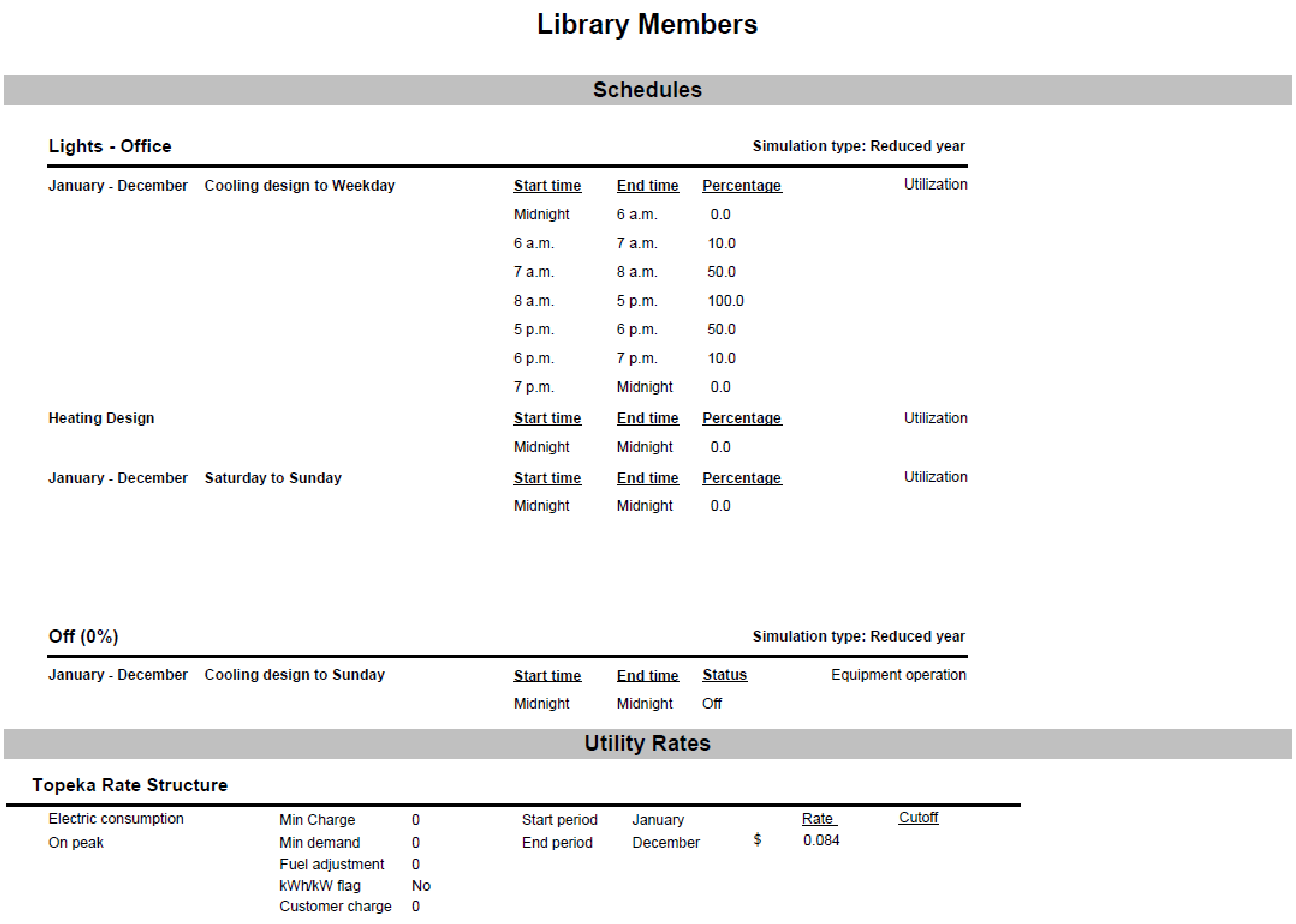
Schedules

Available (100%)

Simulation type: Reduced year

January - December	**Cooling design to Sunday**	**Start time**	**End time**	**Percentage**	Utilization
		Midnight	Midnight	100.0	
Heating Design		**Start time**	**End time**	**Percentage**	Utilization
		Midnight	Midnight	100.0	

Figure F.10 Lighting and Utility Rate Schedules.



Library Members

Schedules

Lights - Office — **Simulation type: Reduced year**

		Start time	End time	Percentage	
January - December	**Cooling design to Weekday**				Utilization
		Midnight	6 a.m.	0.0	
		6 a.m.	7 a.m.	10.0	
		7 a.m.	8 a.m.	50.0	
		8 a.m.	5 p.m.	100.0	
		5 p.m.	6 p.m.	50.0	
		6 p.m.	7 p.m.	10.0	
		7 p.m.	Midnight	0.0	
Heating Design		**Start time**	**End time**	**Percentage**	Utilization
		Midnight	Midnight	0.0	
January - December	**Saturday to Sunday**	**Start time**	**End time**	**Percentage**	Utilization
		Midnight	Midnight	0.0	

Off (0%) — **Simulation type: Reduced year**

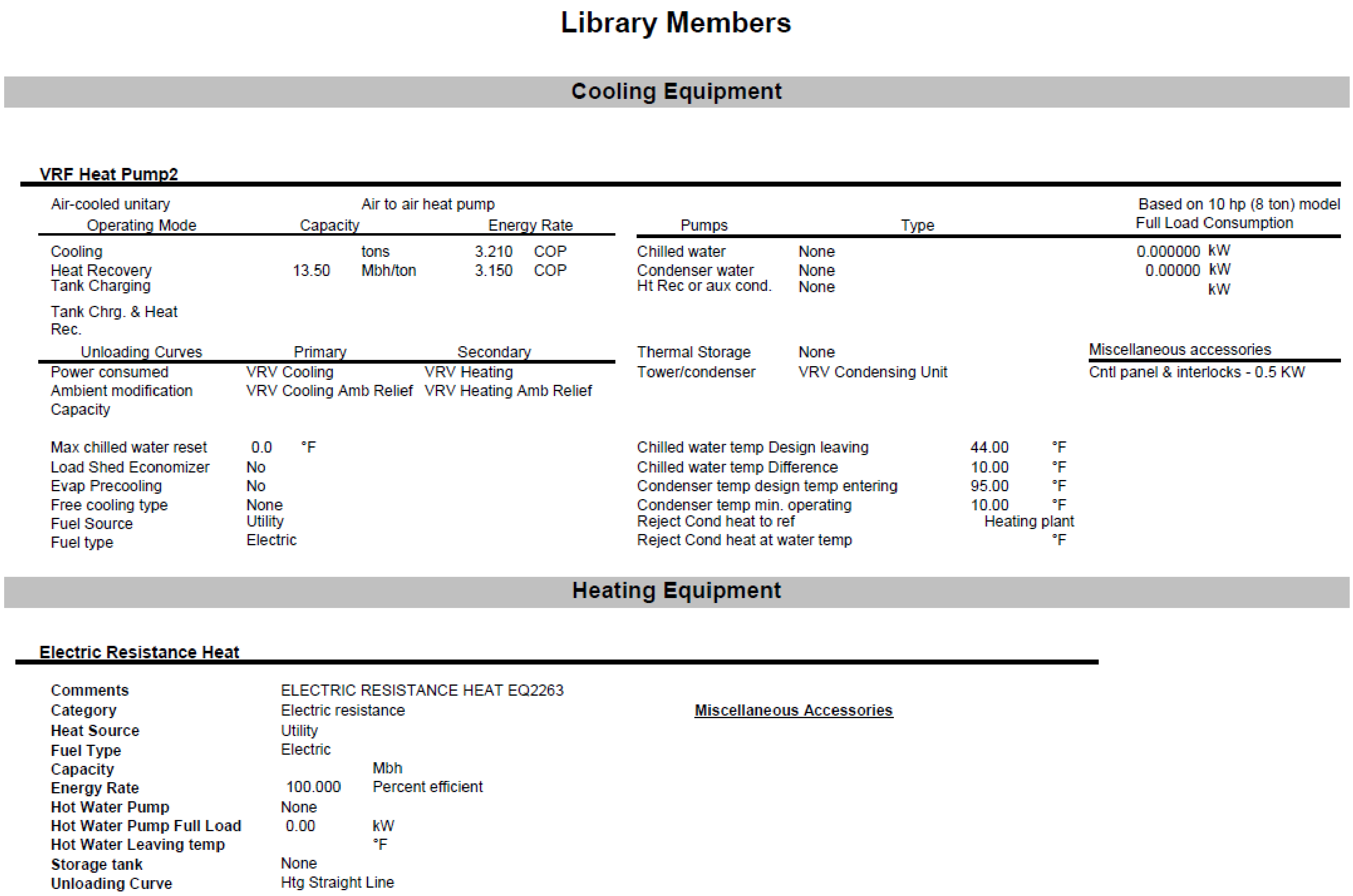
		Start time	End time	Status	
January - December	**Cooling design to Sunday**				Equipment operation
		Midnight	Midnight	Off	

Utility Rates

Topeka Rate Structure

						Rate	Cutoff
Electric consumption	Min Charge	0	Start period	January			
On peak	Min demand	0	End period	December	$	0.084	
	Fuel adjustment	0					
	kWh/kW flag	No					
	Customer charge	0					

Figure F.11 Cooling and Heating Equipment.



