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Performance of HVAC Systems at ASHRAE HQ: Part 2

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When the ASHRAE headquarters building in Atlanta, Georgia 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.[2,3] 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 paper[3], 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 2nd floor, as well as a small stairwell on the first floor. The VRF system for which power measurements are available serves all of the 1st 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 1st floor is meeting rooms, which are used infrequently. During 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 4 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[3], but the critical question is, how much conditioning is provided by each of the two systems?

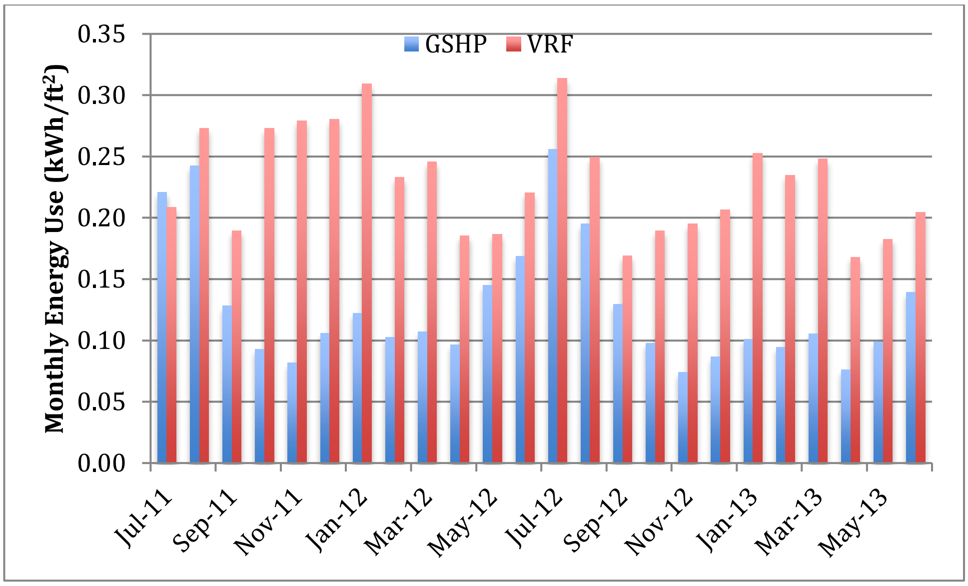

Figure 1 Monthly energy use by the GSHP and VRF systems

In this paper, 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 – 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 2-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 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 schematically 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 air from the plenum. For the other 8 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.

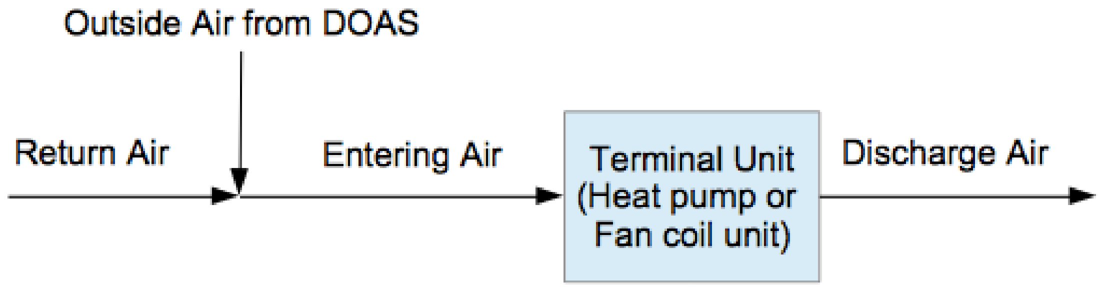

Figure 2 Air flow configuration of terminal units

Table 1 Available measurements

	Heat pumps		VRF system
	Zone 215B	Other heat pumps	
Discharge Air Flow	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

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 air flow 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:

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

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:

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

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 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 outside 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. The use of this correlation for the other zones assumes that the ratio of outdoor air to return air is the same for each of the other zones as it is for zone 215B.

