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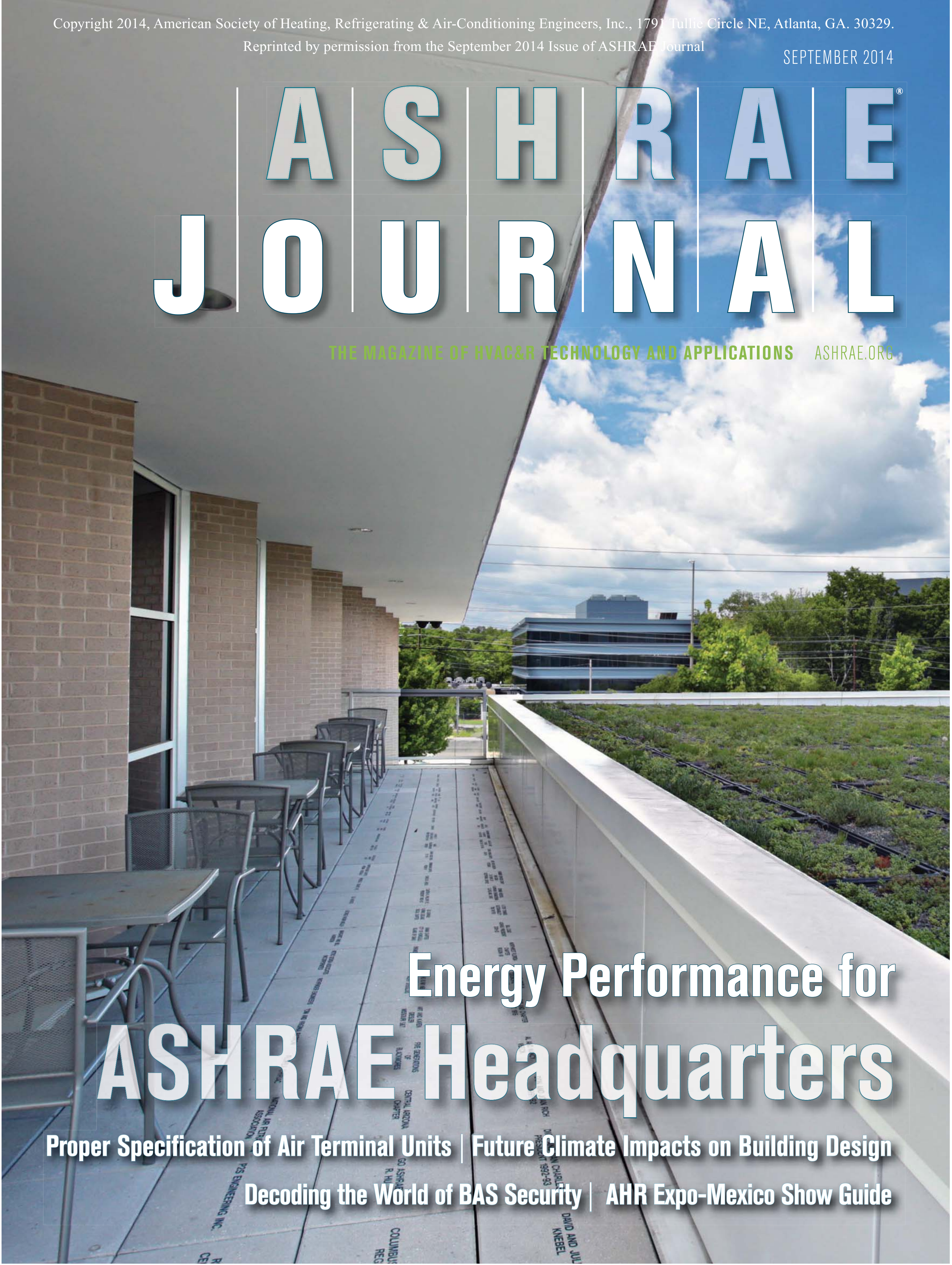

Energy Performance for ASHRAE Headquarters

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

Decoding the World of BAS Security | AHR Expo-Mexico Show Guide

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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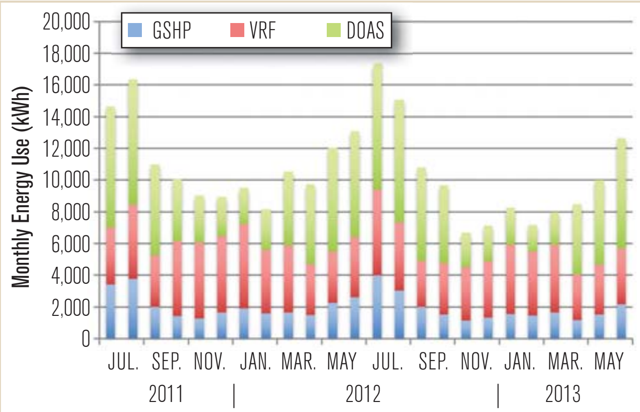

FIGURE 1 Total monthly energy use by each HVAC system.