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Cooling Equipment

VRF Heat Pump2

Air-cooled unitary Air to air heat pump Based on 10 hp (8 ton) model

Operating Mode	Capacity		Energy Rate	
Cooling		tons	3.210	COP
Heat Recovery	13.50	Mbh/ton	3.150	COP
Tank Charging				
Tank Chrg. & Heat Rec.				

Pumps	Type	Full Load Consumption	
Chilled water	None	0.000000	kW
Condenser water	None	0.00000	kW
Ht Rec or aux cond.	None		kW

Unloading Curves	Primary	Secondary
Power consumed	VRV Cooling	VRV Heating
Ambient modification	VRV Cooling Amb Relief	VRV Heating Amb Relief
Capacity		

Thermal Storage	None
Tower/condenser	VRV Condensing Unit

Miscellaneous accessories
Cntl panel & interlocks - 0.5 KW

Max chilled water reset	0.0	°F
Load Shed Economizer	No	
Evap Precooling	No	
Free cooling type	None	
Fuel Source	Utility	
Fuel type	Electric	

Chilled water temp Design leaving	44.00	°F
Chilled water temp Difference	10.00	°F
Condenser temp design temp entering	95.00	°F
Condenser temp min. operating	10.00	°F
Reject Cond heat to ref	Heating plant	
Reject Cond heat at water temp		°F

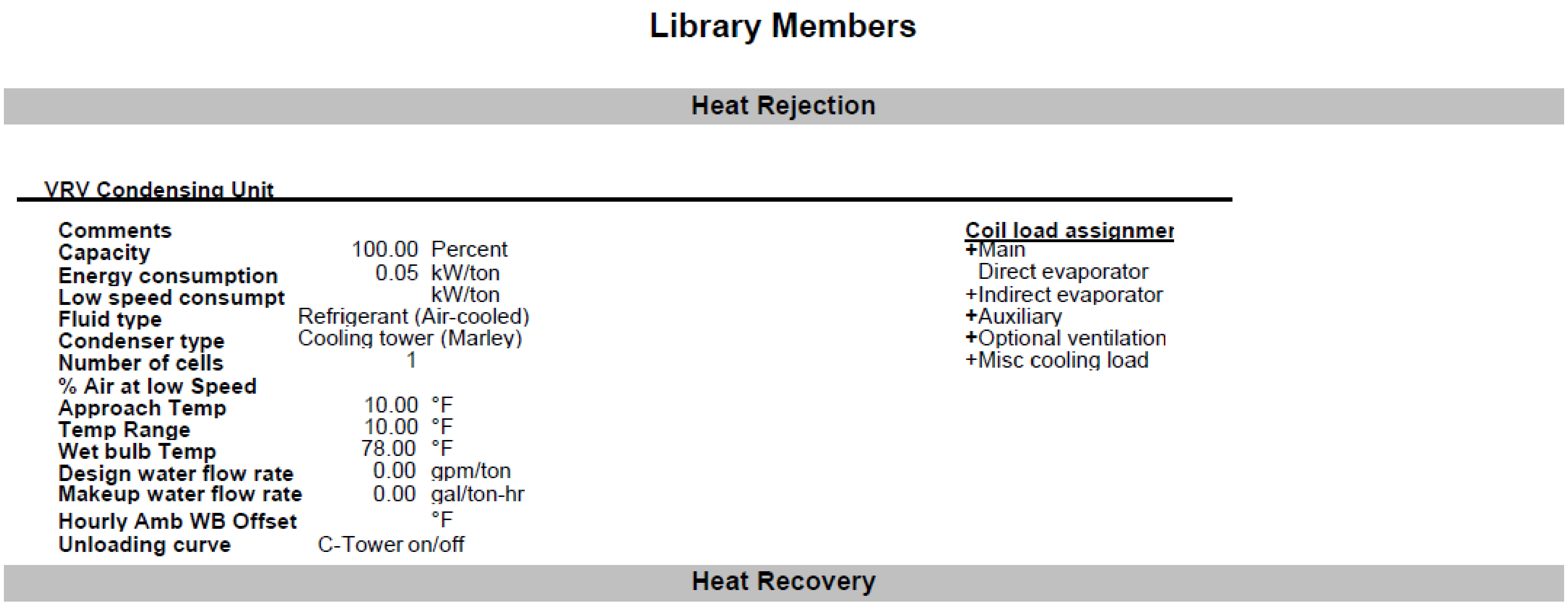
Heating Equipment

Electric Resistance Heat

Comments	ELECTRIC RESISTANCE HEAT EQ2263	
Category	Electric resistance	
Heat Source	Utility	
Fuel Type	Electric	
Capacity		Mbh
Energy Rate	100.000	Percent efficient
Hot Water Pump	None	
Hot Water Pump Full Load	0.00	kW
Hot Water Leaving temp		°F
Storage tank	None	
Unloading Curve	Htg Straight Line	

Miscellaneous Accessories

Figure F.12 Heat Rejection.



