Energy Modeling of Ground Source Heat Pump vs. Variable Refrigerant Flow Systems in Representative US Climate Zones

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Abstract

The ground source heat pump (GSHP) and variable refrigerant flow (VRF) systems are the most competitive HVAC technologies in the current market. However, there are very few studies reporting the comparison of the annual energy consumptions and Electric Peak demand reductions between GSHP and VRF systems because of the limitation of the whole building energy simulation software. Current version of EnergyPlus can model both GSHP and air-source VRF. Therefore, three representative US climate zones including Chicago, Baltimore and Atlanta are selected for conducting this comparison study. The EnergyPlus simulation results show that the GSHP system not only saves more energy than the air-source VRF system but also significantly reduces the Electric Peak demand regardless the climate conditions. This makes the GSHP system a more desirable energy efficient HVAC technology for the utility companies and their clients.

1. Introduction

Goetzler et al. [1] addressed that residential and commercial buildings consume about 40% of US primary energy including 74% of electricity consumption, 56% of natural gas consumption, and significant oil consumption in the Northeastern. Over the long term, buildings are expected to continue to be a significant component of increasing energy demand and a major source of carbon emissions, driven in large part by the continuing trends of urbanization, population and GDP growth, as well as the longevity of building stocks. The increasing importance of building energy efficiency generally, as well as EERE's programmatic focus on net zero energy homes (NZEH) and net zero energy commercial buildings (NZEBs) brings tremendous challenges and opportunities to the Heating, Ventilation, Air-Conditioning, and Refrigeration (HVAC&R) industry. Many new, or relatively new, HVAC&R technologies [2] are promoted with emphasis on their superior energy efficiency. Among these, the ground source heat pump (GSHP) and variable refrigerant flow (VRF) systems are the most competitive HVAC technologies in the current market.

As shown in Fig.1, the GSHP system rejects the heat to the ground (in the cooling mode) or extracts the heat from the ground (in the heating mode). It takes the advantages of the moderate ground temperatures to increase the efficiency and reduce the operating cost of the HVAC system. It usually comprises of multiple water-to- air heat pump indoor units, which are connected with the ground loop heat exchanger through a common two-pipe water loop. Since each of the water-to-air heat pump units can run in either cooling or heating mode independently, the GSHP system can provide simultaneous cooling and heating for different zones of the building. As of 2004, Lund et al. (2009) [3] reported that over a million GSHP units were installed worldwide to provide 12 GW of thermal capacity, with an annual growth rate of 10%.

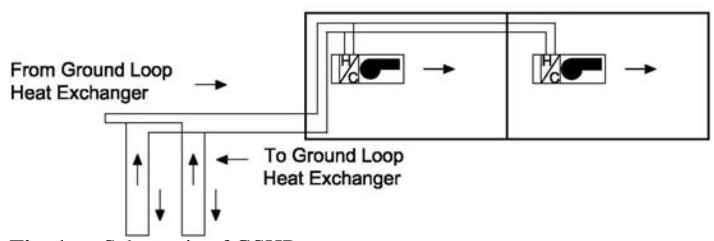

Fig. 1 Schematic of GSHP system

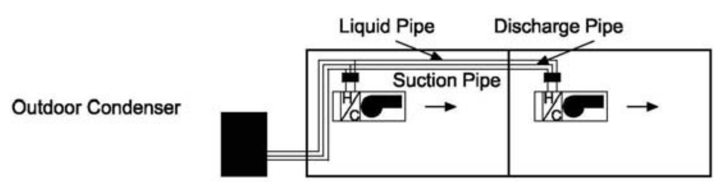

Fig. 2 Schematic of VRF system

The VRF system was first introduced in Japan in 1982 [4] as ductless multi-split air conditioning technology. The key is the refrigerant flow control. Fig.2 shows that the mul-

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tiple indoor units are connected a single outdoor condensing unit. Without the ductwork, the refrigerant is circulated to the indoor units and directly transfers the heat from or to the conditioned spaces. In addition, it can continually control the amount of refrigerant flowing to each of the evaporators, enabling the use of many evaporators of differing capacities and configurations. The system modulates the compressor speed of the outdoor unit to meet the total heating and cooling demands in the building. Therefore, the advantages of the VRF system include individualized comfort control, simultaneous heating and cooling in different zones, and heat recovery from one zone to another.

Because VRF system is still relatively new to the US market and most of HVAC practitioners including building energy modelers in the HVAC industry, there is few published literature comparing the annual energy consumption between GSHP and VRF systems. Liu's study [5] shows the GSHP system saves 9.4% and 24.1% of HVAC energy in Miami and Chicago compared with the ''heat recovery'' type VRF system. Currently, most of energy modeling programs has certain limitations on their simulation capabilities, and can only model either GSHP system or VRF system. As a key part of DOE's building energy-efficiency strategy, the whole building energy simulation program, EnergyPlus has this remarkable energy analysis capability, and then was chosen for this comparison study. EnergyPlus 7.2 [6] has also expanded its modeling capability to allow simultaneous heating and cooling (heat recovery) for the VRF system.