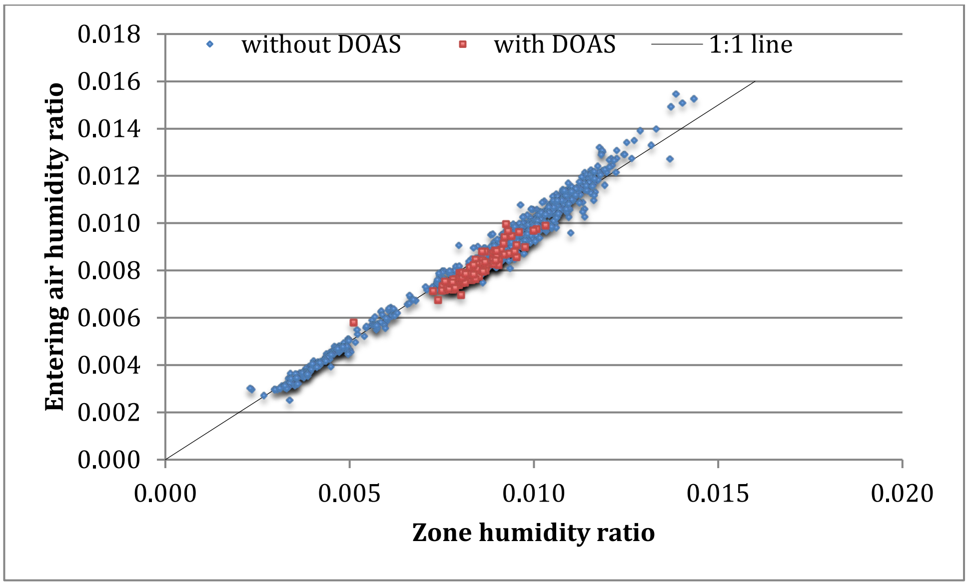

Figure 3 Entering air and zone humidity ratio relationship for zone 215B

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.

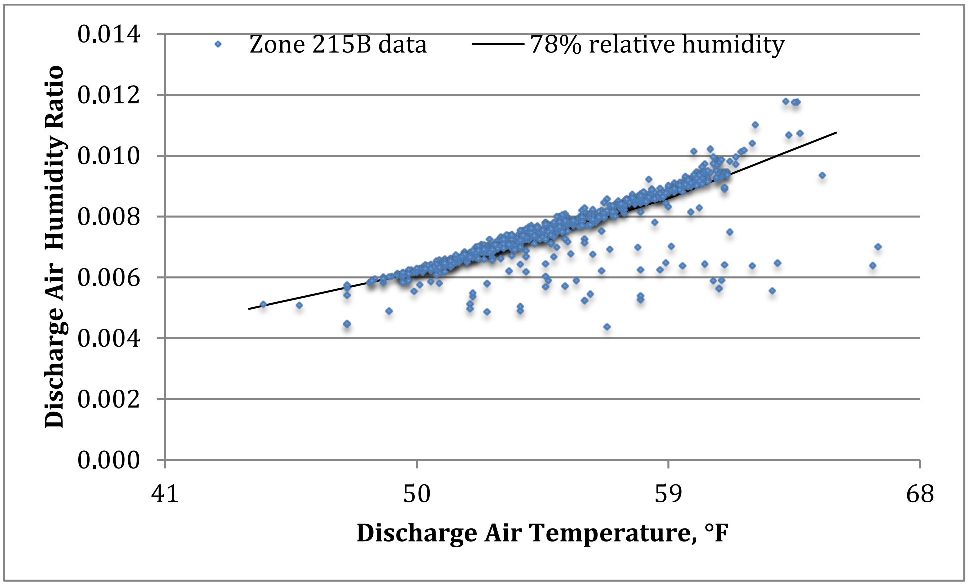

Figure 4 Discharge air humidity ratio and temperature relationship for zone 215B

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 2-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 cool-down operation the DOAS is shut off and, again, this approximation should be good. However, when the building is occupied, pre-conditioned 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.

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 air flow 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.

With these two assumptions we can combine estimates of uncertainties of individual measurements to estimate the uncertainties of aggregate 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 the accuracy of these measurements in the field. Choat[1] describes a case where the flow hoods gave results that were 14% low compared to a measurement made by traversing the duct with a pitot tube. The uncertainty associated with the estimated airflows for each diffuser is ±3.5%. 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 above, 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 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. Therefore, we have taken the manufacturer's data to create a map of sensible heat factor (SHF) which 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, 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.