Library Members

Heat Rejection

VRV Condensing Unit

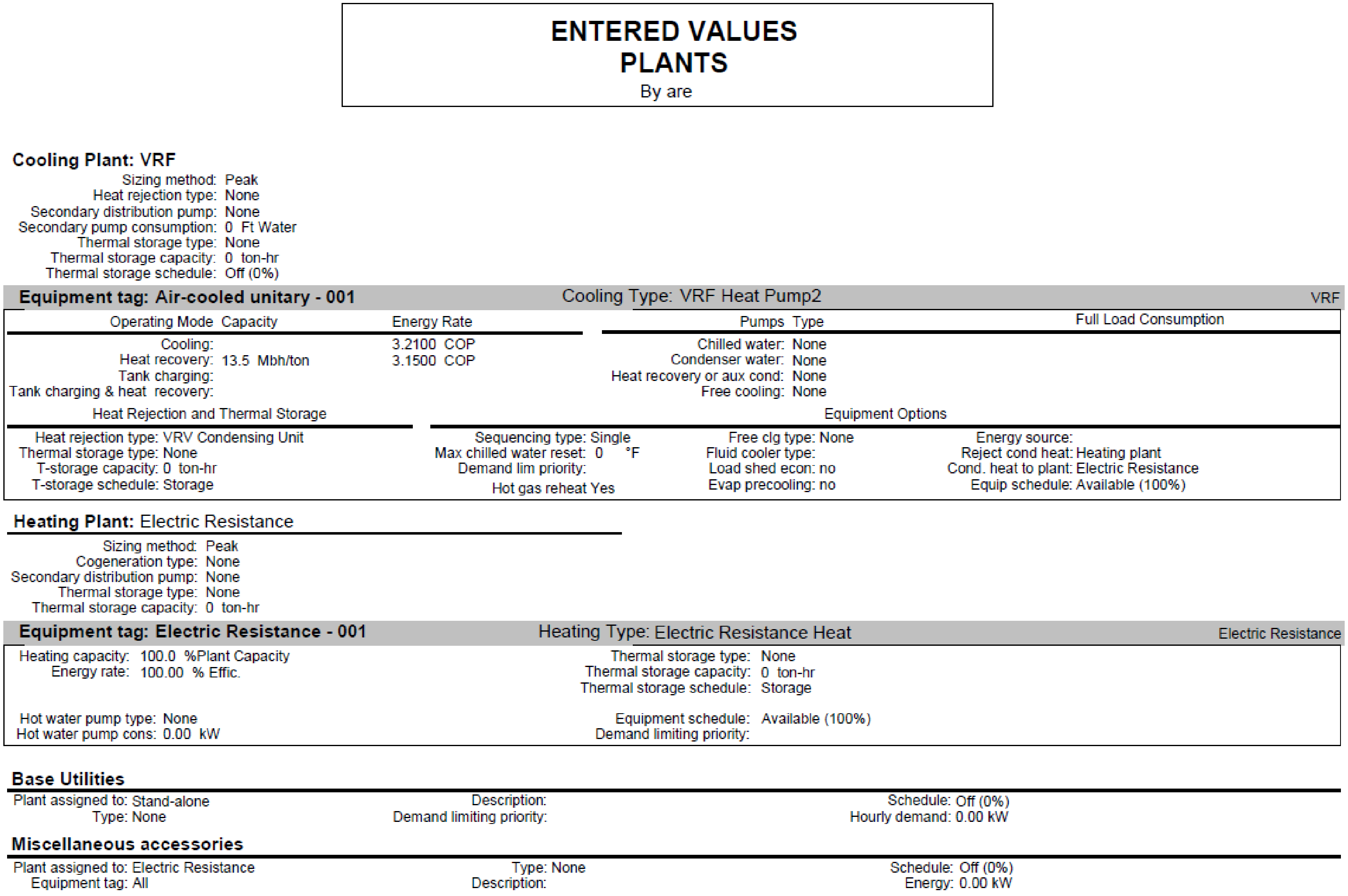
Comments		
Capacity	100.00	Percent
Energy consumption	0.05	kW/ton
Low speed consumpt		kW/ton
Fluid type	Refrigerant (Air-cooled)	
Condenser type	Cooling tower (Marley)	
Number of cells	1	
% Air at low Speed		
Approach Temp	10.00	°F
Temp Range	10.00	°F
Wet bulb Temp	78.00	°F
Design water flow rate	0.00	gpm/ton
Makeup water flow rate	0.00	gal/ton-hr
Hourly Amb WB Offset		°F
Unloading curve	C-Tower on/off	

Coil load assignmer

+Main
  Direct evaporator
+Indirect evaporator
+Auxiliary
+Optional ventilation
+Misc cooling load

Heat Recovery

Figure F.13 Plant Values.

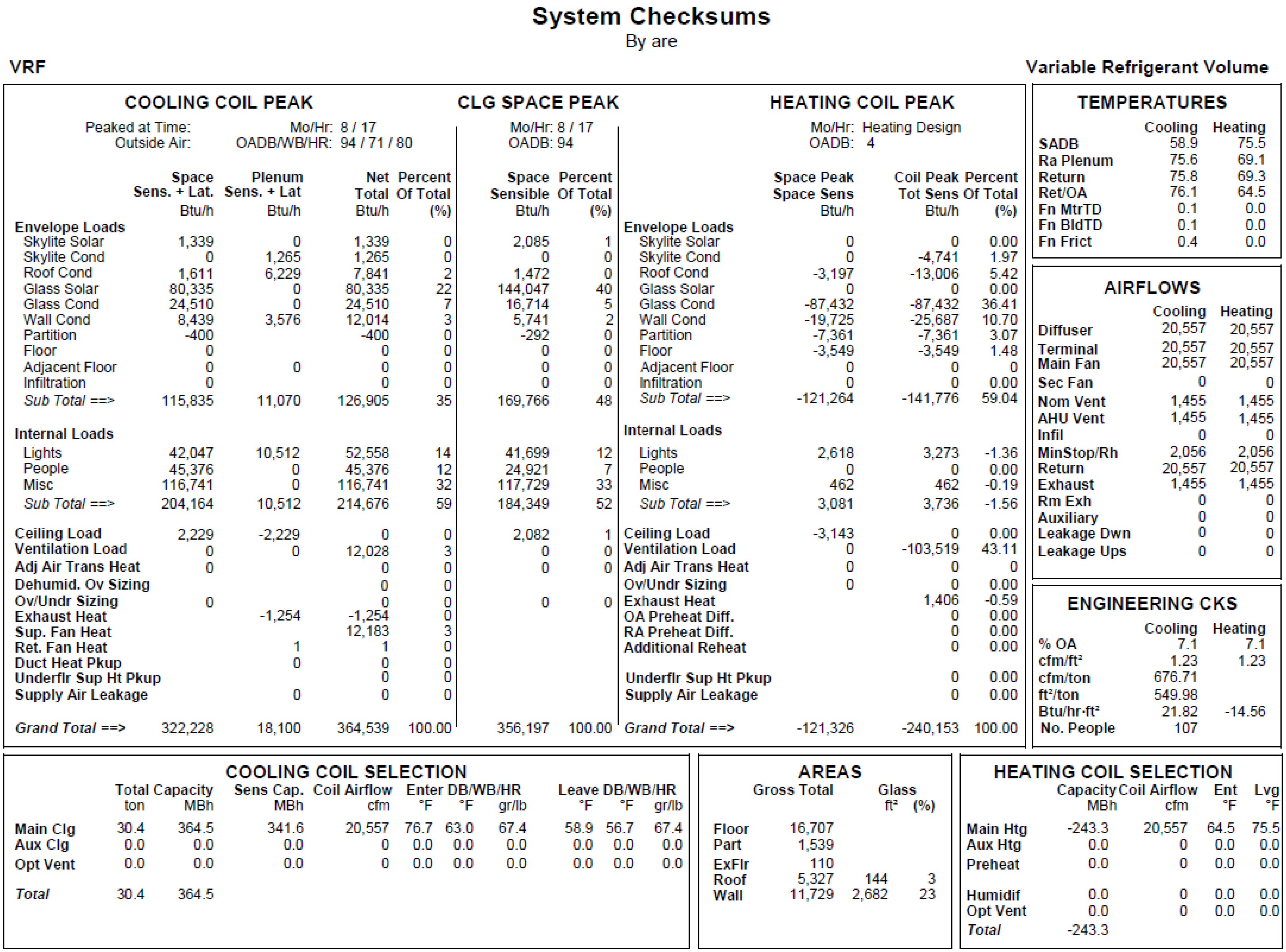


```
                         ENTERED VALUES
                             PLANTS
                             By are

Cooling Plant: VRF
            Sizing method:  Peak
        Heat rejection type:  None
 Secondary distribution pump:  None
Secondary pump consumption:  0  Ft Water
       Thermal storage type:  None
   Thermal storage capacity:  0  ton-hr
   Thermal storage schedule:  Off (0%)
Equipment tag: Air-cooled unitary - 001        Cooling Type: VRF Heat Pump2                                   VRF
        Operating Mode  Capacity        Energy Rate               Pumps  Type                  Full Load Consumption
                Cooling:                3.2100  COP                  Chilled water:  None
          Heat recovery:  13.5  Mbh/ton 3.1500  COP              Condenser water:  None
          Tank charging:                                  Heat recovery or aux cond:  None
Tank charging & heat  recovery:                                       Free cooling:  None
      Heat Rejection and Thermal Storage                         Equipment Options
  Heat rejection type: VRV Condensing Unit   Sequencing type: Single     Free clg type: None        Energy source:
Thermal storage type: None           Max chilled water reset:  0   °F   Fluid cooler type:      Reject cond heat: Heating plant
  T-storage capacity: 0  ton-hr          Demand lim priority:         Load shed econ: no   Cond. heat to plant: Electric Resistance
  T-storage schedule: Storage              Hot gas reheat Yes         Evap precooling: no     Equip schedule: Available (100%)
Heating Plant: Electric Resistance
            Sizing method:  Peak
        Cogeneration type:  None
Secondary distribution pump:  None
      Thermal storage type:  None
  Thermal storage capacity:  0  ton-hr
Equipment tag: Electric Resistance - 001       Heating Type: Electric Resistance Heat                 Electric Resistance
 Heating capacity:  100.0  %Plant Capacity              Thermal storage type:  None
      Energy rate:  100.00  % Effic.                Thermal storage capacity:  0  ton-hr
                                                    Thermal storage schedule:  Storage

 Hot water pump type:  None                            Equipment schedule:  Available (100%)
 Hot water pump cons:  0.00  kW                   Demand limiting priority:

Base Utilities
Plant assigned to: Stand-alone                Description:                     Schedule: Off (0%)
             Type: None           Demand limiting priority:               Hourly demand: 0.00 kW
Miscellaneous accessories
Plant assigned to: Electric Resistance               Type: None               Schedule:  Off (0%)
    Equipment tag: All                        Description:                      Energy:  0.00 kW
```

F.2 TRACE 700 Outputs

System loads, energy consumption, and economic costs for Topeka, Kansas are all included in this section as Figures F.14 through F.18.