2. Description of simulated building

As shown in Fig. 3, a small office was selected for this comparison study. The office has a rectangular footprint and total conditioned space of 465 m2, which has four thermal zones in the perimeter and one core zone in the interior, as illustrated in Fig. 4. The floor to floor height is 3.66 m with 0.61 m high return plenum. The building is oriented 30 degrees east of north.

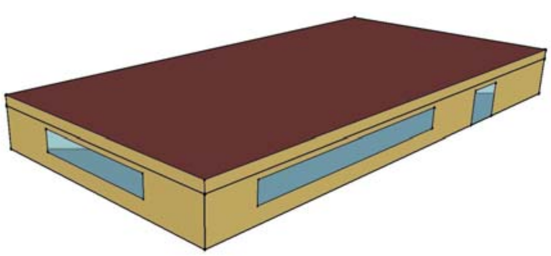

Fig. 3 3D view of the simulated small office building

The same office was assumed to be located in three representative US climate zones as described in the ASHRAE standard 90.1-2010. The three climate zones include Mixed-Humid (Zone 4A), Cool-humid (Zone 5A) and Warm – Humid (Zone 3A). Baltimore, Chicago and Atlanta were selected to represent these climate zones, respectively. Table 1 lists the construction details of the small office

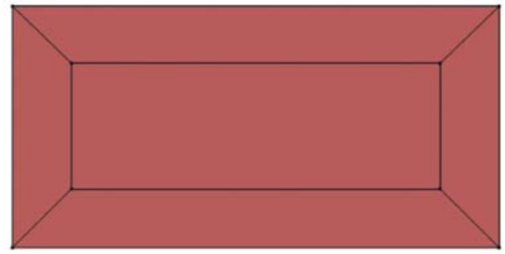

Fig. 4 Floor plan of the simulated small office building

Table 1 Construction of the small office building

Building Envelope	Construction Detail
Exterior wall	Wood shingle over plywood with R-11
Roof	Built-up roof with R-3 mineral board insulation and plywood
Floor	Slab-on-grade with R-30 insulation
Windows	1) Double pane clear, 3mm glass, 13mm air gap 2) Double pane clear, 3mm glass, 13mm argon gap 3) Double pane clear, 6mm glass, 6mm air gap 4) Double pane lowE, 6mm lowE glass outside, 6mm air gap, 6mm clear glass
Door	Single pane grey, 3mm glass

Table 2 Internal loads of the small office building

Internal Load	Unit
Light power density	16.1 w/m2
Equipment load	10.8 w/m2
Occupant density	11 people/100 m2

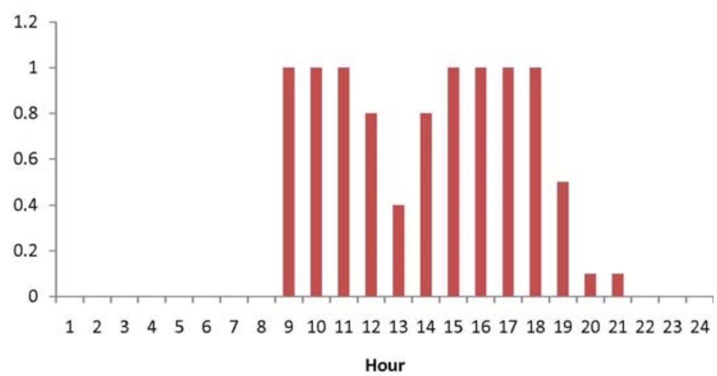

Fig. 5 Building occupancy schedule

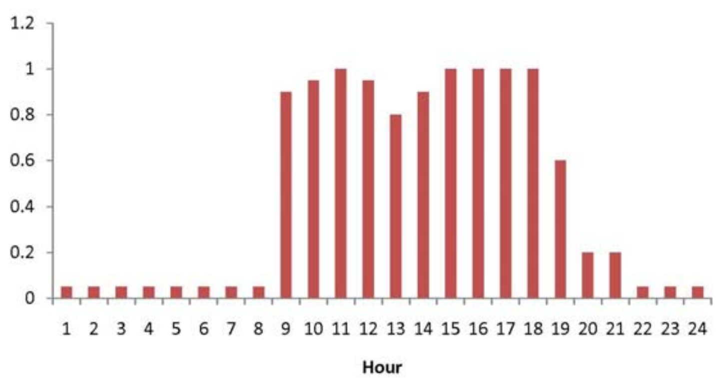

Fig. 6 Building lighting schedule

building. The corresponding internal loads are shown in Table 2 including lighting power density, equipment load and occupant density.

