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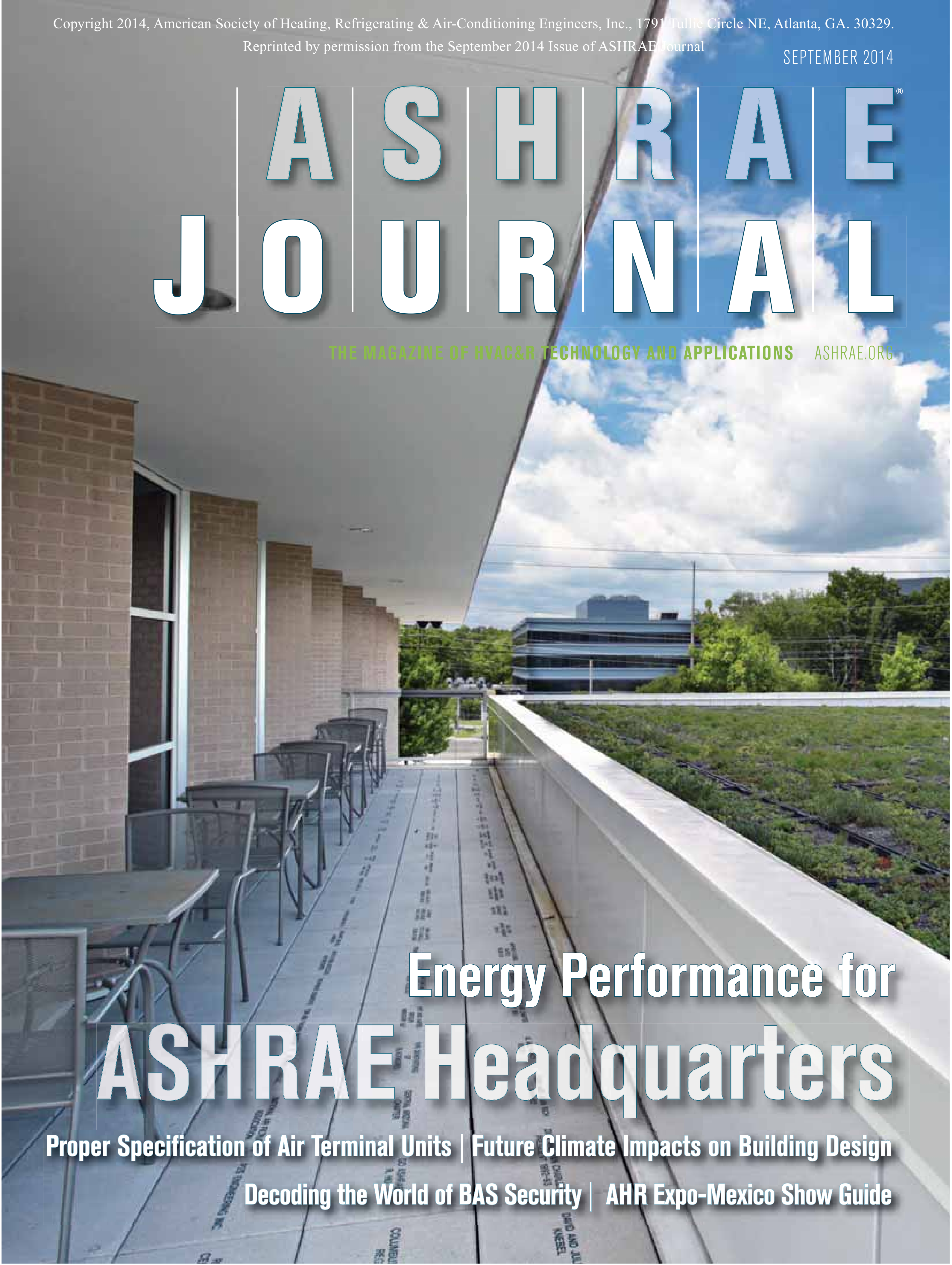

Energy Performance for ASHRAE Headquarters

Proper Specification of Air Terminal Units | Future Climate Impacts on Building Design

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ASHRAE Headquarters, Atlanta, Ga.

Part One

Performance of HVAC Systems at ASHRAE HQ

BY L.E. SOUTHARD, P.E., MEMBER ASHRAE, XIAOBING LIU, PH.D., MEMBER ASHRAE; AND J.D. SPITLER, PH.D., P.E., FELLOW ASHRAE

When ASHRAE headquarters in Atlanta was renovated in 2008, one goal was to create a living lab that could be accessed by members to learn about commercial building performance and state-of-the-art sustainable technology. As a part of this living lab concept, the building uses three separate HVAC systems: a variable refrigerant flow (VRF) system for spaces on the first floor, a ground source heat pump (GSHP) system, primarily for spaces on the second floor, and a dedicated outdoor air system (DOAS), which supplies fresh air to both floors.

Another important aspect of the living lab is the extensive array of sensors that monitor the operation of the HVAC equipment and the conditions in each zone in the building. Both historical and current data from these sensors have been trended via the building automation system and are available to interested parties through the ASHRAE website.

The authors have been researching the relative performance of the VRF and GSHP systems that control temperature in the spaces. This has involved determining the energy consumption of each system (described here) and determining the amount of heating and cooling required by the building (described in a future article.)

The VRF system that provides cooling and heating to the first floor includes two multi-zone inverter driven heat-recovery units. The multi-zone heat-recovery units are connected to a total of 22 fan coil units (FCU) with two speed fans. The cooling capacity of the heat-recovery units is 28 tons (98 kW). Several zones on the first floor are served by three dedicated split systems.

The GSHP system that serves the second floor includes 14 individual water-to-air heat pumps (two 0.75 ton [2.6 kW] units, six 2 ton [7 kW] units and six 3 ton [10.5 kW] units) connected to a ground loop consisting of 12,400 ft (122 m) deep vertical boreholes, for a total of 31.5 tons (111 kW) of cooling capacity. The heat pumps have variable speed fans (driven with electronically commutated motors) with three selected speeds.

The DOAS includes six staged air-cooled condensing units to provide cooling and two heat recovery wheels

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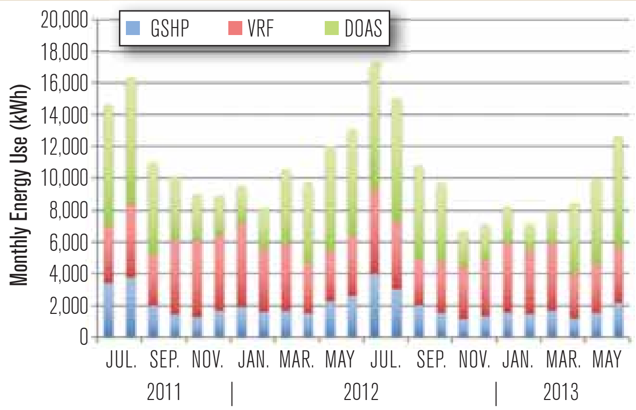

FIGURE 1 Total monthly energy use by each HVAC system.

The dedicated outdoor air system.

to precool or preheat the outdoor air. The total cooling capacity of the condensing units is 28.6 tons (100.6 kW).

Two years of data relating to the operation of the different HVAC systems have been collected and analyzed in an attempt to evaluate the performance of the systems. These data cover the time span from July 1, 2011 through June 30, 2013. Data points that have been collected include operating mode (off/heat/cool), zone temperature and discharge air temperature for each individual FCU or heat pump. Ground loop supply and return water temperatures and flow rate were also collected for the GSHP system. For the DOAS, the flow rate of the supply air to each floor and the supply and return air temperatures and humidity levels were collected.

Metered energy used by each system was also collected. For the GSHP system, the power that is metered and recorded includes the power for all 14 heat pumps as well as the ground loop water circulation pumps. For the VRF system, the power that is metered and recorded is only the power for the two heat-recovery units and the 22 FCUs that are connected to them. The power for the three dedicated split systems is metered through a different panel that also includes the power for computer servers and other equipment in the computer room.