Figure F.14 System Checksums.



System Checksums

By are

VRF | **Variable Refrigerant Volume**

COOLING COIL PEAK / CLG SPACE PEAK

Cooling Coil Peak — Peaked at Time: Mo/Hr: 8 / 17; Outside Air: OADB/WB/HR: 94 / 71 / 80

CLG Space Peak — Mo/Hr: 8 / 17; OADB: 94

	Space Sens. + Lat. Btu/h	Plenum Sens. + Lat Btu/h	Net Total Btu/h	Percent Of Total (%)	Space Sensible Btu/h	Percent Of Total (%)
Envelope Loads						
Skylite Solar	1,339	0	1,339	0	2,085	1
Skylite Cond	0	1,265	1,265	0	0	0
Roof Cond	1,611	6,229	7,841	2	1,472	0
Glass Solar	80,335	0	80,335	22	144,047	40
Glass Cond	24,510	0	24,510	7	16,714	5
Wall Cond	8,439	3,576	12,014	3	5,741	2
Partition	-400		-400	0	-292	0
Floor	0		0	0	0	0
Adjacent Floor	0	0	0	0	0	0
Infiltration	0		0	0	0	0
Sub Total ==>	115,835	11,070	126,905	35	169,766	48
Internal Loads						
Lights	42,047	10,512	52,558	14	41,699	12
People	45,376	0	45,376	12	24,921	7
Misc	116,741	0	116,741	32	117,729	33
Sub Total ==>	204,164	10,512	214,676	59	184,349	52
Ceiling Load	2,229	-2,229	0	0	2,082	1
Ventilation Load	0	0	12,028	3	0	0
Adj Air Trans Heat	0		0	0	0	0
Dehumid. Ov Sizing			0	0		
Ov/Undr Sizing	0		0	0	0	0
Exhaust Heat		-1,254	-1,254	0		
Sup. Fan Heat			12,183	3		
Ret. Fan Heat		1	1	0		
Duct Heat Pkup		0	0	0		
Underflr Sup Ht Pkup			0	0		
Supply Air Leakage		0	0	0		
Grand Total ==>	322,228	18,100	364,539	100.00	356,197	100.00

HEATING COIL PEAK

Mo/Hr: Heating Design; OADB: 4

	Space Peak Space Sens Btu/h	Coil Peak Tot Sens Btu/h	Percent Of Total (%)
Envelope Loads			
Skylite Solar	0	0	0.00
Skylite Cond	0	-4,741	1.97
Roof Cond	-3,197	-13,006	5.42
Glass Solar	0	0	0.00
Glass Cond	-87,432	-87,432	36.41
Wall Cond	-19,725	-25,687	10.70
Partition	-7,361	-7,361	3.07
Floor	-3,549	-3,549	1.48
Adjacent Floor	0	0	0
Infiltration	0	0	0.00
Sub Total ==>	-121,264	-141,776	59.04
Internal Loads			
Lights	2,618	3,273	-1.36
People	0	0	0.00
Misc	462	462	-0.19
Sub Total ==>	3,081	3,736	-1.56
Ceiling Load	-3,143	0	0.00
Ventilation Load	0	-103,519	43.11
Adj Air Trans Heat	0	0	0
Ov/Undr Sizing	0	0	0.00
Exhaust Heat		1,406	-0.59
OA Preheat Diff.		0	0.00
RA Preheat Diff.		0	0.00
Additional Reheat		0	0.00
Underflr Sup Ht Pkup		0	0.00
Supply Air Leakage		0	0.00
Grand Total ==>	-121,326	-240,153	100.00

TEMPERATURES

	Cooling	Heating
SADB	58.9	75.5
Ra Plenum	75.6	69.1
Return	75.8	69.3
Ret/OA	76.1	64.5
Fn MtrTD	0.1	0.0
Fn BldTD	0.1	0.0
Fn Frict	0.4	0.0

AIRFLOWS

	Cooling	Heating
Diffuser	20,557	20,557
Terminal	20,557	20,557
Main Fan	20,557	20,557
Sec Fan	0	0
Nom Vent	1,455	1,455
AHU Vent	1,455	1,455
Infil	0	0
MinStop/Rh	2,056	2,056
Return	20,557	20,557
Exhaust	1,455	1,455
Rm Exh	0	0
Auxiliary	0	0
Leakage Dwn	0	0
Leakage Ups	0	0

ENGINEERING CKS

	Cooling	Heating
% OA	7.1	7.1
cfm/ft²	1.23	1.23
cfm/ton	676.71	
ft²/ton	549.98	
Btu/hr·ft²	21.82	-14.56
No. People	107	

COOLING COIL SELECTION

	Total Capacity ton	Total Capacity MBh	Sens Cap. MBh	Coil Airflow cfm	Enter DB/WB/HR °F	Enter DB/WB/HR °F	Enter DB/WB/HR gr/lb	Leave DB/WB/HR °F	Leave DB/WB/HR °F	Leave DB/WB/HR gr/lb
Main Clg	30.4	364.5	341.6	20,557	76.7	63.0	67.4	58.9	56.7	67.4
Aux Clg	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Opt Vent	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Total	30.4	364.5								

AREAS

	Gross Total	Glass ft²	Glass (%)
Floor	16,707		
Part	1,539		
ExFlr	110		
Roof	5,327	144	3
Wall	11,729	2,682	23

HEATING COIL SELECTION

	Capacity MBh	Coil Airflow cfm	Ent °F	Lvg °F
Main Htg	-243.3	20,557	64.5	75.5
Aux Htg	0.0	0	0.0	0.0
Preheat	0.0	0	0.0	0.0
Humidif	0.0	0	0.0	0.0
Opt Vent	0.0	0	0.0	0.0
Total	-243.3			

Figure F.15 Monthly Utility Costs.

