High-Efficiency HVAC Shootout at ASHRAE Headquarters

When ASHRAE International Headquarters in Atlanta was renovated in 2008, one goal was to create a living laboratory in order to learn more about commercial building performance and state-of-the-art HVAC technology.

As a part of this 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 to condition the second floor, and a dedicated outdoor air system (DOAS) to supply fresh air to both floors. Of key interest was directly comparing the performance of the GSHP system with the VRF system.

To evaluate the performance of the different systems, data relating to their operation was collected for two years, from July 1, 2011 through June 30, 2013. An extensive array of sensors that monitor the HVAC equipment and conditions in each zone was also installed. Both historical and current data from these sensors have been trended via the building automation system.

Specifics

The VRF system includes two multi-zone, inverter-driven heat-recovery units. These are connected to a total of 22 fan coil units (FCU) with two-speed fans. Their total cooling capacity is 28 tons.

The GSHP system that serves the second floor includes 14 individual water-to-air heat pumps (two, 0.75 ton units, six, 2-ton units and six, 3-ton units) connected to a ground loop consisting of 12 vertical boreholes at 400 feet deep, for a total of 31.5 tons of cooling capacity. The heat pumps have variable speed fans with three selected speeds.

The DOAS includes six, air-cooled condensing units and two heat recovery wheels. The total cooling capacity of the condensing units is 28.6 tons.

Collected data points include operating mode (off/heat/cool), zone temperature and discharge air temperature for each individual FCU or heat pump. Ground loop supply and return temperatures and flow rate were also collected for the GSHP system. The DOAS flow rates, 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 for all 14 heat pumps as well as the ground loop water circulation pumps was recorded. For the VRF system, only the power for the two heat-recovery units and the 22 FCUs that are connected to them was recorded.

Figure 1 shows the monthly energy used 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.

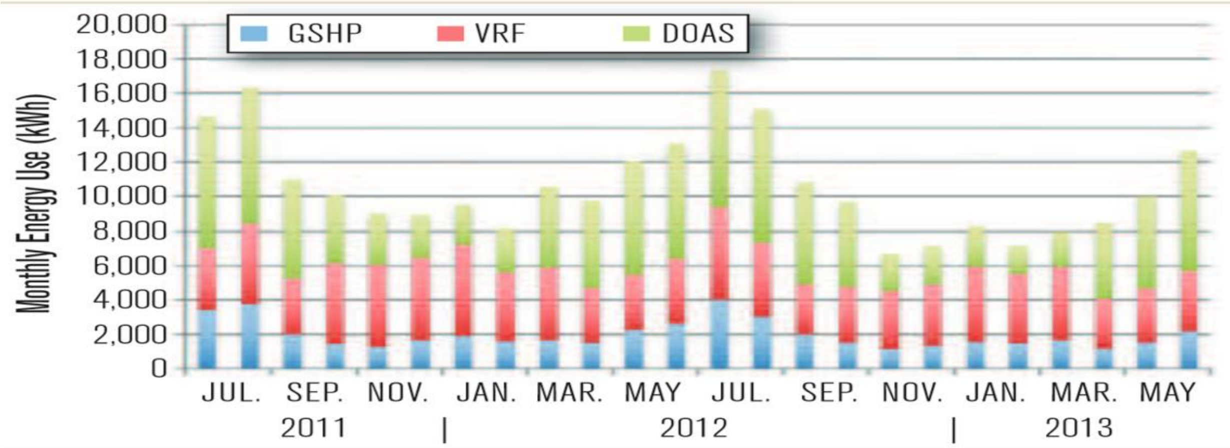


Figure 1

So what caused such significant differences in the energy use, despite the fact that both VRF and GSHPs are considered "high-efficiency" systems? This is the result of several factors:

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

Different Loads

The GSHP system serves 15,558 sq. ft. of office and meeting space primarily on the second floor with a normal occupancy of 60 people. The VRF units serve a total of 17,213 sq. ft. on the first floor, which includes offices, large meeting spaces and storage areas. But the normal occupancy of this area is only 43 people.

The DOAS airflow 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, compared to the second floor average of 1,480 cfm.

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/ft2) of the GSHP system over the two-year period is 44 percent less than that of the VRF system.