Figure 1 shows the monthly energy use by each system. These raw data indicate that the VRF system used twice as much energy as the GSHP system over the two-year time span. However, it is of great interest to the HVAC industry to know what caused such significant differences in the energy use of the two systems. The energy consumptions are affected by several factors including:

- The heating and cooling loads of the conditioned floor spaces;
- The control strategies of the two systems; and
- The operating conditions and operational efficiencies of the two systems.

The characteristics and contributions of each of these factors will be briefly discussed in this article. More detailed information will be provided in successive articles and in a final report.

Different Loads

The GSHP system serves 15,558 ft^2 (1445 m^2) of office and meeting space primarily on the second floor with a normal occupancy of 60 people. The VRF units for which power measurements are available serve a total of 17,213 ft^2 (1559 m^2) on the first floor, which includes offices, large meeting spaces and storage areas. The normal occupancy of the area served by the VRF system is 43 people. The areas served by both systems had the same measured average combined lighting and plug load density of 0.45 W/ft^2 (4.8 W/m^2) for the two-year study period.

The DOAS, which conditions outdoor air to 55°F (13°C), satisfied part of the cooling load in summer, but contributed to the heating load in winter. The average DOAS airflow rate to the first floor was 2,560 cfm (1208 L/s), which is significantly higher than the average flow rate to the second floor of 1,480 cfm (699 L/s). In accordance with ASHRAE/IES Standard 90.1, which requires supply air temperature to be reset in response to building loads or outdoor air temperature, the DOAS sequence of operations includes a provision for the supply air temperature to be reset to 60°F (16°C) if all space temperatures are below their cooling setpoints and the outside air enthalpy is below a minimum threshold. It also includes a provision to raise the supply air temperature to 65°F (18°C) if 80% of the zone temperatures are below their heating setpoints.

FIGURE 2 Normalized monthly energy use per square foot.

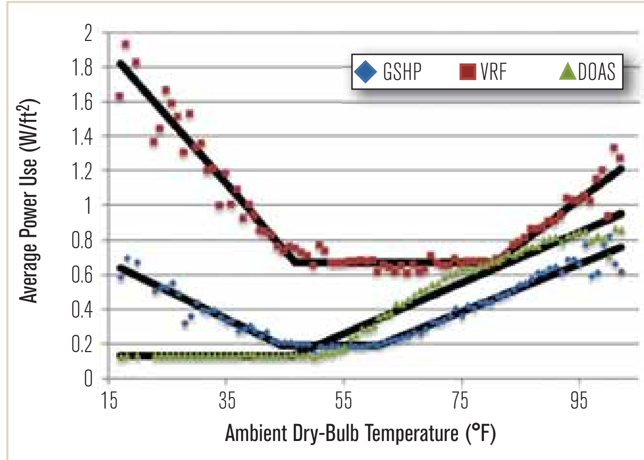

FIGURE 3 Average power use vs. ambient temperature with change-point models.

The measured power consumptions of the GSHP and VRF systems were normalized with the floor space conditioned by each system. As shown in *Figure 2,* the normalized energy use (kWh/ft^2) of the GSHP system over the two-year period is 44% less than that of the VRF system.

If the cooling and heating output that is provided by each system could be measured, then a COP or EER could be used to compare the systems. Unfortunately, it was not feasible to install the amount of instrumentation (temperature, humidity and airflow sensors for the discharge air, return air, and the outdoor air in every zone) necessary to measure the cooling or heating provided by each of the systems. Estimation of the cooling and heating output will be discussed in the next article.

Different Control Strategies

Energy use for both the GSHP and VRF systems peaks in the summer cooling season, but the VRF system shows unexpectedly high energy use during the winter as well as the fall and spring shoulder seasons, which in Atlanta can still have days when a substantial amount of cooling is needed.

Figure 3 shows instantaneous power usage for all three systems during occupied building hours (7 a.m. to 6 p.m. on workdays) averaged for each 1°F (0.55°C) outdoor air temperature bin and normalized by the floor area served by each system. The VRF system shows unexpectedly high power use at times with mild temperatures. The normalized instantaneous power use of the three systems was correlated to the coincident ambient air dry-bulb temperature using a change-point regression model.[1]

The condensing units for the DOAS system.

As shown in *Figure 3,* the minimum power use (i.e. the horizontal portion of the change point regression model) for the GSHP system is 0.19 W/ft^2 (2 W/m^2) over a temperature range of 44°F to 61°F (6.6°C to 16°C). The minimum power use for the VRF system is 0.67 W/ft^2 (7.2 W/m^2) over a much wider range of 47°F to 81°F (8°C to 27°C). The minimum power use for the DOAS is 0.13 W/ft^2 (1.4 W/m^2), and the change-point occurs at 46°F (7.7°C). Blower power use for the GSHP system when all heat pumps are running in ventilation mode is about 0.06 W/ft^2 (0.65 W/m^2). For the VRF system, few data points are available when all FCUs are running in ventilation mode, but blower power use may be as high as 0.15 W/ft^2 (1.6 W/m^2). The VRF system consumed more than three times as much power as the GSHP system when the ambient air temperature was colder than 61°F (16°C). It consumed 50% more power than the GSHP system

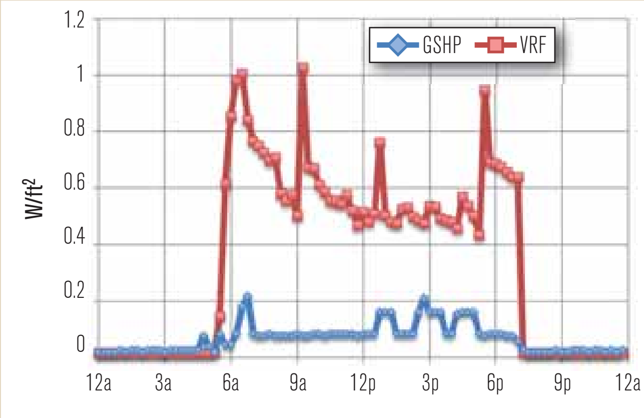

FIGURE 4 GSHP and VRF power use on April 3, 2013.

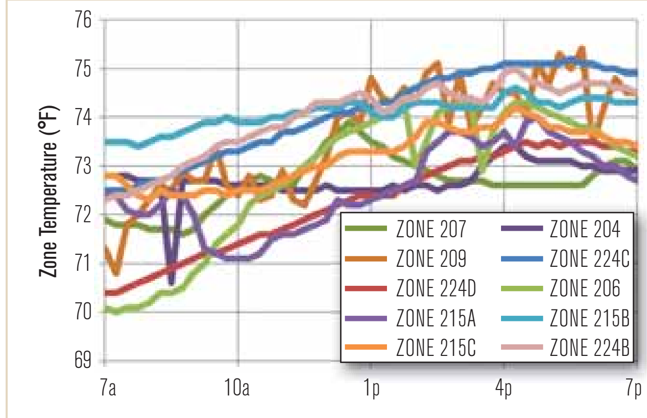

FIGURE 5 GSHP zone temperatures on April 3, 2013.

when the ambient air temperature peaked at about 100°F (38°C). The DOAS cooled the outdoor air to 55°F (13°C), and its power use increased linearly as the ambient air became warmer.

The different power use in mild weather appears to result from the control strategies of the two systems and the interactions with the DOAS. Throughout the building the thermostats have BAS-specified base setpoints that the occupants can adjust ±3°F (1.67°C) to suit individual comfort levels. When the weather is mild, the fresh air supplied by the DOAS is adequate to maintain most of the zones on the second floor within the heating and cooling setpoints for the GSHP system (typically 68°F and 74°F [20°C and 23°C], respectively). As a result, few heat pumps compressors operated then, with most heat pumps running in ventilation mode. However, during the same time periods, a much higher proportion of FCUs in the VRF system were on with some of the units operating in cooling mode while others ran in heating mode.

Each zone in the VRF system has a single setpoint, which according to the manufacturer, is valid for the current operation mode. We do not have complete information about how the control strategy works, but our interpretation is that an FCU can run in one mode, maintaining a temperature within about ±1°F (0.55°C) until such time as the temperature moves a certain amount away from the setpoint in the opposite direction from the system's operation in the current mode. For example, if the system is in heating mode and the zone rises about 4.5°F (2.5°C) above the setpoint temperature, the FCU will change to cooling mode, and bring the zone temperature back to within ±1°F (0.55°C) of setpoint.