MONTHLY UTILITY COSTS

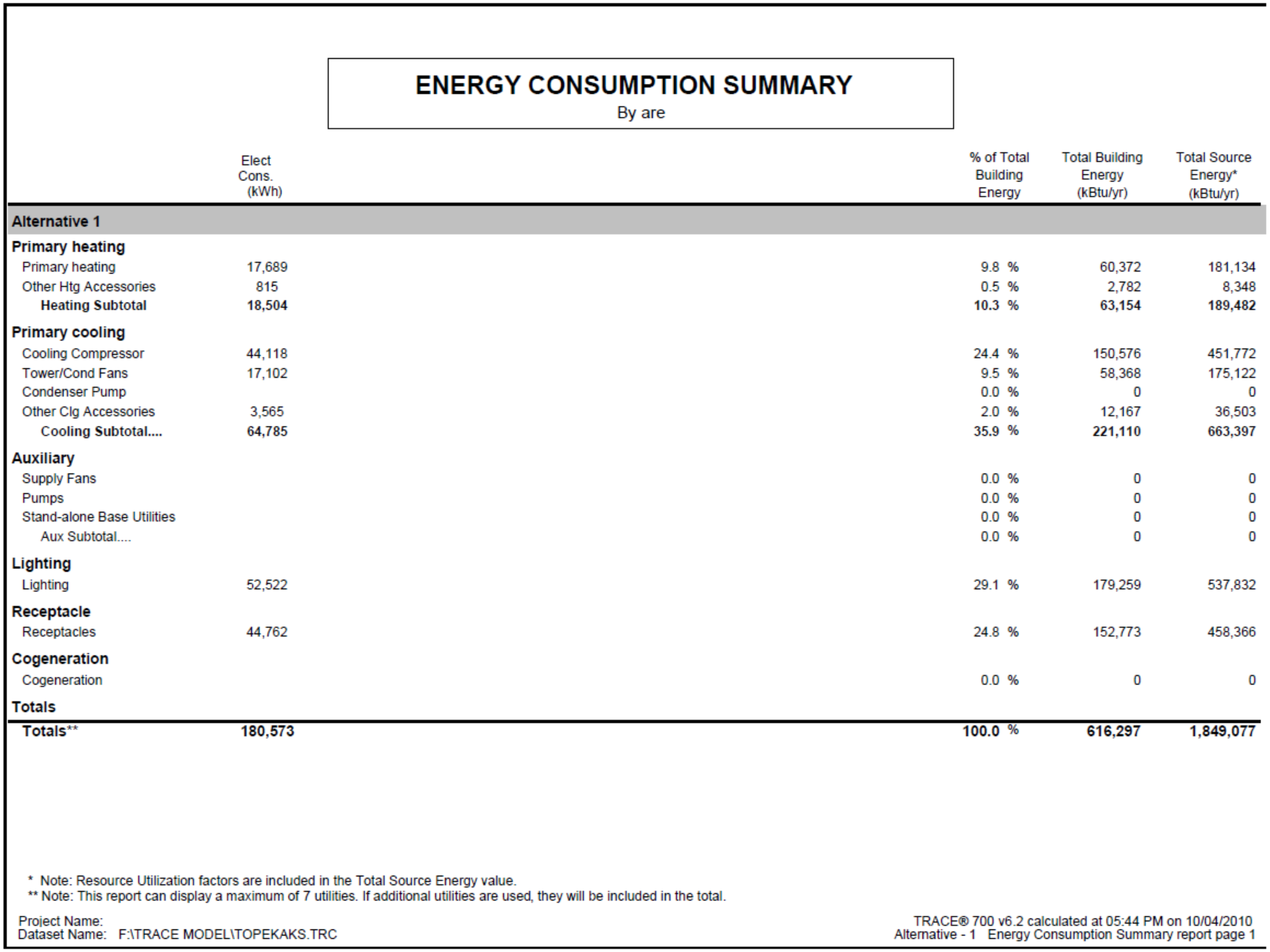
By are

						------- Monthly Utility Costs -------							
Utility	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec	Total
Alternative 1													
Electric													
On-Pk Cons. ($)	1,814	1,140	1,132	1,009	1,215	1,375	1,408	1,508	1,212	1,134	1,084	1,138	15,168
Monthly Total ($):	1,814	1,140	1,132	1,009	1,215	1,375	1,408	1,508	1,212	1,134	1,084	1,138	15,168

Building Area = 16,707 ft²

Utility Cost Per Area = 0.91 $/ft²

Figure F.16 Energy Consumption Summary.



ENERGY CONSUMPTION SUMMARY

By are

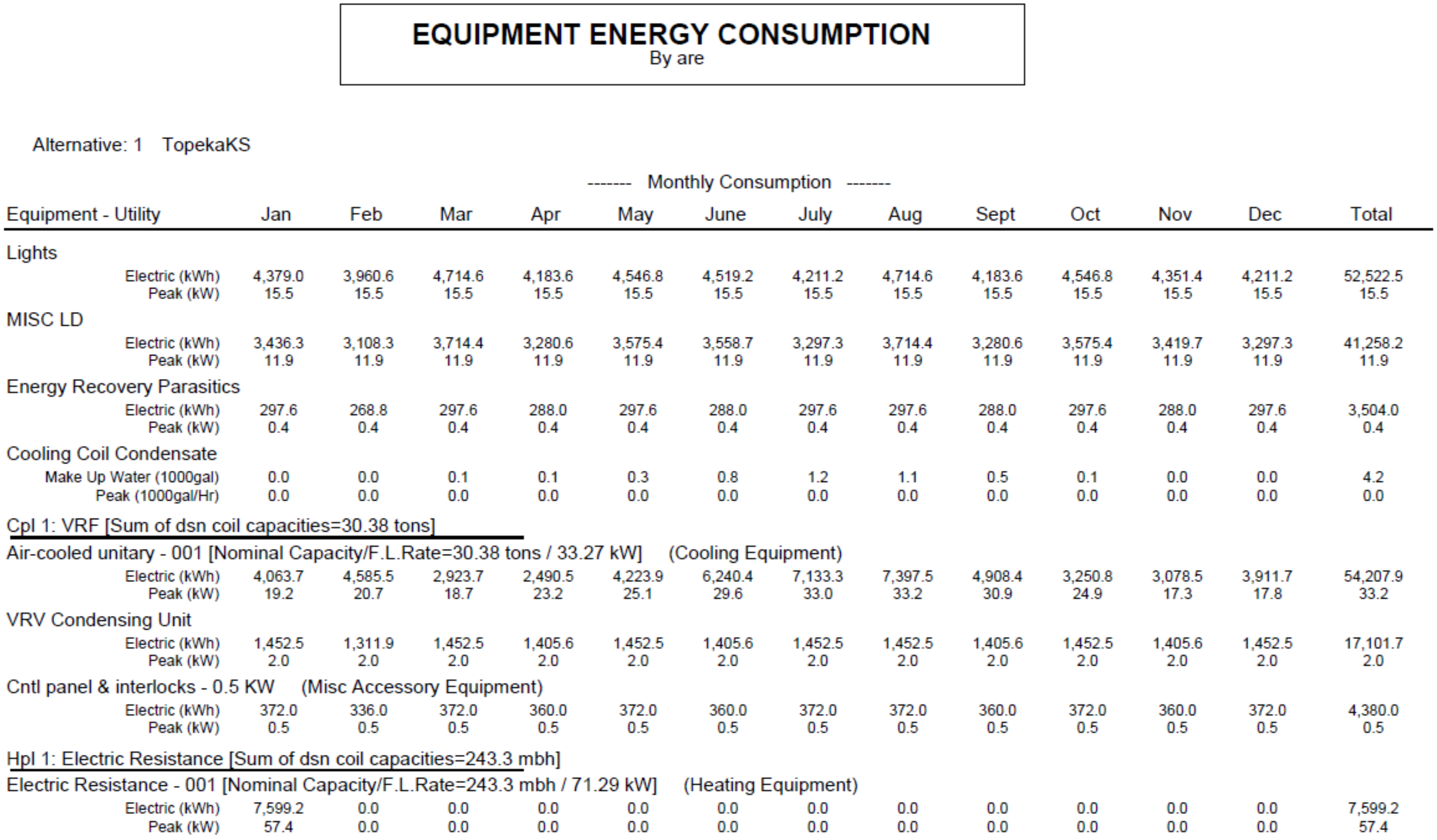
	Elect Cons. (kWh)	% of Total Building Energy	Total Building Energy (kBtu/yr)	Total Source Energy* (kBtu/yr)
Alternative 1				
Primary heating				
Primary heating	17,689	9.8 %	60,372	181,134
Other Htg Accessories	815	0.5 %	2,782	8,348
Heating Subtotal	**18,504**	**10.3 %**	**63,154**	**189,482**
Primary cooling				
Cooling Compressor	44,118	24.4 %	150,576	451,772
Tower/Cond Fans	17,102	9.5 %	58,368	175,122
Condenser Pump		0.0 %	0	0
Other Clg Accessories	3,565	2.0 %	12,167	36,503
Cooling Subtotal....	**64,785**	**35.9 %**	**221,110**	**663,397**
Auxiliary				
Supply Fans		0.0 %	0	0
Pumps		0.0 %	0	0
Stand-alone Base Utilities		0.0 %	0	0
Aux Subtotal....		0.0 %	0	0
Lighting				
Lighting	52,522	29.1 %	179,259	537,832
Receptacle				
Receptacles	44,762	24.8 %	152,773	458,366
Cogeneration				
Cogeneration		0.0 %	0	0
Totals				
Totals**	**180,573**	**100.0 %**	**616,297**	**1,849,077**

* Note: Resource Utilization factors are included in the Total Source Energy value.
** Note: This report can display a maximum of 7 utilities. If additional utilities are used, they will be included in the total.