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 paper[3], 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.

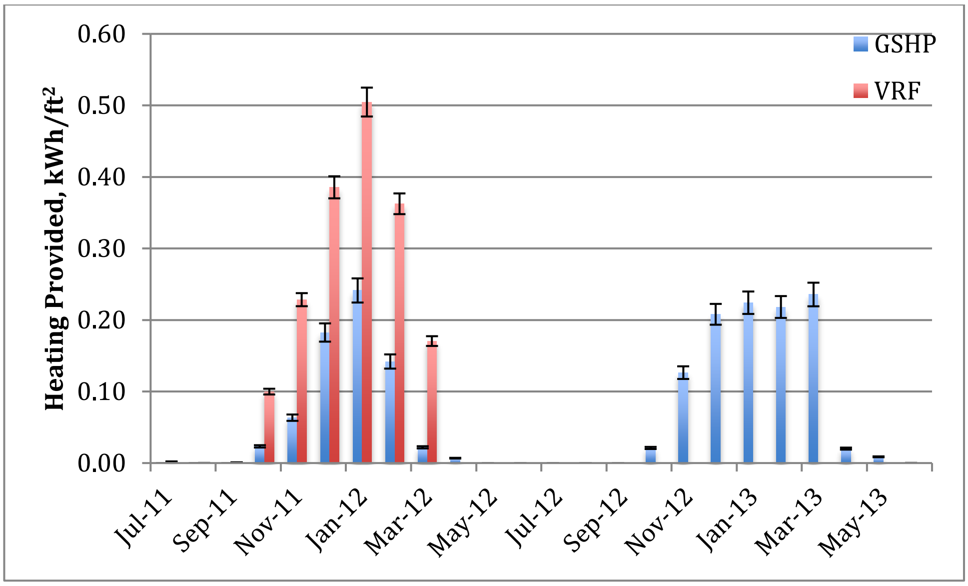

Figure 5 Estimated monthly heating provided

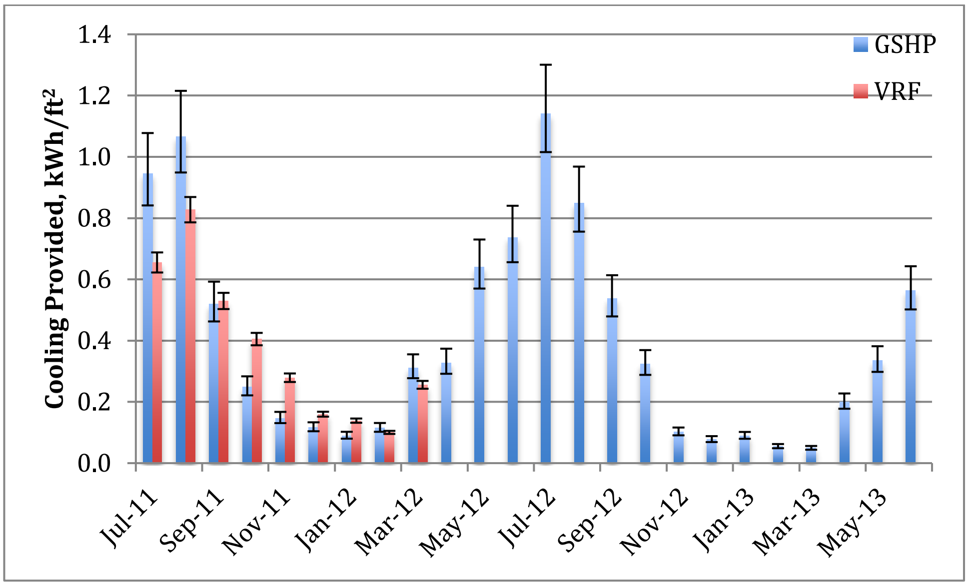

Figure 6 Estimated monthly cooling provided

In order 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).