The open office floor plan presents challenges for control schemes based on a single set point for each zone.

We welcome input from VRF experts on this point. As to whether or not this control strategy is consistent with Standard 90.1 requirements of at least a 5°F (3°C) deadband between heating and cooling setpoints we leave for others to judge. The current control strategy does seem to prevent any single unit from switching back and forth between modes, but it does not prevent adjacent FCUs in the open plan office space from "fighting" each other. Example 1 illustrates this situation.

Example 1

On Wednesday, April 3, 2013 ambient temperatures were cool with a morning low of 43°F (6°C) and an afternoon high of 63°F (17°C). *Figure 4* shows that the power use by the VRF system was much higher than the power use by the GSHP system during this day. Only four of the heat pumps ran during the workday: two heat pumps operated in heating mode for five minutes each, and two operated in cooling mode for several hours. The

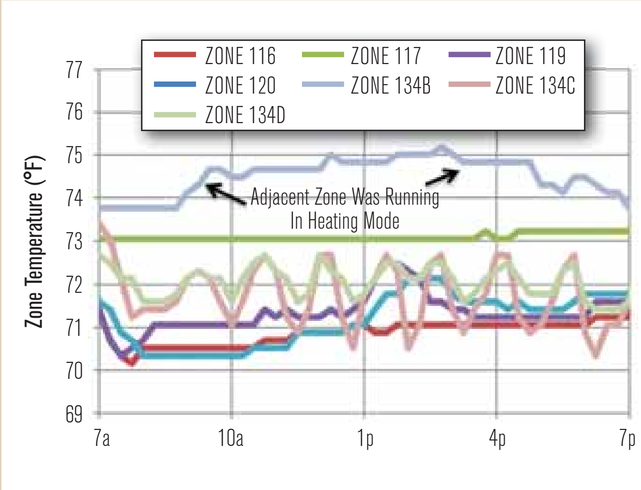

FIGURE 6 VRF zone temperatures for units running in cooling mode on April 3, 2013.

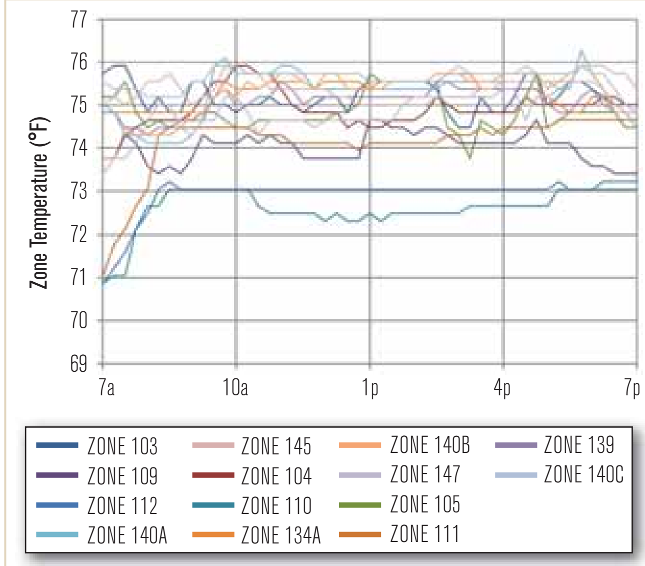

FIGURE 7 VRF zone temperatures for units running in heating mode on April 3, 2013.

zone temperatures in the other 10 zones floated between 70°F and 75°F (21°C and 24°C) during occupied hours as shown in *Figure 5*.

Meanwhile, all 22 of the VRF FCUs ran, 14 exclusively in heating mode and 8 exclusively in cooling mode. *Figures 6* and *7* show that the zone temperatures in the zones with FCUs operating in heating mode were generally maintained between 74°F and 76°F (23°C and 24.4°C), while in the zones with FCUs operating in cooling mode temperatures were usually between 70°F and 73°F (21°C and 22.7°C) during occupied hours. At first, this may seem counterintuitive, but the FCUs for zones with lower setpoints (72°F [22.2°C]) ran in cooling mode to satisfy the cooling demands of those zones, while the FCUs for zones with higher setpoints (74°F [23°C]) ran in heating mode to satisfy the heating demands of their zones. The zone temperatures show that the FCUs were meeting the demands of their specific zones.

This example demonstrates the energy expense associated with trying to maintain each individual zone temperature at a single independent setpoint by the VRF system. It is not clear whether the precise temperature control in each individual zone offers any benefits of thermal comfort. A thermal comfort survey may be necessary to answer this question.

Given the precise temperature control offered by the VRF system, the interaction between the DOAS and the VRF systems is sensitive and can cause some FCUs to run in heating mode even on hot days. The next example illustrates such operation.

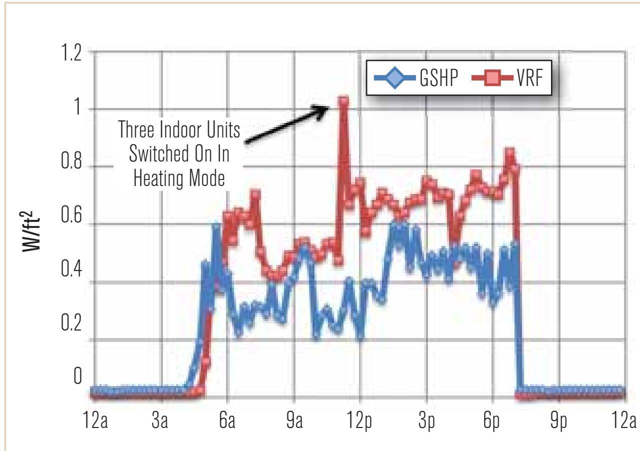

FIGURE 8 GSHP and VRF system power use on June 14, 2013.

Example 2

On Friday, June 14, 2013 the ambient temperatures were warm with a morning low of 68°F (20°C) and an afternoon high of 86°F (30°C).

Ten of the 14 heat pumps in the GSHP system operated intermittently in cooling mode for an average of 5.5 hours each during the workday. Meanwhile, all 22 of the FCUs in the VRF system ran. Eleven of the FCUs operated in cooling mode for the entire time when the building was occupied between 6:45 a.m. and 6:45 p.m. Six other FCUs operated intermittently in cooling mode, four FCUs operated in heating mode for a short period in the morning and in cooling mode later in the day, and the FCU for the library operated in heating mode only for a

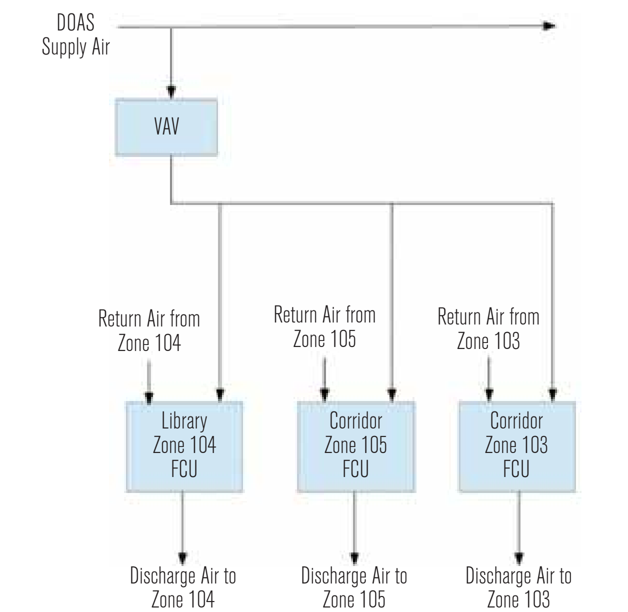

FIGURE 9 Diagram of DOAS supply to Zone 104.

TABLE 1 Sequence of events on June 14, 2013.