Project Name:
Dataset Name: F:\TRACE MODEL\TOPEKAKS.TRC

TRACE® 700 v6.2 calculated at 05:44 PM on 10/04/2010
Alternative - 1 Energy Consumption Summary report page 1

Figure F.17 Equipment Energy Consumption.



EQUIPMENT ENERGY CONSUMPTION

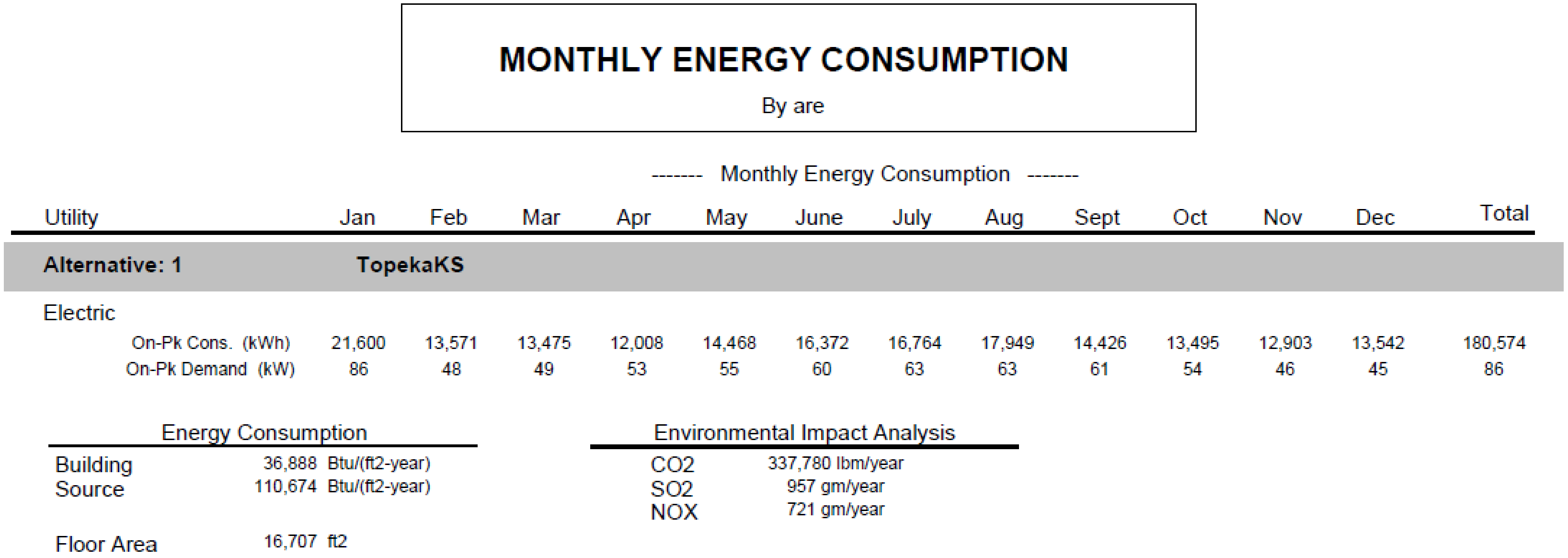
By are

Alternative: 1 TopekaKS

------- Monthly Consumption -------

Equipment - Utility	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec	Total
Lights													
Electric (kWh)	4,379.0	3,960.6	4,714.6	4,183.6	4,546.8	4,519.2	4,211.2	4,714.6	4,183.6	4,546.8	4,351.4	4,211.2	52,522.5
Peak (kW)	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5
MISC LD													
Electric (kWh)	3,436.3	3,108.3	3,714.4	3,280.6	3,575.4	3,558.7	3,297.3	3,714.4	3,280.6	3,575.4	3,419.7	3,297.3	41,258.2
Peak (kW)	11.9	11.9	11.9	11.9	11.9	11.9	11.9	11.9	11.9	11.9	11.9	11.9	11.9
Energy Recovery Parasitics													
Electric (kWh)	297.6	268.8	297.6	288.0	297.6	288.0	297.6	297.6	288.0	297.6	288.0	297.6	3,504.0
Peak (kW)	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4
Cooling Coil Condensate													
Make Up Water (1000gal)	0.0	0.0	0.1	0.1	0.3	0.8	1.2	1.1	0.5	0.1	0.0	0.0	4.2
Peak (1000gal/Hr)	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Cpl 1: VRF [Sum of dsn coil capacities=30.38 tons]													
Air-cooled unitary - 001 [Nominal Capacity/F.L.Rate=30.38 tons / 33.27 kW] (Cooling Equipment)													
Electric (kWh)	4,063.7	4,585.5	2,923.7	2,490.5	4,223.9	6,240.4	7,133.3	7,397.5	4,908.4	3,250.8	3,078.5	3,911.7	54,207.9
Peak (kW)	19.2	20.7	18.7	23.2	25.1	29.6	33.0	33.2	30.9	24.9	17.3	17.8	33.2
VRV Condensing Unit													
Electric (kWh)	1,452.5	1,311.9	1,452.5	1,405.6	1,452.5	1,405.6	1,452.5	1,452.5	1,405.6	1,452.5	1,405.6	1,452.5	17,101.7
Peak (kW)	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0
Cntl panel & interlocks - 0.5 KW (Misc Accessory Equipment)													
Electric (kWh)	372.0	336.0	372.0	360.0	372.0	360.0	372.0	372.0	360.0	372.0	360.0	372.0	4,380.0
Peak (kW)	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Hpl 1: Electric Resistance [Sum of dsn coil capacities=243.3 mbh]													
Electric Resistance - 001 [Nominal Capacity/F.L.Rate=243.3 mbh / 71.29 kW] (Heating Equipment)													
Electric (kWh)	7,599.2	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	7,599.2
Peak (kW)	57.4	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	57.4

Figure F.18 Monthly Energy Consumption.



MONTHLY ENERGY CONSUMPTION

By are

------- Monthly Energy Consumption -------

Utility	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec	Total
Alternative: 1	**TopekaKS**												
Electric													
On-Pk Cons. (kWh)	21,600	13,571	13,475	12,008	14,468	16,372	16,764	17,949	14,426	13,495	12,903	13,542	180,574
On-Pk Demand (kW)	86	48	49	53	55	60	63	63	61	54	46	45	86

Energy Consumption

Building	36,888	Btu/(ft2-year)
Source	110,674	Btu/(ft2-year)
Floor Area	16,707	ft2