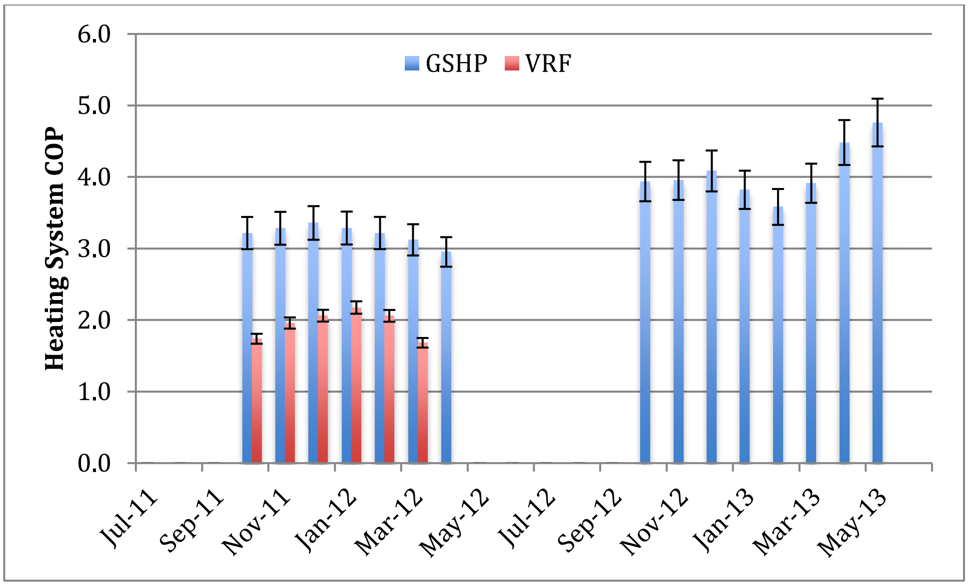

Figure 7 Estimated monthly system heating COP

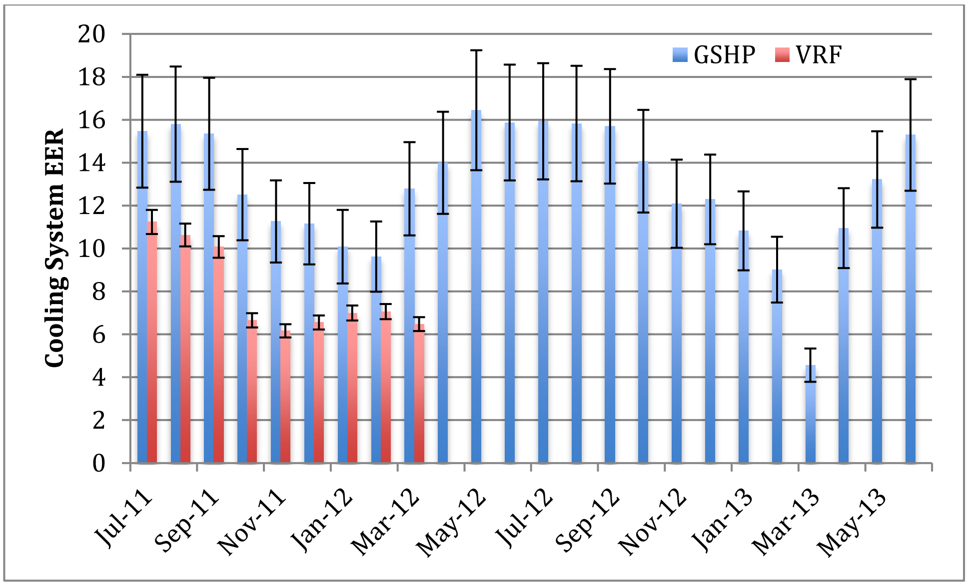

Figure 8 Estimated monthly system cooling EER

During the winter of 2011-2012, the estimated GSHP system heating COP was 3.3±0.2 and the estimated VRF system heating COP was 1.9±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 set point 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 2012-2013 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 set points of 15 psi and 8 psi. Figure 9 shows the effect of the differential pressure set point on the pumping power. VRF system heating COPs could not be estimated during the winter of 2012-2013 because of the equipment modifications in the VRF system.