TIME	EVENT
11:00 a.m.	Library FCU blower is running in ventilation mode. Coils are not in use. Discharge air temperature is 65°F, zone temperature is 73.8°F. Zone setpoint is 74°F.
11:08 a.m.	FCU for a corridor zone turns on in cooling mode.
11:15 a.m.	Library coils are still not in use. Blower is still running in ventilation mode. Discharge air temperature is now 56°F, zone temperature is 73.6°F. Total VAV airflow to the 3 zones has not changed. It's likely that the balance of fresh air to each zone has changed with less DOAS airflow going to the corridor zone and more going to the library.
11:16 a.m.	Library FCU turns on in heating mode.
11:30 a.m. to 12:00 p.m.	Library discharge air temperature is 92°F – 94°F. Zone temperature is 73.2°F – 73.6°F.
12:04 p.m.	Library FCU turns off.
12:15 p.m.	Library discharge air temperature is 65°F, zone temperature is 73.8°F.

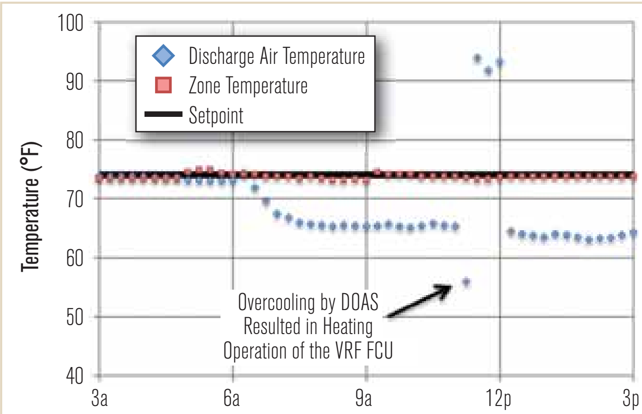

FIGURE 10 Library zone temperatures on June 14, 2013.

short time period. *Figure 8* (Page 19) shows the power use by each system during the day.

The operation of the library FCU in heating mode is of particular interest. Drawings of the ductwork for the building show that a single DOAS VAV terminal supplies fresh air to three different zones: the library and two corridors. For each of these zones, the 55°F (13°C) air from the DOAS mixes with the return air and flows through the FCU duct to the zone as shown in *Figure 9*. The sequence of events that led to the library FCU operating in heating mode is described in *Table 1*. *Figure 10* shows the discharge air temperature, zone temperature and system setpoint for the library during the day.

This is just one example of how the interactions between the DOAS and the VRF systems can create a need for simultaneous heating and cooling that is not caused by inherent internal or building envelope loads. In this case, the VRF system is serving to "reheat" the cool air provided by the DOAS.

Both the GSHP system and the VRF system can provide simultaneous heating and cooling to various zones of the building. *Figures 11* and *12* show the contributions of heating and cooling operations to the total VRF and GSHP system power use. Power uses for the heating and cooling operations were approximately allocated based on the nominal capacity of the FCUs or the heat pumps running in heating and cooling modes, respectively. These figures show that when ambient temperatures are between 50°F and 70°F (10°C and 21°C) the GSHP system uses less than 0.3 W/ft^2 (3 W/m^2), primarily for cooling. In the same range, the VRF system uses over 0.6 W/ft^2 (6 W/m^2) with much of the power being used for heating.

Although the heat-recovery type VRF system can make use of otherwise wasted condensing/evaporating energy to provide space heating and cooling to different zones without consuming additional power to run multiple compressors (as the GSHP system does), the longer run-times and conflicting heating and cooling operations in adjacent zones in the open office environment due to the single setpoint control (as shown in Example 1) resulted in higher power use than the GSHP system when the weather was mild.

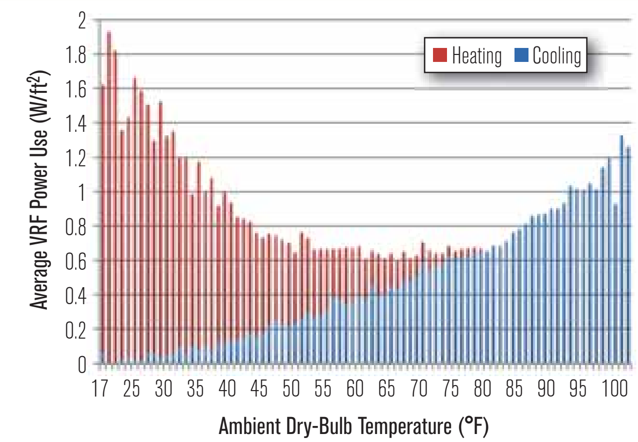

FIGURE 11 Contributions of heating and cooling to VRF system power use.

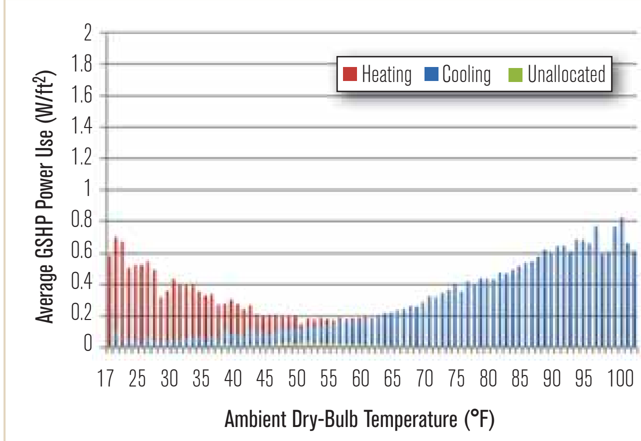

FIGURE 12 Contributions of heating and cooling to GSHP system power use.

Different Operating Conditions and Operational Efficiencies

Another difference between the GSHP system and the air-source VRF system lays in the heat sink and source—ground vs. ambient air. The ground loop supply temperature and the ambient air temperature are the key parameters affecting the operational efficiency of the GSHP and VRF systems, respectively. As shown in *Figure 13*, the ground loop supply temperatures are more favorable than the ambient air temperature for the operation of the vapor compression cycle—lower when cooling is needed and warmer when heating is demanded. The temperature differential between the ground loop supply temperature and the ambient air temperature is much larger in winter than in the summer, which indicates larger energy efficiency advantages of the GSHP system in the wintertime.

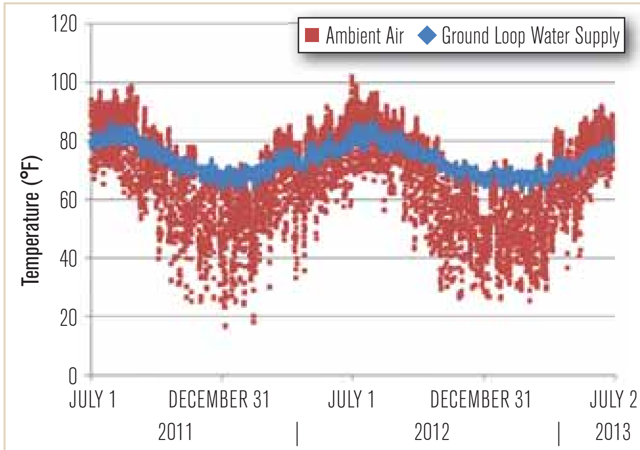

FIGURE 13 Ground loop supply temperature vs. ambient temperature.

TABLE 2 Average operating source temperatures and catalog performance.

	VRF			GSHP*		
	MID 90% SOURCE (AIR) TEMPERATURE RANGE, °F	MEDIAN SOURCE (AIR) TEMPERATURE, °F	COP	MID 90% SOURCE (WATER) TEMPERATURE RANGE, °F	MEDIAN SOURCE (WATER) TEMPERATURE, °F	COP
Cooling	42 – 89	67	5.9	68 – 83	75	6.1 – 6.4
Heating	35 – 76	57	4.5	65 – 71	68	5.6 – 5.8

*GSHP COPs are for the first stage of operation; the range represents different units.

Table 2 shows manufacturers' data for the cooling and heating efficiency of the VRF system and the GSHP equipment at source temperatures. While it is difficult to directly compare systems that use different sources, under the conditions that they are operating at, the heat pumps ran in a much narrower range of source temperatures than the VRF system and have higher efficiency than the VRF system at most conditions. While the cooling efficiency of the GSHP equipment is only moderately higher than that of the VRF system, the GSHP equipment has much higher heating efficiency than the VRF system due to more favorable operating conditions supplied by the ground loop.