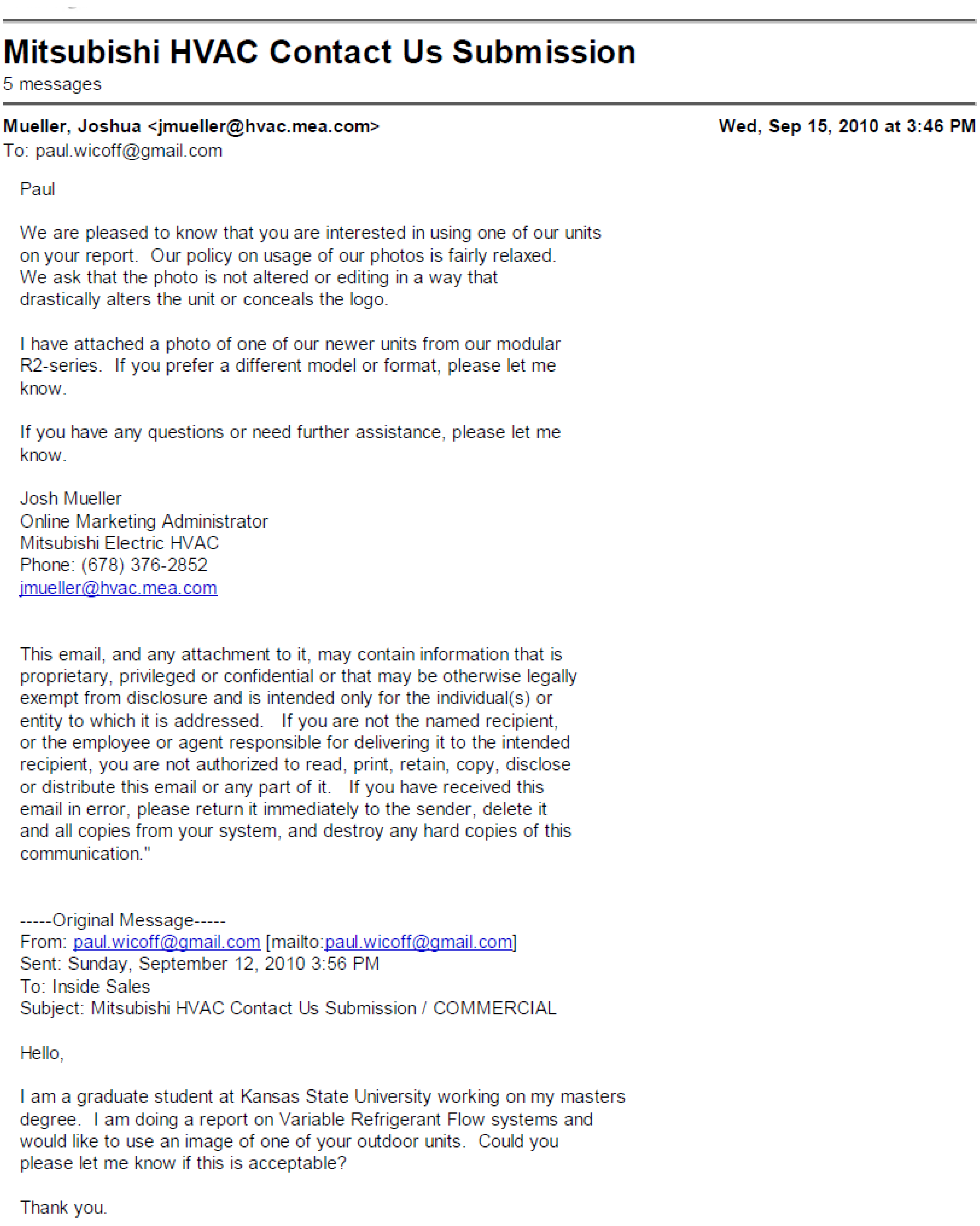
Environmental Impact Analysis

CO2	337,780 lbm/year
SO2	957 gm/year
NOX	721 gm/year

Appendix G - Releases

This appendix includes all written permission to use graphics. This includes permission from the following individuals:

- Joshua Mueller on behalf of Mitsubishi Electric
- Jane Scott on behalf of Daikin AC
- Ammi Amarnath on behalf of EPRI
- Wilson Siah on behalf of the Singapore National Environment Agency



Mitsubishi HVAC Contact Us Submission

5 messages

Mueller, Joshua <jmueller@hvac.mea.com> **Wed, Sep 15, 2010 at 3:46 PM**
To: paul.wicoff@gmail.com

Paul

We are pleased to know that you are interested in using one of our units on your report. Our policy on usage of our photos is fairly relaxed. We ask that the photo is not altered or editing in a way that drastically alters the unit or conceals the logo.

I have attached a photo of one of our newer units from our modular R2-series. If you prefer a different model or format, please let me know.

If you have any questions or need further assistance, please let me know.

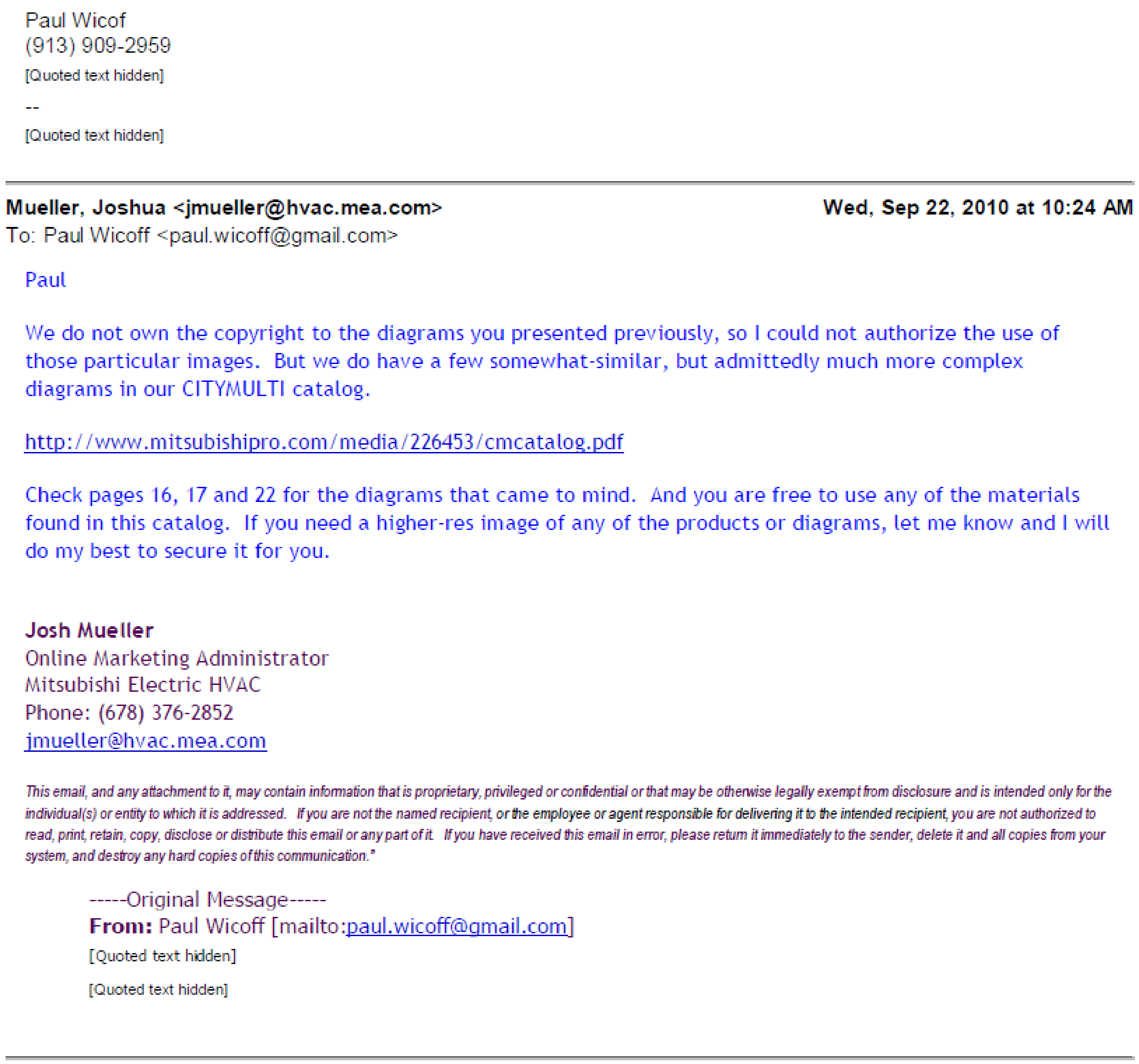
Josh Mueller
Online Marketing Administrator
Mitsubishi Electric HVAC
Phone: (678) 376-2852
jmueller@hvac.mea.com

-----Original Message-----
From: paul.wicoff@gmail.com [mailto:paul.wicoff@gmail.com]
Sent: Sunday, September 12, 2010 3:56 PM
To: Inside Sales
Subject: Mitsubishi HVAC Contact Us Submission / COMMERCIAL

Hello,

I am a graduate student at Kansas State University working on my masters degree. I am doing a report on Variable Refrigerant Flow systems and would like to use an image of one of your outdoor units. Could you please let me know if this is acceptable?

Thank you.



Paul Wicof
(913) 909-2959
[Quoted text hidden]

--

[Quoted text hidden]

Mueller, Joshua <jmueller@hvac.mea.com> **Wed, Sep 22, 2010 at 10:24 AM**
To: Paul Wicoff <paul.wicoff@gmail.com>

Paul

We do not own the copyright to the diagrams you presented previously, so I could not authorize the use of those particular images. But we do have a few somewhat-similar, but admittedly much more complex diagrams in our CITYMULTI catalog.

http://www.mitsubishipro.com/media/226453/cmcatalog.pdf

Check pages 16, 17 and 22 for the diagrams that came to mind. And you are free to use any of the materials found in this catalog. If you need a higher-res image of any of the products or diagrams, let me know and I will do my best to secure it for you.