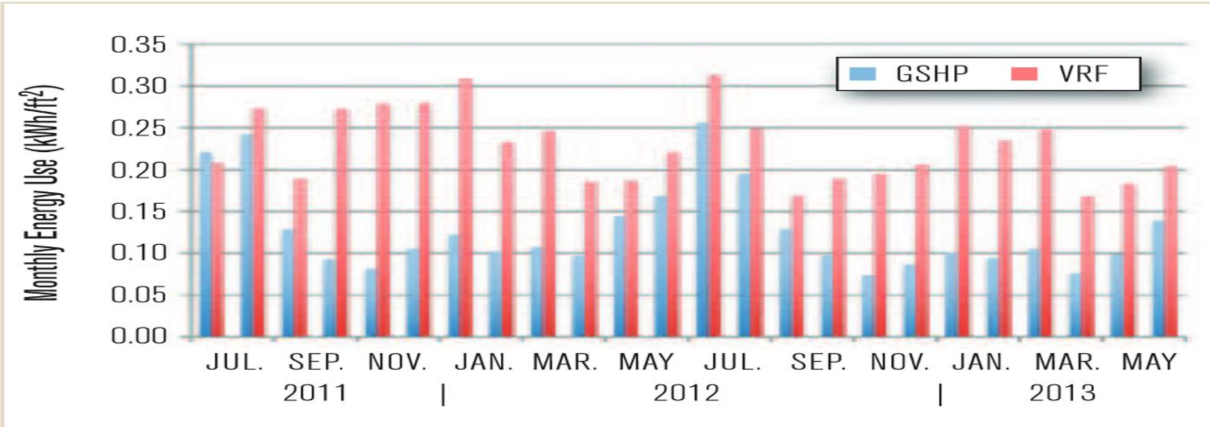


Figure 2

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 and shoulder seasons.

Figure 3 shows instantaneous power usage for all three systems during occupied building hours averaged for each 1°F outdoor air temperature bin and normalized by the floor area served by each system. The VRF system shows unexpectedly high power use even at times with mild temperatures.

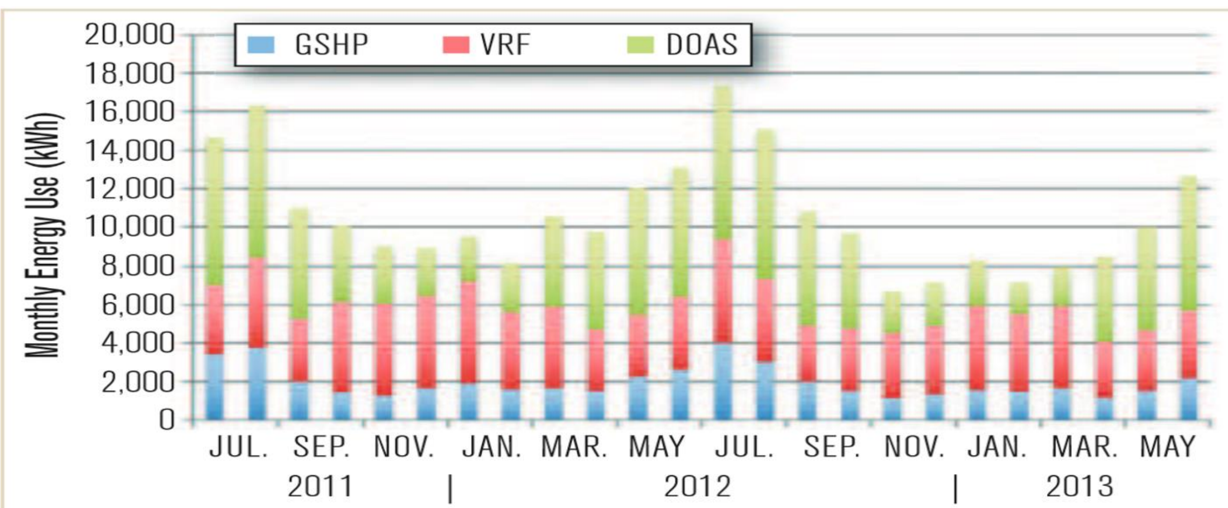


Figure 3

As shown in *Figure 3,* the minimum power use for the GSHP system is 0.19 W/ft2 over a temperature range of 44°F to 61°F. The minimum power use for the VRF system is 0.67 W/ft2 over a much wider range of 47°F to 81°F.