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.

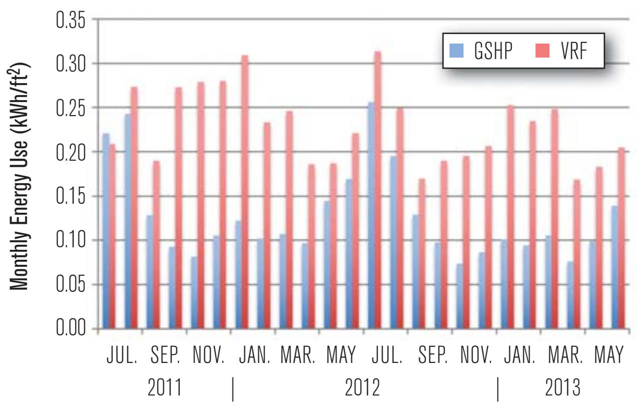

FIGURE 2 Normalized monthly energy use per square foot.

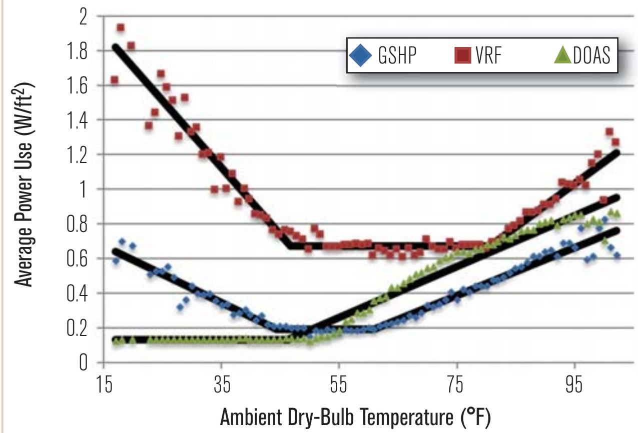

FIGURE 3 Average power use vs. ambient temperature with change-point models.

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

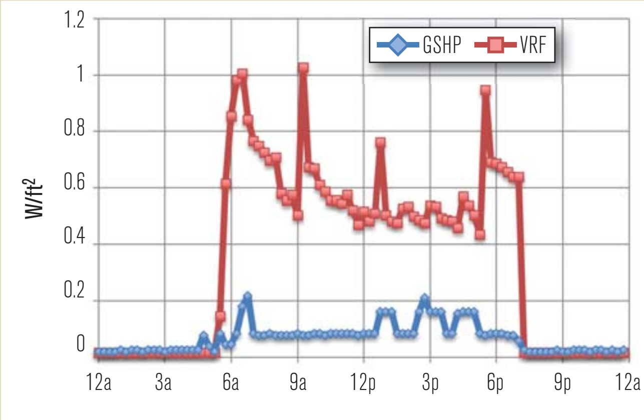

FIGURE 4 GSHP and VRF power use on April 3, 2013.

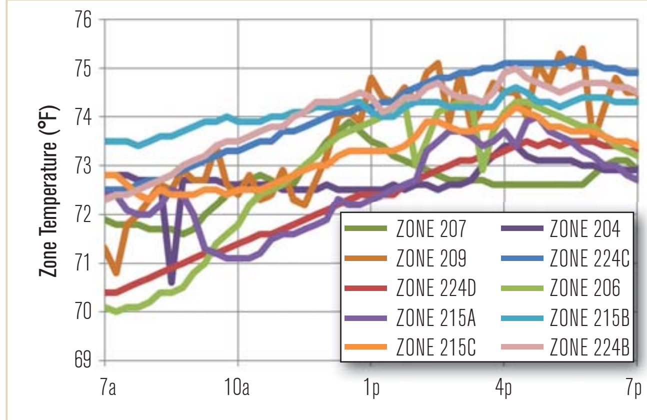

FIGURE 5 GSHP zone temperatures on April 3, 2013.