Josh Mueller
Online Marketing Administrator
Mitsubishi Electric HVAC
Phone: (678) 376-2852
jmueller@hvac.mea.com

> -----Original Message-----
> **From:** Paul Wicoff [mailto:paul.wicoff@gmail.com]
> [Quoted text hidden]
>
> [Quoted text hidden]

Permission to use Daikin images

2 messages

Scott, Jane <jane.scott@daikinac.com> **Thu, Sep 2, 2010 at 7:56 AM**
To: paul.wicoff@gmail.com

Paul – You are welcome to use the images as long as they are labeled, **Printed courtesy of Daikin AC. www.daikinac.com**. Can you send me the photos you want to use?

Thanks

Jane Tetley Scott

Marketing

Media Coordinator

972-512-1956

jane.scott@daikinac.com

image001.jpg
175K

Paul Wicoff <paul.wicoff@gmail.com> **Thu, Sep 2, 2010 at 6:24 PM**
To: "Scott, Jane" <jane.scott@daikinac.com>

Jane,

The images will all be labeled as requested. Please see the attached photos. Note that I have taken all photos of the indoor units and combined them into one graphic.

I do not remember if I asked you, but I would also like to use your image of the HRV system, also attached.

Thank you for the timely response.

Sincerely,

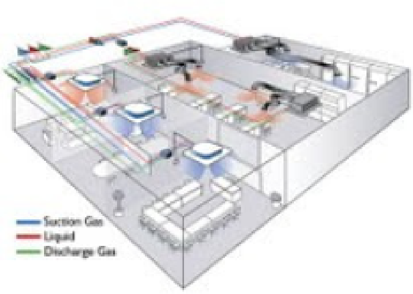
Paul Wicoff

[Quoted text hidden]

--
Paul Wicoff
(913)909-2959

---

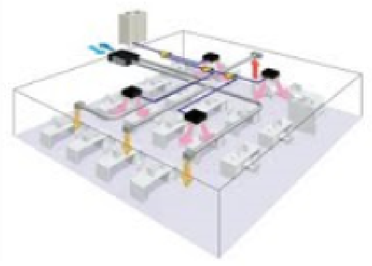
5 attachments



Heat Recovery.jpg
169K

Outdoor Unit 1.png
56K

Outdoor Unit 2.png
74K



HRV.jpg
40K

Indoor Units.jpg
36K

Request for Release

2 messages

Paul Wicoff <paul.wicoff@gmail.com> **Tue, Aug 31, 2010 at 4:31 PM**
To: aamarnath@epri.com

Mr. Amarnath-

I am a graduate student at Kansas State University doing my master's report on variable refrigerant flow. I would like to request the use of Figure 1, showing VRF operation in a 4 zone building, in your article "Variable Refrigerant Flow: Demonstration of Efficient Space Condititioning Technology Using Variable Speed Drives". Please let me know if this is ok.

Thank you for your consideration.

Sincerely,

--
Paul Wicoff
(913)909-2959

Amarnath, Ammi <aamarnath@epri.com> **Tue, Aug 31, 2010 at 5:05 PM**
To: Paul Wicoff <paul.wicoff@gmail.com>

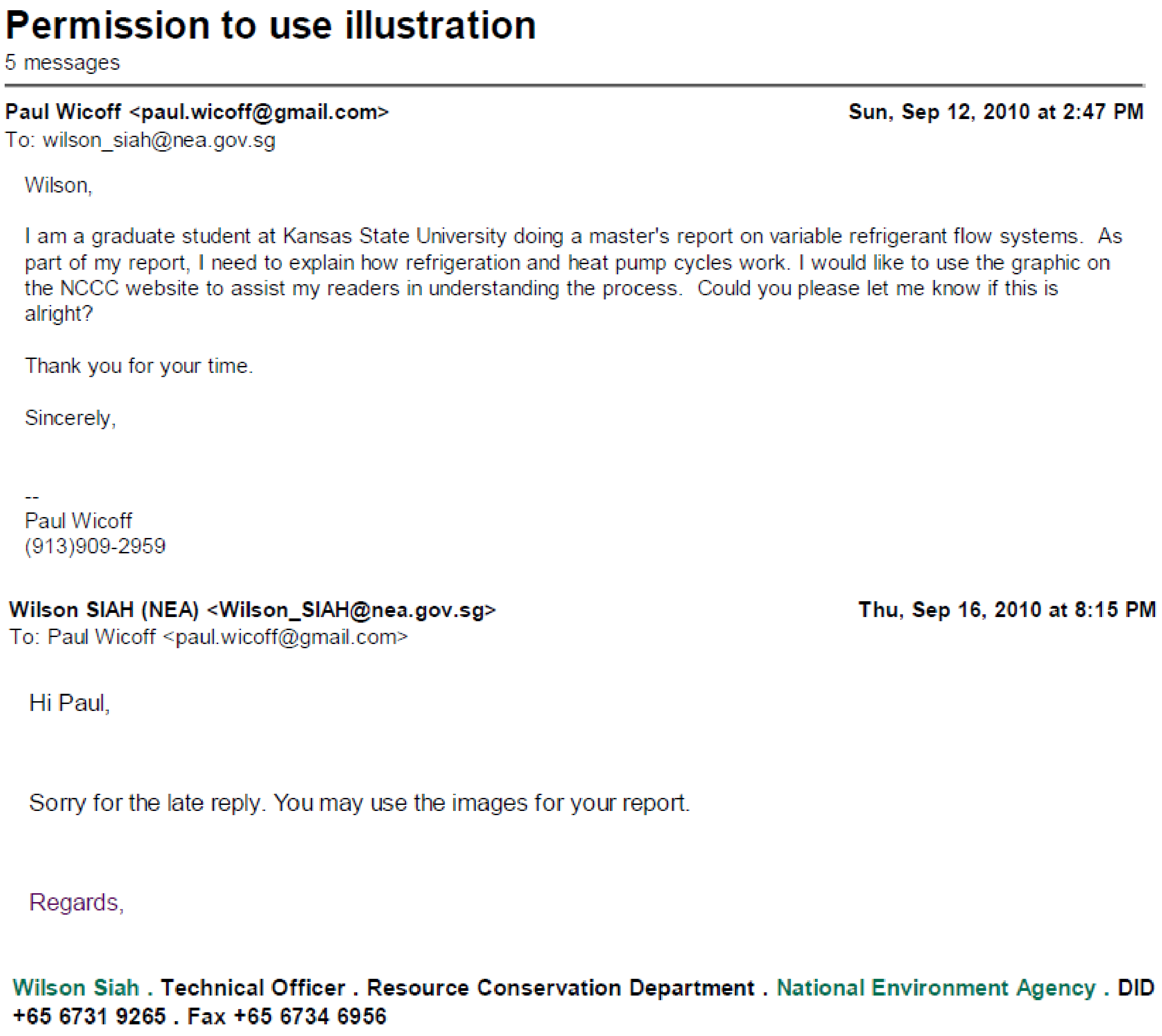
Ok.

Please make sure to give credit to EPRI.

Ammi

From: Paul Wicoff [mailto:paul.wicoff@gmail.com]
Sent: Tuesday, August 31, 2010 2:31 PM
To: Amarnath, Ammi
Subject: Request for Release

[Quoted text hidden]



Permission to use illustration

5 messages

Paul Wicoff <paul.wicoff@gmail.com> **Sun, Sep 12, 2010 at 2:47 PM**

To: wilson_siah@nea.gov.sg

Wilson,

I am a graduate student at Kansas State University doing a master's report on variable refrigerant flow systems. As part of my report, I need to explain how refrigeration and heat pump cycles work. I would like to use the graphic on the NCCC website to assist my readers in understanding the process. Could you please let me know if this is alright?

Thank you for your time.

Sincerely,

--
Paul Wicoff
(913)909-2959

Wilson SIAH (NEA) <Wilson_SIAH@nea.gov.sg> **Thu, Sep 16, 2010 at 8:15 PM**

To: Paul Wicoff <paul.wicoff@gmail.com>

Hi Paul,

Sorry for the late reply. You may use the images for your report.

Regards,

Wilson Siah . Technical Officer . Resource Conservation Department . National Environment Agency . DID +65 6731 9265 . Fax +65 6734 6956