The different power use in mild weather appears to result, at least in part, from the control strategies of the two systems. In mild weather, 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). As a result, few heat pump compressors operated then, with most units 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. This resulted in adjacent FCUs “fighting” each other.

Ultimately, the VRF system’s proprietary controls further hindered the performance of the system by making FCUs run without reason.

Different Sources and Operational Efficiencies

GSHP systems and the air-source VRF systems gather and reject energy in very different ways; ground vs. ambient air, respectively. This makes a huge difference in terms of efficiency. Shown in *Figure 4,* the ground loop supply temperatures are far more favorable than the ambient air temperature - lower when cooling is needed and warmer when heating is demanded.

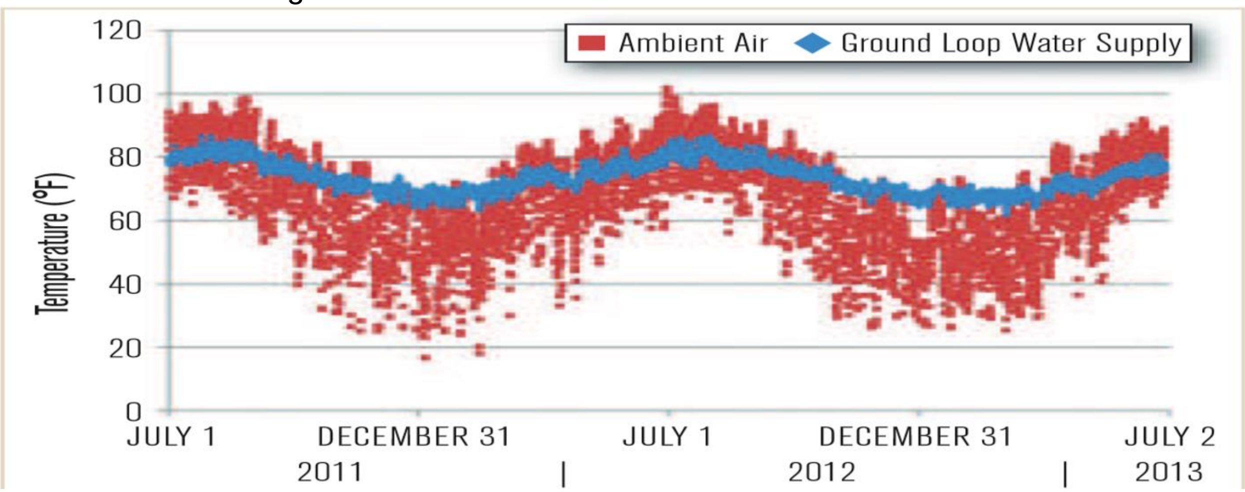


Figure 4

The temperature differential between the ground loop supply and the ambient air is much larger in winter than in the summer, which indicates a major efficiency advantage for the GSHP system in the winter.

Table 1 shows manufacturers’ data for the cooling and heating efficiency of the VRF system and the GSHP equipment at source temperatures. While it’s difficult to directly compare systems that use different sources, 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.

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 1

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. For these heating-only data points, VRF system power use is more than double GSHP system power use. See *Figure 5*