Note that these efficiencies are for manufacturers' rated performance and do not take into account the pumping power required for the GSHP system nor the part load effects on the VRF system. In contrast, the metered power data that this article has presented include all of the operational power used by each system.

By filtering the data to include only hours with no VRF units operating in heating mode, the effects of

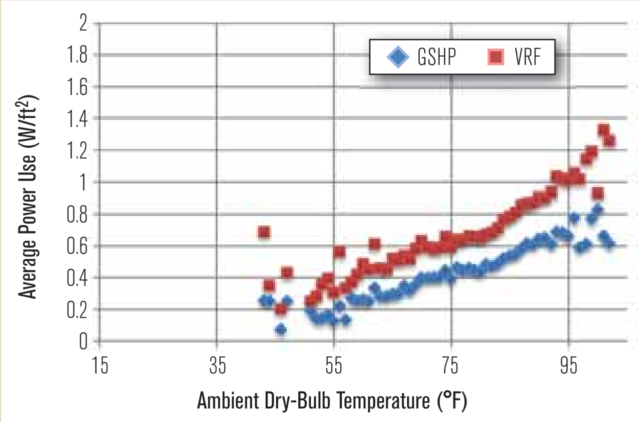

FIGURE 14 Average power use vs. ambient temperature for cooling only.

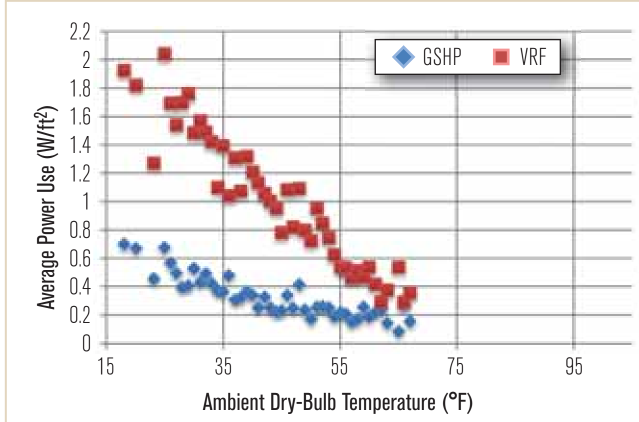

FIGURE 15 Average power use vs. ambient temperature for heating only.

having different units running in heating and cooling modes simultaneously can be eliminated; although zones may still have different setpoints. This reduced set of data points was again grouped into 1°F (0.55°C) temperature bins and the average power use was calculated for each system for the set of data points in each temperature bin. *Figure 14* shows that when simultaneous heating and cooling is eliminated the amount of power used by the VRF system is about 30% higher than the amount used by the GSHP system. *Figure 15* shows the same analysis for data points when no VRF units operate in cooling mode. For these heating-only data points, VRF system power use is more than double GSHP system power use.

One of the 14-ton VRF heat recovery units mounted on the roof of the ASHRAE headquarters building.

Conclusions

The ASHRAE headquarters living lab is a valuable resource of information regarding the real performance of high efficiency HVAC systems in an operational office building environment. The efforts described in this article have barely touched the surface of the vast opportunities that are available to researchers through this resource.

During the two-year period that this study encompassed the GSHP system used about 20% and 60% less energy than the VRF system in the summer and winter/shoulder seasons, respectively, while maintaining similar zone temperatures. Factors contributing to the differences in energy use include:

Ground loop water supply temperatures were more favorable than ambient air temperatures for heat pump operation. This allows the GSHP equipment to operate at higher efficiencies.

The control strategy of the VRF system resulted in longer runtimes than the GSHP system, especially in mild weather. These longer runtimes coincided with significant amounts of simultaneous cooling and heating in adjacent spaces.

Other factors, specifically the differences in heating loads and cooling loads will be considered in the next article.

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1. Kissock, K., J. Haberl, and D. Claridge. 2002. Development of a toolkit for calculating linear, change-point linear and multiple-linear inverse building energy analysis models (RP-1050). ASHRAE Research Project, *Final Report.*

Acknowledgments

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One of two ClimateMaster Tranquility® Console (TRC) Series units installed in ASHRAE headquarters.

A sign on a door on the second floor of ASHRAE headquarters, which is conditioned by a ClimateMaster ground-source heat pump system.

A television informs visitors to ASHRAE headquarters about ASHRAE's ground-source heat pump system.

Part of the second floor of ASHRAE headquarters that is conditioned by a ClimateMaster ground-source heat pump system.

The exterior of ASHRAE headquarters

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First floor.

Second floor.

Part Two

Performance of HVAC Systems at ASHRAE HQ

BY L.E. SOUTHARD, P.E., MEMBER ASHRAE; XIAOBING LIU, PH.D., MEMBER ASHRAE; AND J.D. SPITLER, PH.D., P.E., FELLOW ASHRAE

When the ASHRAE headquarters building in Atlanta was renovated in 2008, a variable refrigerant flow (VRF) system was installed to provide conditioning for spaces on the first floor, while a ground source heat pump (GSHP) system was installed, primarily for spaces on the second floor. Details about these two systems are available in previous articles.[1,2] Data relating to the operation of the different HVAC systems have been collected and analyzed for the two-year time span from July 1, 2011 through June 30, 2013 in an attempt to evaluate the performance of the systems.

As we showed in our previous article,[2] during the two-year study period, the space-averaged annual energy use of the GSHP system was 1.5 kWh/ft^2·yr (17 kWh/m^2·yr) while the space-averaged annual energy use of the VRF system was 2.7 kWh/ft^2·yr (30 kWh/m^2·yr). As previously discussed, the GSHP serves all of the second floor, as well as a small stairwell on the first floor. The VRF system for which power measurements are available serves all of the first floor except for the vestibule, reception area, stairwells and computer equipment room. For both systems, the areas that are served are primarily office and meeting space; although a larger fraction of the space on the first floor is meeting rooms, which are used infrequently. During

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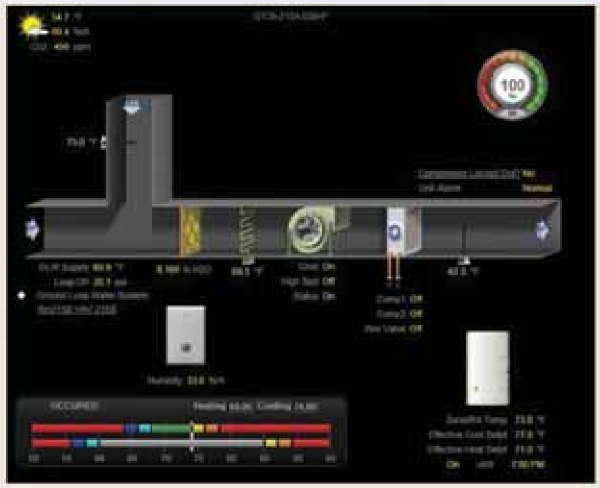
Zone 215A.



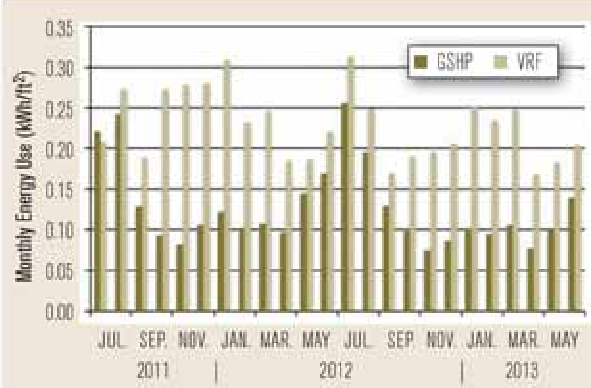
Zone 215B.

the two-year study period the median monthly use of the meeting room in the new first floor addition was 26.5 hours/month and of any of the smaller rooms in the renovated part of the first floor was four hours/month. *Figure 1* shows the monthly energy use of each system. Different zone temperature control strategies and different equipment efficiencies at the source operating temperatures account for some of the difference in energy use between the two systems,[2] but the critical question is, how much conditioning is provided by each of the two systems?