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

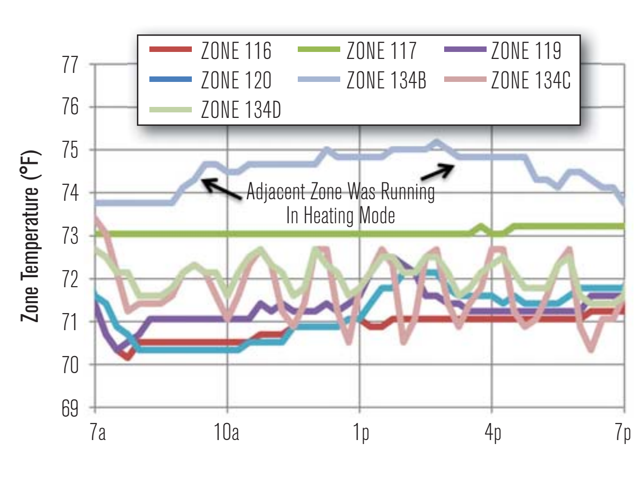

FIGURE 6 VRF zone temperatures for units running in cooling mode on April 3, 2013.

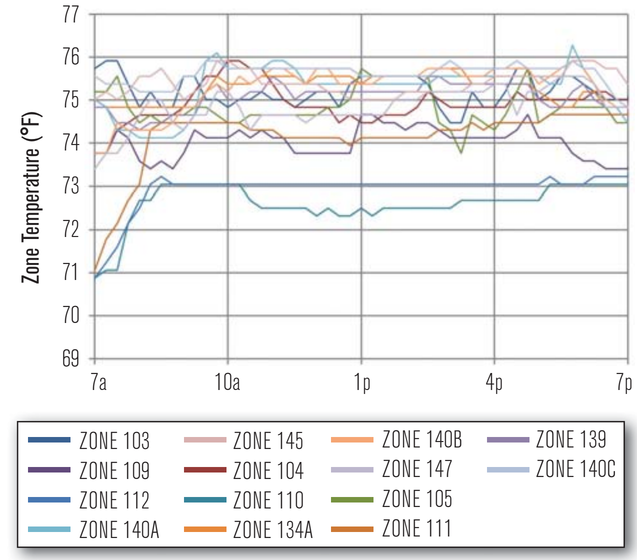

FIGURE 7 VRF zone temperatures for units running in heating mode on April 3, 2013.

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.

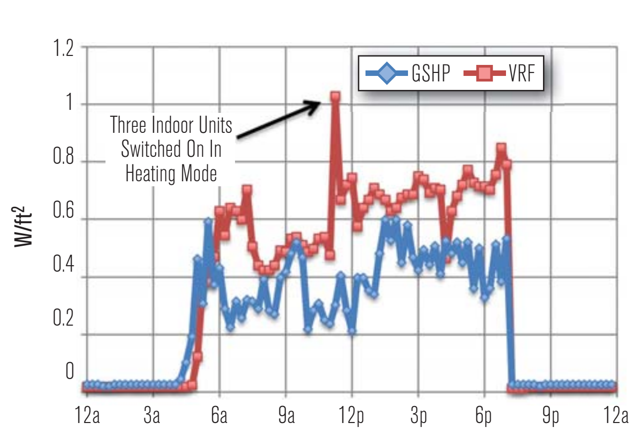

FIGURE 8 GSHP and VRF system power use on June 14, 2013.

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

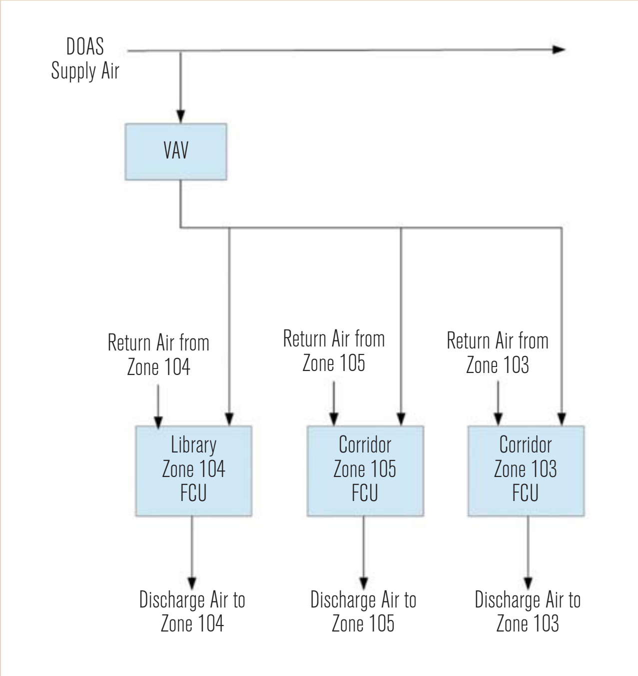

FIGURE 9 Diagram of DOAS supply to Zone 104.