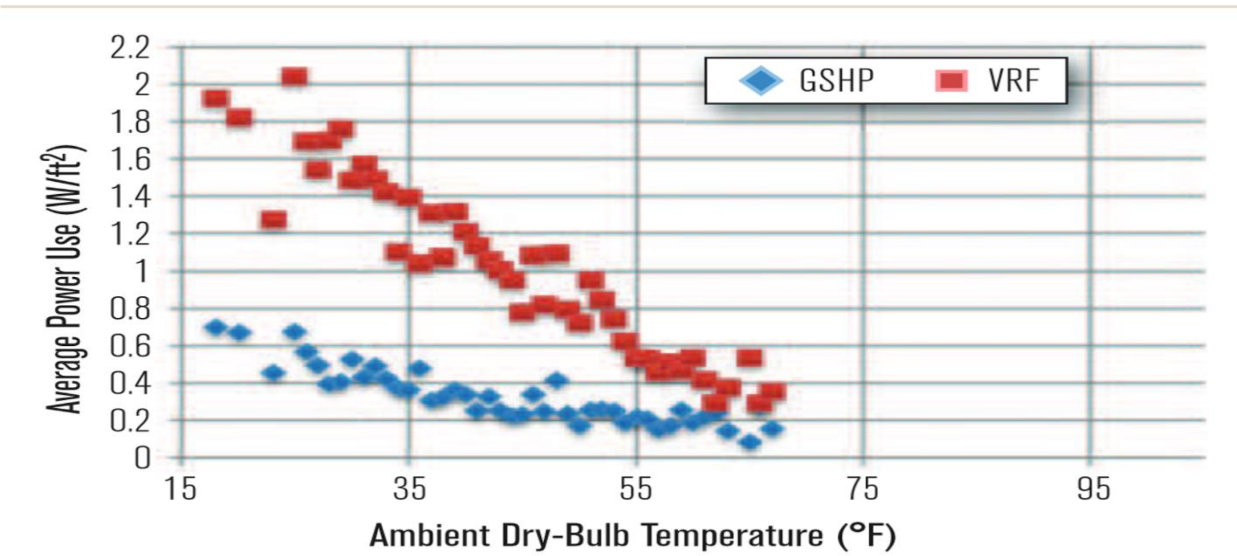


Figure 5

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 presented in this article include all of the operational power used by each system.

True findings

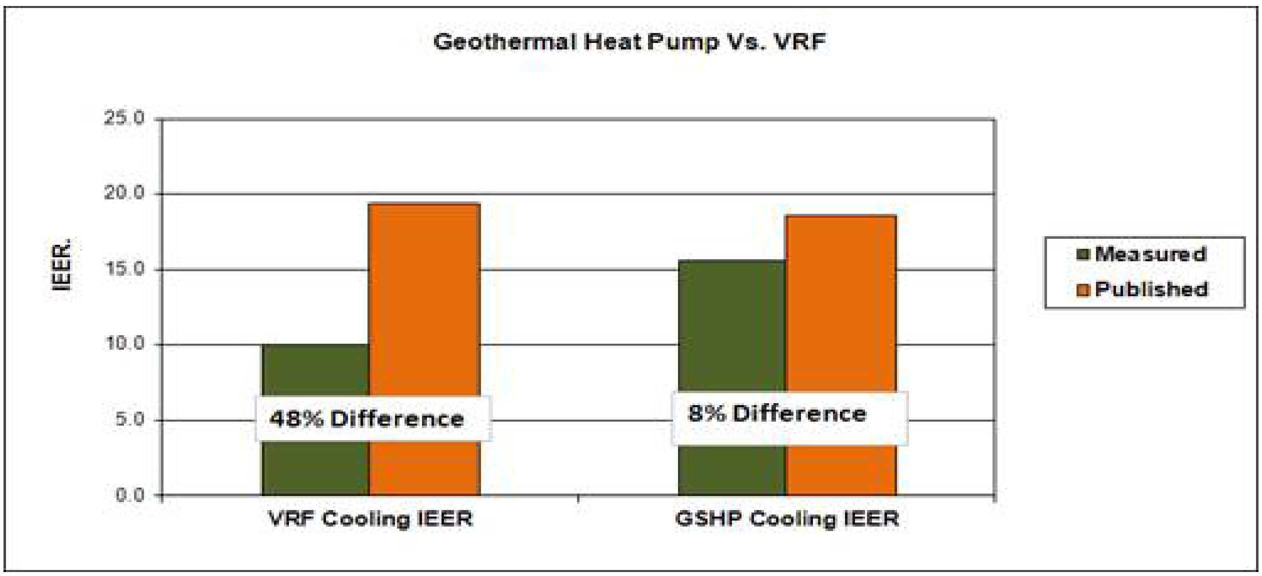
During the two-year period, the GSHP system used about 20 percent less energy than the VRF system during the summer, and roughly 60 percent less energy in the winter and shoulder seasons while maintaining similar zone temperatures.