In this article, we first estimate the cooling and heating provided by both the GSHP and VRF systems based on experimental measurements between July 2011 and March 2012. We then present system COPs and EERs for both systems. We also estimate the cooling and heating provided, and the system COPs and EERs for the GSHP system for April 2012 to June 2013.

Beginning in April 2012, runtime fractions for many of the VRF system fan coil units (FCUs) increased dramatically with cooler discharge air temperatures, while zone temperatures remained steady. The FCUs have two-speed fans with the higher speed used during fan coil operation (with heating/cooling output) and the lower speed used for ventilation mode (without any heating/cooling output). With unchanged zone loads, this increase in runtime and decrease in discharge temperatures led us to conclude that discharge flow rates during FCU operation must have decreased. ASHRAE personnel indicated that the manufacturer had replaced the control boards in 21 of the 22 FCUs

FIGURE 1 Monthly energy use by the GSHP and VRF systems.



on April 14 and 15, 2012. It seems likely that at the time of the control board replacement, the flow rates of the discharge air changed, but since there has been no subsequent testing and balancing, the new values are unknown. Spot measurements taken during a site visit confirm that airflows from the FCUs during fan coil operation are lower than the measurements taken during the initial testing and balancing. For this reason, the heating and cooling provided by the VRF system could not be estimated for dates after the equipment modifications.

Experimental Measurements

Figure 2 shows the airflows entering and exiting the heat pumps and 14 of the 22 VRF fan coil units. Outside air from the dedicated outdoor air system (DOAS) is ducted to a plenum box where it mixes with return

FIGURE 2 Airflow configuration of terminal units.

air from the plenum. For the other eight VRF fan coil units, outside air is provided directly from the DOAS to the zone without passing through the FCUs. *Table 1* shows the measurements that are available for the different units.

As shown in *Table 1*, one zone (215B), which is served by a heat pump, is instrumented more heavily than the other zones. With temperature and humidity sensors on both the entering air and the discharge air, and an airflow meter, all of the necessary measurements are available to calculate the heating and cooling provided to the zone.

The heating that is provided to each zone can be calculated as:

$$q = \dot{m} c_p \Delta T \quad (1)$$

Thus, the temperature differential and airflow rate are all that is needed. For cooling there is a latent load, so the cooling that is provided must be calculated from:

$$q = \dot{m} \Delta h \quad (2)$$

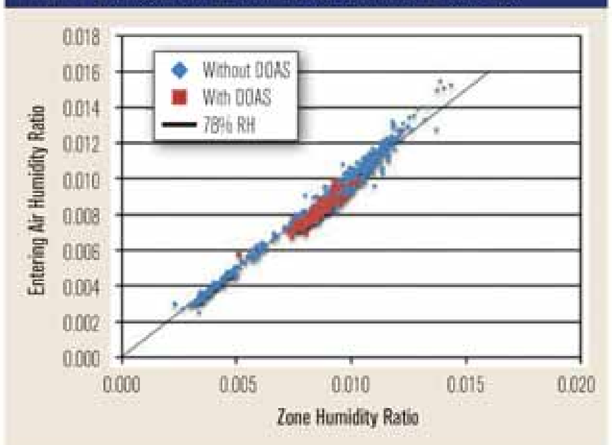
where Δh is the enthalpy differential between the entering air and the discharge air, and data for humidity levels are necessary. For the remaining heat pumps, only entering air, discharge air and zone temperatures, and zone humidity are measured. The flow rate of the discharge air and the entering air and discharge air humidity levels have to be estimated. The flow rates for the discharge air (when the heat pump operates at various modes) that are listed in the building renovation design documents and the testing and balancing report were assumed to be valid for all of the other zones. For Zone 215B, the average airflow rate is within 2% of the flow rate listed in the testing and balancing report.

As can be seen from *Figure 3*, for Zone 215B, the entering air humidity ratio was found to be closely related to the zone air humidity ratio. In fact, the mixed air humidity ratio is a little higher than the zone humidity ratio when the zone humidity is high and a little lower

TABLE 1 Available measurements.

	HEAT PUMPS		VRF SYSTEM
	ZONE 215B	OTHER HEAT PUMPS	
DISCHARGE AIRFLOW	Available	N/A	N/A
DISCHARGE AIR TEMPERATURE	Available	Available	Available
DISCHARGE AIR HUMIDITY	Available	N/A	N/A
ENTERING AIR TEMPERATURE	Available	Available	N/A
ENTERING AIR HUMIDITY	Available	N/A	N/A
ZONE TEMPERATURE	Available	Available	Available
ZONE HUMIDITY	Available	Available	Available

FIGURE 3 Entering air and zone humidity ratio relationship for zone 215B.



when the zone humidity is low. Because the zone air dew points are already low (close to that of the OA supplied by the DOAS) whenever the DOAS is running, the outdoor air from the DOAS has little effect on the entering air humidity ratio. A linear correlation was fitted and used to estimate entering air humidity for the remaining zones.

Likewise, the discharge air humidity ratio and temperature for cooling operation were plotted for Zone 215B, as shown in *Figure 4*. Analysis of these data showed that the relative humidity was nearly constant at 78%, so for the remaining zones, the discharge air relative humidity was approximated to be 78% for cooling operation.

For the zones that are conditioned by the VRF system, the only measured data are discharge air temperature and zone conditions. Since the FCUs have two-speed fans with a single high speed used during fan coil operation and a low speed for ventilation mode, the flow rates for the discharge air during fan coil operation were estimated to be those listed in the testing and balancing report. The entering air temperature is not measured, so it was estimated to be the same as the zone temperature. For eight of the VRF zones, the outdoor air is provided directly to the zone, so this approximation should be reasonably close. For the other 14 zones, during morning warm-up or cooldown operation the DOAS is shut off and, again, this approximation should be good. However, when the building is occupied, preconditioned outdoor air from the DOAS is mixed with the return air from these zones and this assumption will cause the estimates of cooling provided to these 14 zones to be slightly high, and the estimates of heating provided to be slightly low. For estimating cooling provided, when data for humidity levels is needed, entering air humidity was again estimated using the same correlation that was used for the zones in the GSHP system. Since humidity levels leaving the VRF system FCUs are not measured, we have taken the manufacturer's data to create a map of sensible heat factor (SHF) for each FCU. This SHF depends on entering wet-bulb temperature and the outdoor air temperature. The SHF and discharge temperature were then used to estimate the latent cooling provided by each FCU using the relationship:

$$\text{Total Cooling} = \frac{\text{Sensible Cooling}}{SHF} \tag{3}$$

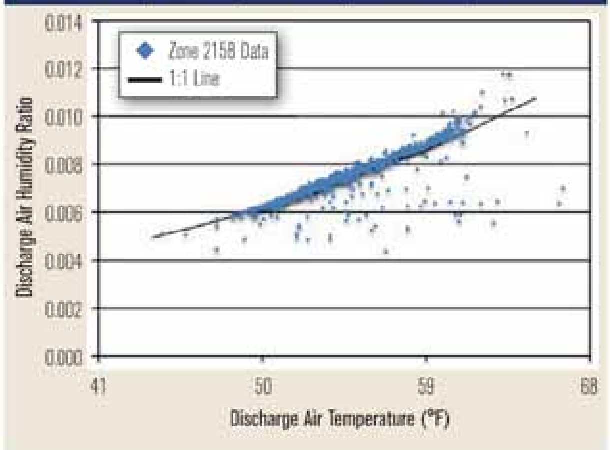
Uncertainty

A detailed uncertainty analysis was performed, taking into account the accuracy of the instruments, the effects of aggregating measurements for individual heat pumps, and the uncertainties associated with estimating humidity levels and airflow rates. Uncertainty analyses necessarily involve assumptions about the nature of the uncertainty! Two key assumptions are:

1. Random errors are normally distributed. This has an important implication for this work; we are attempting to estimate the total cooling and total heating provided by each system, by adding the cooling and heating provided by a number of individual heat pumps or fan coil units. To the extent these uncertainties are random, they tend to cancel each other out. So, if the uncertainty for the amount of heating provided by an individual fan coil unit is ±10% and we are trying to find the total amount of heating provided by 10 fan coil units, the uncertainty of the total is not ±10% but rather ±3%. In some cases, we may also have systematic error that has to be accounted for separately.