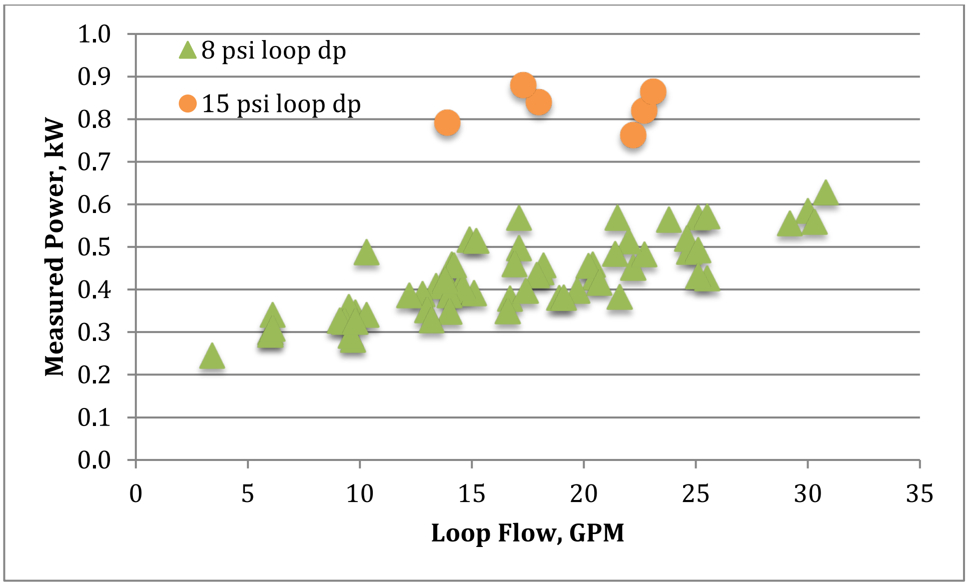

Figure 9 Pumping power at different ground loop differential pressure set points

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 3-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 air flow. 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[3], 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.

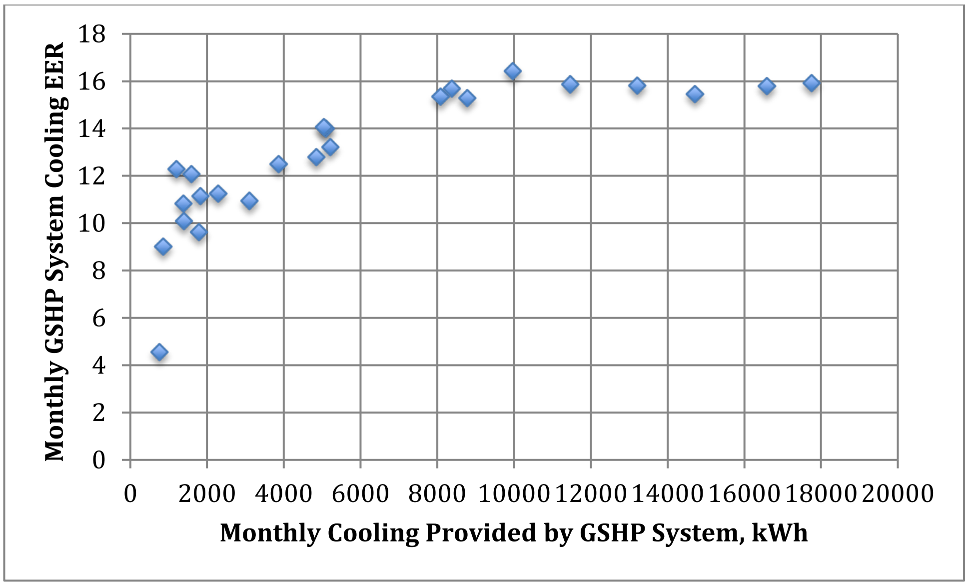

Figure 10 GSHP system monthly cooling EER vs. cooling provided

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 set point 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 set point in an open office environment that prevents a single unit from switching back and forth between heating and cooling but which can allow the terminal units for adjacent zones running in opposite modes simultaneously.[3]

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.[3] The

DOAS supply air temperature set point is reset if all zone temperatures are below cooling set point and outside air enthalpy is below a threshold level, or if 80% of zone temperatures are below heating set point. 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 air flow 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.

References

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