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.

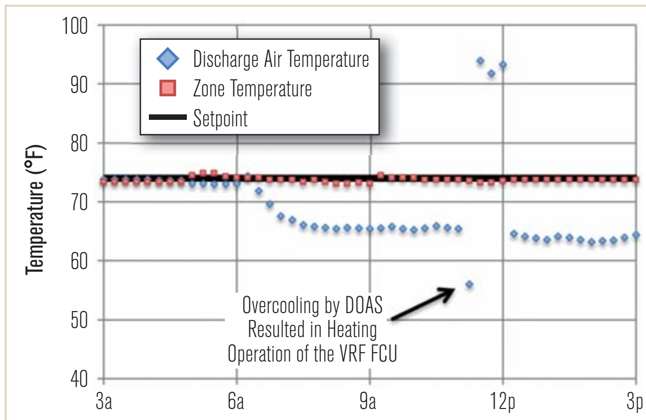

FIGURE 10 Library zone temperatures on June 14, 2013.

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 runtimes 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.

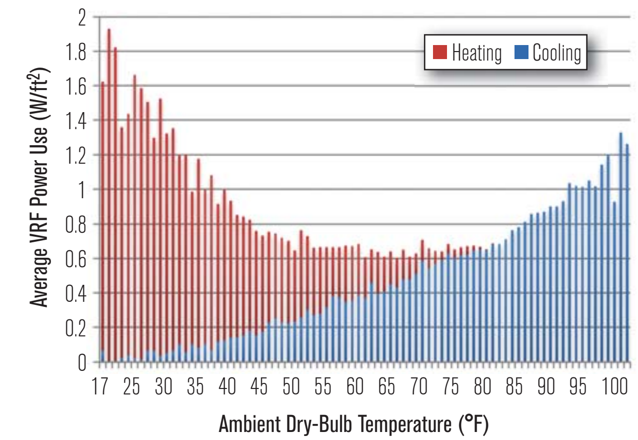

FIGURE 11 Contributions of heating and cooling to VRF system power use.

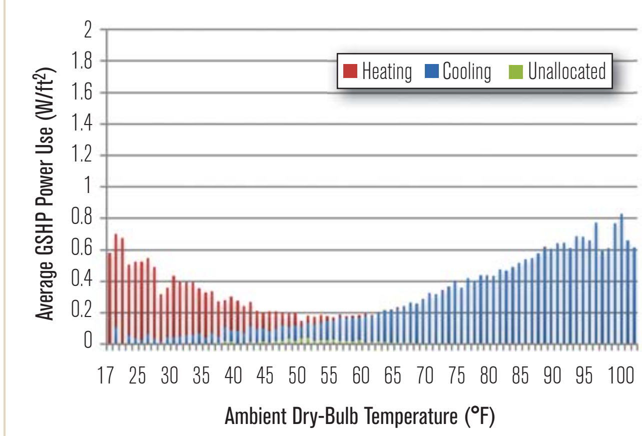

FIGURE 12 Contributions of heating and cooling to GSHP system power use.

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.

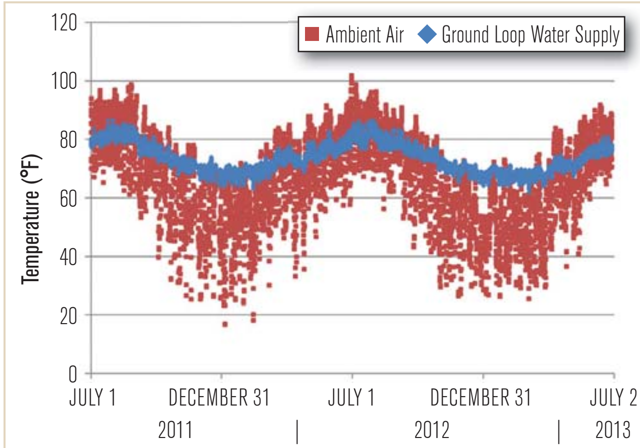

FIGURE 13 Ground loop supply temperature vs. ambient temperature.

Table 2 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.

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.