2. Errors of individual measurements are independent from each other. So, for example, when computing the heat transfer rate of a heat pump, we assume that the errors in airflow rate measurement are independent of the errors in measuring the temperature difference.

FIGURE 4 Discharge air humidity ratio and temperature relationship for Zone 215B.



With these two assumptions we can combine estimates of uncertainties of individual measurements to estimate the uncertainties of aggregated measures such as total cooling and heating provided. However, estimates of the uncertainties of individual measurements can also be problematic—manufacturers typically provide uncertainties for their sensors, but of course, the sensors may not meet the rated accuracy, and poor installation or usage can further compromise the accuracy. On the other hand, it is easy to grossly overestimate the uncertainty by choosing very-worst-case values for each individual measurement. The often-unstated standard for uncertainty that we are using is the 95% confidence level. However, in many cases that has to be applied with engineering judgment rather than strict quantitative analysis. With this in mind, the uncertainties associated with individual measurements are as follows.

- The temperature sensors used in the building have a manufacturer-rated accuracy of ±0.2°C (±0.5°F), which we used.

- Airflows for each heat pump and VRF FCU are based on the test and balance contractor's measurements. The contractor used a calibrated flow hood with manufacturer-rated accuracy of ±3% ±7 cfm. There has been relatively little peer-reviewed literature checking

FIGURE 5 Estimated monthly heating provided.

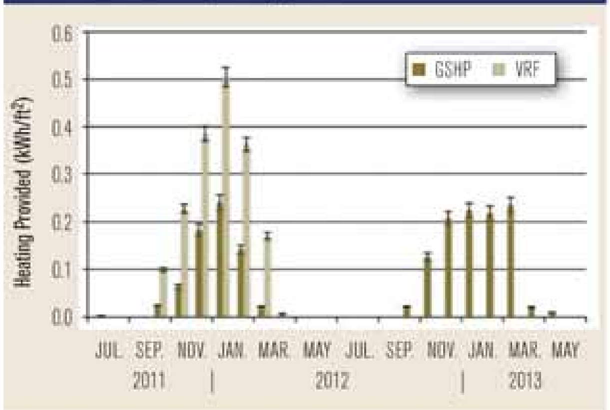
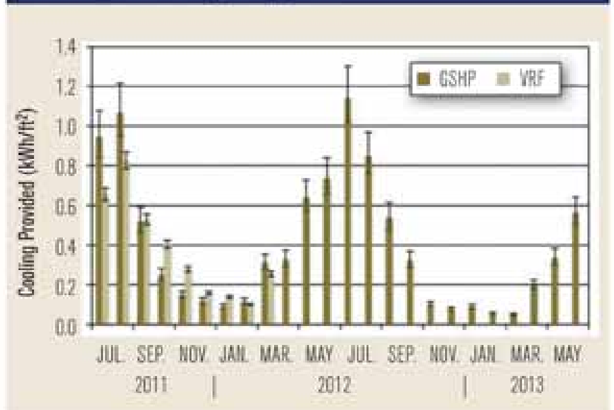


FIGURE 6 Estimated monthly cooling provided.



the accuracy of these measurements in the field. Choat[3] describes a case where the flow hoods gave results that were 14% lower compared to a measurement made by traversing the duct with a pitot tube. We chose to rate the uncertainty of the measurement for each heat pump or terminal unit as ±11.5%. However, it is important to note that this does not lead to an uncertainty of ±11.5% for total cooling or total heating provided. Rather, because the total cooling or total heating depends on the total flow, and as described earlier, random errors tend to cancel each other out when aggregated, the resulting uncertainty in the total flow is lower, but depends on the number of units operating at any one time and their relative capacities. The fewer the number of units on, the higher the uncertainty. We chose a value of uncertainty corresponding to three units of ±7%.

- The estimated humidity level entering all heat pumps is approximated as being the zone humidity level. The estimated uncertainty has two components: the uncertainty of the sensor (±3% RH) and the uncertainty due to using the zone humidity level: (+3%/–0%). The latter value is based on the effect (for some units) of mixing zone return air with DOAS exiting air.
- Humidities leaving the heat pumps are based on our finding that, for the living lab heat pump, the uncertainty of the measured relative humidity is (to a 95% confidence level) ±5.5%. This value is taken as the uncertainty for the humidity levels leaving each heat pump.
- Humidity levels leaving the VRF system FCUs are not measured by the building energy management system. Therefore, we have taken the manufacturer's data to create a map of SHF that depends on entering wet-bulb temperature and the outdoor air temperature. We made spot measurements and found the actual unit SHF to be within ±0.07 of the catalog data, so we have taken the uncertainty in SHF to be ±0.08. With this uncertainty in SHF, we can estimate the uncertainty in total cooling provided at each measurement and for seasonal values.

The resulting uncertainties for the individual heat pumps vary but are around +23/–18% for cooling and ±12% for heating (when there is no dehumidification). When aggregated together, the uncertainty in the total cooling provided is +14/–11% and that for the total heating provided is ±7%. For the VRF system, the uncertainty in cooling provided by a single FCU is +16/–15% and for heating it is ±12%. Typically, there are more FCUs running than there are heat pumps, so when aggregated together the uncertainty in the total cooling provided by the VRF system is ±5% and that for the total heating provided is ±4%. Compared to the uncertainties in estimating the cooling and heating provided, the uncertainties in measuring the electrical energy consumed are negligible, and therefore the uncertainties in the calculated COP and EER are approximately the same as the uncertainties in the total heating and total cooling provided.

Heating and Cooling Provided

The estimated heating and cooling provided by each system are shown in *Figures 5* and *6*, respectively. For the time period from July 1, 2011 until March 31, 2012, which is the time period during which the conditioning provided by the VRF system could be estimated, the GSHP system only provided 38% of the heating that the VRF system provided. During the same time span, the GSHP system provided 6% more cooling than the VRF system provided.

FIGURE 7 Estimated monthly system heating COP.

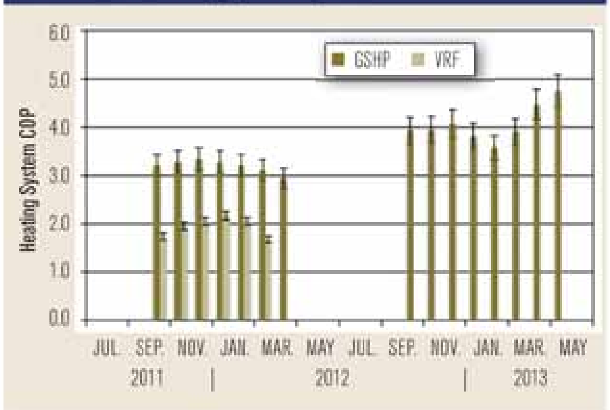
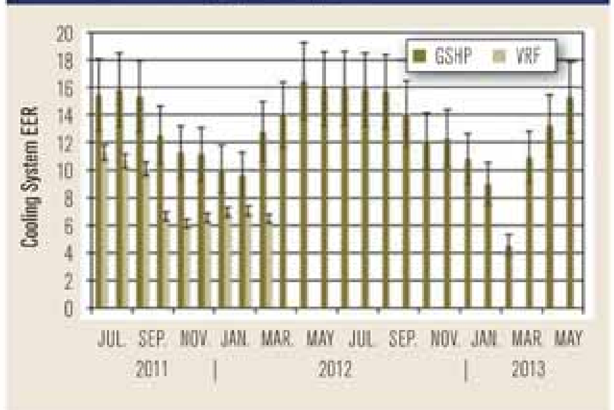


FIGURE 8 Estimated monthly system cooling EER.