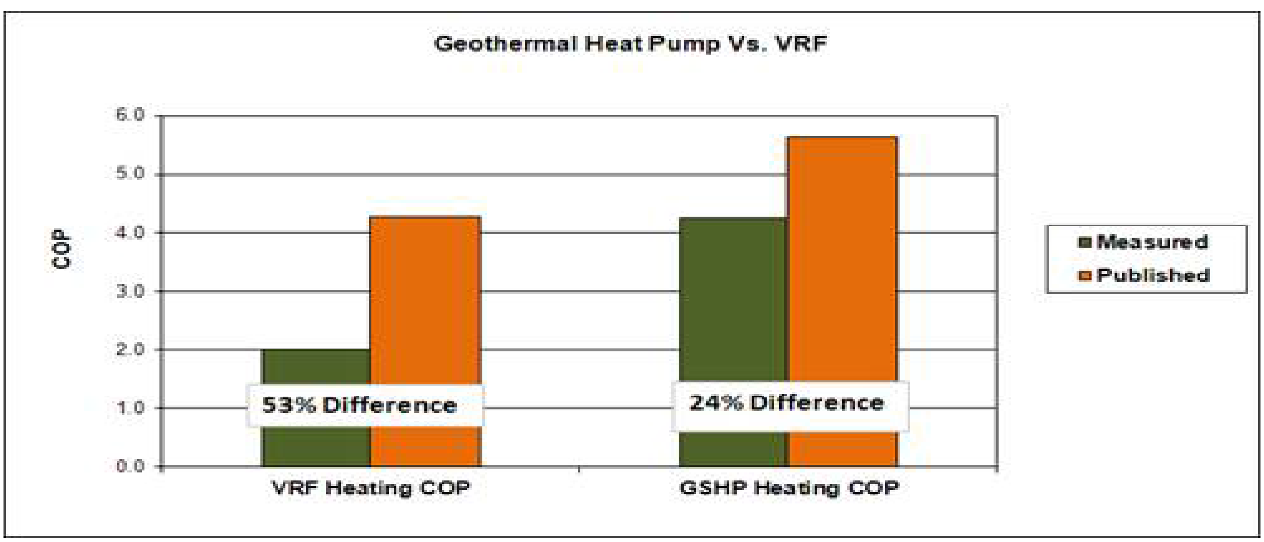
Another, all-inclusive way to look at the results is to compare the space-averaged annual energy use. For the GSHP system, this was 1.5 kWh/ft² per year, while VRF system consumed 2.7 kWh/ft² per year. The VRF system used nearly twice the energy that the GSHP system used.

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. In contrast, the VRF system's heating COP was 2.0±0.1, and the average system cooling EER was 8.5±0.4.

VRF efficiency ratings may be misleading

The heating and cooling efficiencies of both systems, as tested at the ASHRAE building, were lower than that listed in the manufacturers' catalog data. But the published data for the VRF system particularly could be very misleading. The following graphs compare the measured to published data for both systems.





Note that for both heating and cooling, the VRF system's measured COPs and IEERs are roughly 50 percent less than the published data. For the water source heat pump system, the differences are significantly smaller and to be expected when system energy transfer – pump horsepower – is accounted for.

Much of the discrepancy for the VRF system comes from how the systems are tested, vs. how they are actually installed, namely the pumping energy as the compressor becomes the system pump. There's roughly 300 ft. of refrigerant pipe in the ASHRAE building, causing performance de-rate for the VRF efficiency. When VRF units are tested for efficiency rating, the compressor and the evaporator are only separated by 12 feet.

Additional loss of efficiency is contained in multiple de-rates for operation outside the AHRI test points; hotter or colder ambient temperatures. More are due to failure of the units to function at the part-load conditions contained in the AHRI certification formula.

Conclusion

The VRF system energy consumption is up to 80 percent higher than the water source geothermal system.

Simply put, VRF systems can't fairly be compared with GSHP systems. The difference is so high that a basic boiler-cooling tower WSHP (water-source heat pump) application would be more energy efficient than VRF. Because the difference between the two is greatest in heating season, imagine if this comparison had taken place in Chicago instead of Atlanta.

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For a more detailed account of the VRF and GSHP comparison conducted at ASHRAE headquarters, please see the pages below:

Click to read Part 1 of original ASHRAE Journal article
Click to read Part 2 of original ASHRAE Journal article