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

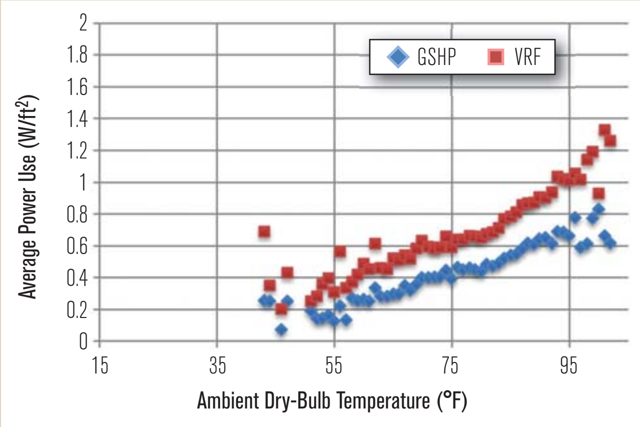

FIGURE 14 Average power use vs. ambient temperature for cooling only.

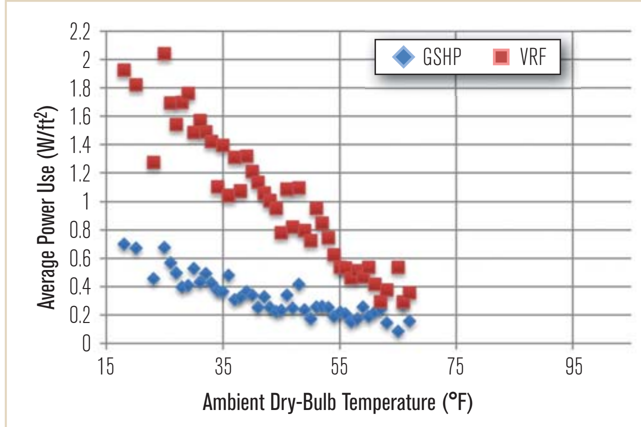

FIGURE 15 Average power use vs. ambient temperature for heating only.

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.

References

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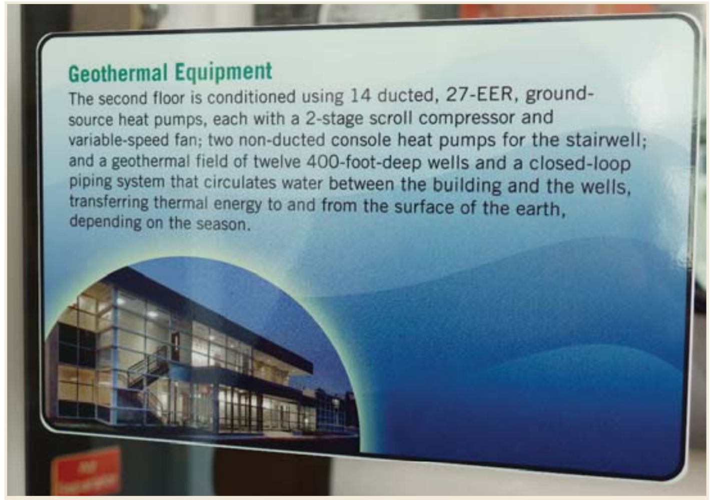
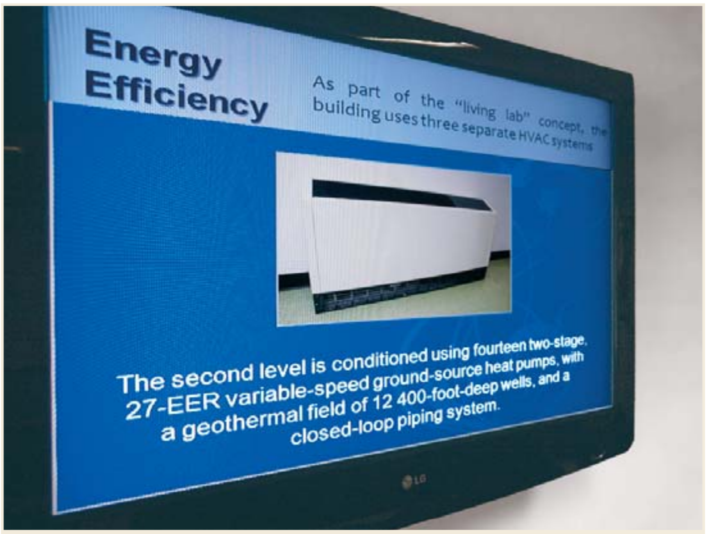
One of two ClimateMaster Tranquility® Console (TRC) Series units installed in ASHRAE headquarters.



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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