Several factors contribute to the large difference in loads between the two systems. First, the DOAS provided nearly twice as much cooling to the first floor (58 MWh/year average during the study period) as to the second floor (33 MWh/year average). This reduces the cooling load, but increases the heating load for the VRF system. As noted in our first article,[2] at times zones on the first floor are overcooled by the outdoor air, causing the FCU for those zones to operate in heating mode to effectively provide reheat. The first floor has lower regular occupancy than the second floor, and the meeting rooms are used infrequently, so it is unclear why the DOAS airflow to the first floor is higher. Also, the temperature control scheme of the VRF system causes the FCUs in adjacent zones in the open office environment to, at times, operate in conflicting modes simultaneously. The loads from this conflicting operation are a larger part of the total heating loads than the total cooling loads because the heating loads due to envelope losses that are not counterbalanced by solar and internal heat gains are relatively small for this building and climate. The conflicting operations can occur in both summer and winter, but the heating loads in summer are small compared to the loads in winter, so they do not show in the scale of *Figure 5*.

To quantify system efficiency, it is necessary to know how much energy was used for each mode of operation (heating or cooling) but only total system power measurements are available. When all units in a system are running in the same mode, the energy used can be allocated accordingly. When individual units were running in different modes simultaneously, system energy use was allocated to heating and cooling based on the total capacity of the units that were running in each mode at the particular time. Allocating the energy use in this way, total system heating COPs and cooling EERs can be estimated, as shown in *Figures 7* and *8*. The error bars reflect the +14/–11% uncertainty in the estimates of cooling provided and the ±7% uncertainty in the estimates of heating provided for the GSHP system and the ±5/4% uncertainty for the VRF system. These system COPs include all of the energy used by each system including fan power for units that are running in ventilation mode, standby power for unit control boards when the building is unoccupied, and pumping power (for the GSHP system).

During the winter of 2011 through 2012, the estimated GSHP system heating COP was 3.3±0.2 and the estimated VRF system heating COP was 2.0±0.1. The following winter the estimated GSHP system heating COPs increased by 18% to 3.9±0.3, in part because the differential pressure setpoint on the ground loop had been decreased from 20 psi to 8 psi, which reduced pumping power. Another contributing factor to the increased COP during the winter of 2012 through 2013 is colder weather, which increased the runtime of the heat pumps and thus proportionately decreased the "overhead" system power use associated with ventilation blowers and pumps. During a May 2014 site visit, a power meter was installed on the pumps for a short time, and power was recorded at differential pressure setpoints of 15 psi and 8 psi. *Figure 9* shows the effect of the differential pressure setpoint on the pumping power. VRF system heating COPs could not be estimated during the winter of 2012 through 2013 because of the equipment modifications in the VRF system.

FIGURE 9 Pumping power at different ground loop differential pressure setpoints.

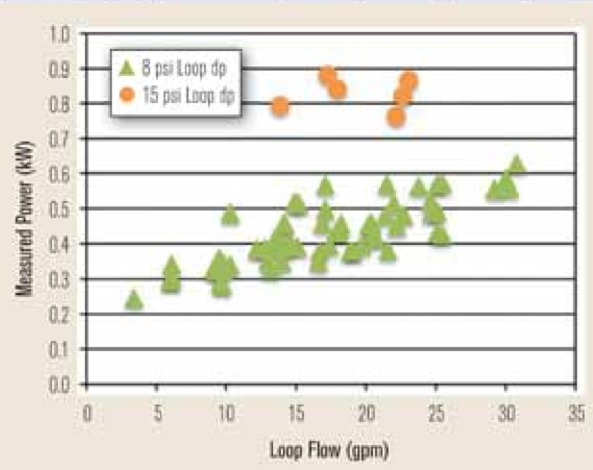
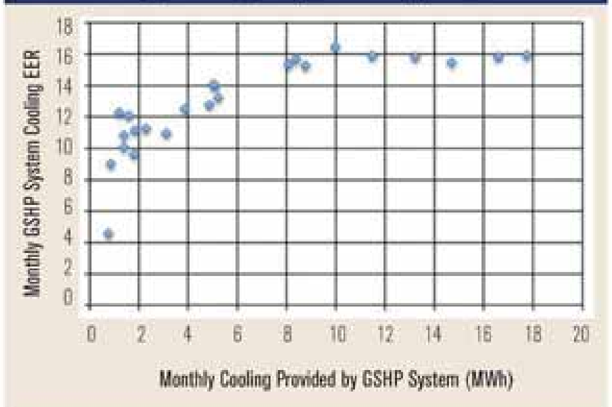


FIGURE 10 GSHP system monthly cooling EER vs. cooling provided.



For July to September 2011, the estimated GSHP system cooling EER was 15.6+2.2/–1.7, while the estimated VRF system cooling EER for the same period was 10.7±0.5. The following summer the estimated GSHP system cooling EER was 15.8. These EERs are lower than what might be expected purely from unit ratings published in manufacturer's catalog data since they account for all of the energy consumption by the heat pumps, fans, and pumps (for the GSHP system) and various operating conditions during the three-month time period. A contributing factor to the relatively low system EERs is the power consumption of the blowers. The fans on all of the heat pumps and VRF FCUs run continuously when the building is occupied even if there is not any heating or cooling demand, in which case the fans run in ventilation mode with reduced airflow. A detailed analysis of the power use by the GSHP system shows that this ventilation-only fan operation accounts for 10% of the total GSHP system energy use. The power use when all units are running in ventilation mode is higher for the VRF system than for the GSHP system,[2] so the reduction in the system energy efficiency due to ventilation-only fan operation is even larger for the VRF system.

Surprisingly, *Figure 8* shows that GSHP system cooling EER is lower in winter when temperatures are more favorable for cooling. This is because only a few units are running in cooling mode, providing only a small amount of cooling, while there is still a significant amount of system energy use associated with running the blowers in ventilation mode for all of the remaining units. Also, with only a small number of units running, the water loop flow rates are low, and the circulation pump and variable speed drive are less efficient at the lower flow rates. *Figure 10* shows the effect of small cooling loads on the system cooling EER.

Conclusions

The living lab at the ASHRAE headquarters building provides an excellent opportunity to learn about the performance of high efficiency HVAC equipment in an operational office building environment.

Based on measured heating and cooling provided, for the first nine months of the study, the average system heating COP of the GSHP system was 3.3±0.2 and the average system cooling EER was 14.2+2.0/-1.6. For the same nine months, the average system heating COP of the VRF system was 2.0±0.1 and the average system cooling EER was 8.5±0.4. For the entire two-year study period, the GSHP system heating COP was 3.6±0.3 and the system cooling EER was 14.5+2.0/–1.6. The heating and cooling efficiencies of both systems are lower than that listed in the manufacturer's catalog data, particularly for the VRF system.

The GSHP system performance improved when the ground loop differential setpoint was decreased from 20 psi to 8 psi. System performance for both systems could be improved if the power use by fans that are running in ventilation mode could be reduced. Since the DOAS system has VAV boxes, if the DOAS blowers are adequate to supply fresh air without the need for additional blowers to boost the air pressure, it might be possible to eliminate ventilation mode blower operation.

Improvements could also be made in the zone temperature control strategies for the VRF system. The current control strategy uses an occupant-adjustable single setpoint in an open office environment that prevents a single unit from switching back and forth between heating and cooling, but can allow the terminal units for adjacent zones to run in opposite modes simultaneously.[2]

There is also the potential to reduce overall building energy consumption by optimizing the DOAS operation. Presently, the DOAS occasionally overcools some zones, causing the zone equipment to act as reheat for the DOAS.[2] The DOAS supply air temperature setpoint is reset if all zone temperatures are below cooling setpoint and outdoor air enthalpy is below a threshold level, or if 80% of zone temperatures are below heating setpoint. At some other ambient air conditions it might be possible to transfer a portion of the cooling and dehumidification provided by the DOAS to the VRF or GSHP systems if they can operate at higher efficiencies than the DOAS.

As always, more knowledge leads to more questions. An abundance of data is available for the ASHRAE headquarters building, but a few more critical pieces of information (such as FCU airflow rates and entering air temperatures) would enable a more complete and accurate analysis. And with all of the data that are available, many more aspects of system operation and design could be investigated, possibly leading to improved performance of the existing system and improved design of future systems.

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