The office operated from 6 am to 7 pm during the course of the year. In the cooling mode, the thermostat setpoints were 24°C during occupied hours and 30°C during unoccupied hours. In the heating mode, 20°C was selected as occupied room temperature and 15°C was used in the unoccupied hours. In order to conduct a fair comparison study, the indoor fan was assumed to run continuously with the constant air flow rate during the occupied hours, which was autosized by EnergyPlus. The fan efficiency and motor efficiency were 0.7 and 0.9 with 329 Pa pressure rise. Fig.5- Fig.7 show the daily building occupancy, lighting and equipment schedules.

3. Model description

3.1 VRF System

Because there is no performance data available, especially at the part load conditions, the performance curves in the EnergyPlus VRF template [7] was adopted in this study. In the EnergyPlus model, the simulated VRF systems have the rated cooling COP 3.29 and heating COP 3.55. The rated capacity is autosized in the model. The independent variables used for the cooling performance curves of the VRF system are indoor wet-bulb temperature (IWBT) entering the indoor terminal units and outdoor dry-bulb temperature (ODBT) entering the outdoor condenser. Similarly, the heating performance of the VRF system is characterized by the indoor dry-bulb temperature (IDBT) and outdoor wet-bulb temperature (OWBT). These performance curves can be divided into two separated low and high operating temperature zones which are based on ODBT in the cooling mode and OWBT in the heating mode. The boundary curves are also needed to define the ranges of these two temperature zones as shown in the following equations [7]. The boundaries of performance curves depend on the IWBT for cooling and IDBT for heating in the following two equations. For cooling, the range of IWBT is between 11°C and 30°C. For heating, IDBT is in the range of 15°C and 27°C. Table 3 presents the low and high temperature ranges at the rated IWBT for cooling and IDBT for heating, respectively.

VRFCoolCapFTBoundary=a+b×IWBT+c×IWBT2 Where

a: coefficient constant1, = 25.73473775

b: coefficient constant2, = -0.03150043

c: coefficient constant3, = -0.01416595

VRFHeatCapFTBoundary=a+b×IDBT+c×IDBT2 Where

a: coefficient constant1, = -7.6000882

b: coefficient constant2, = 3.05090016

c: coefficient constant3, = -0.1162844

Table 3 Low and high temperature ranges

Mode	Temperature	Low range	High range
Cooling	IWBT: 19.4°C	ODBT: -5°C ~19.8°C	ODBT: 19.8°C ~43°C
Heating	IDBT: 21.1°C	OWBT: -20°C ~5°C	OWBT: 5°C ~15°C

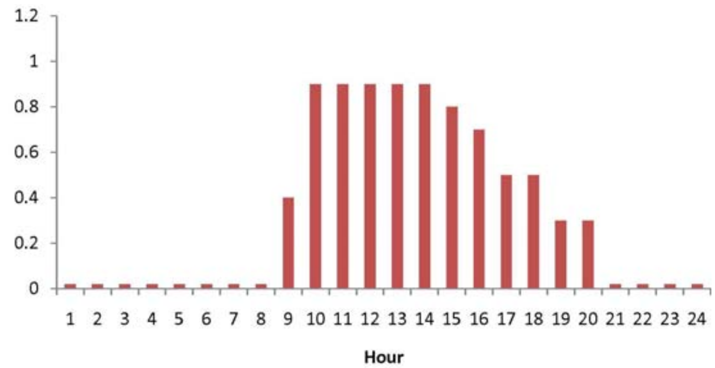

Fig. 7 Building equipment schedule

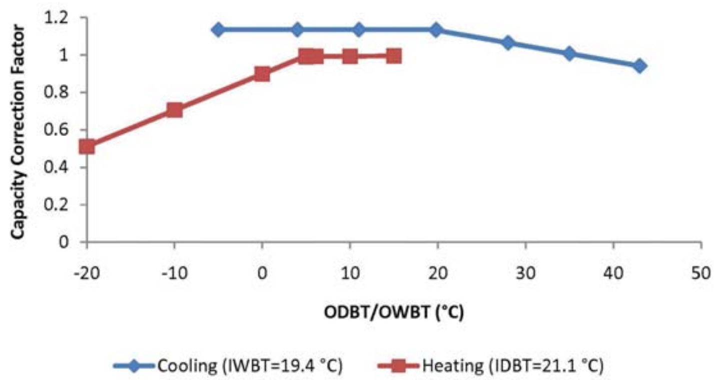

Fig. 8 VRF Cooling and heating capacity correction factor

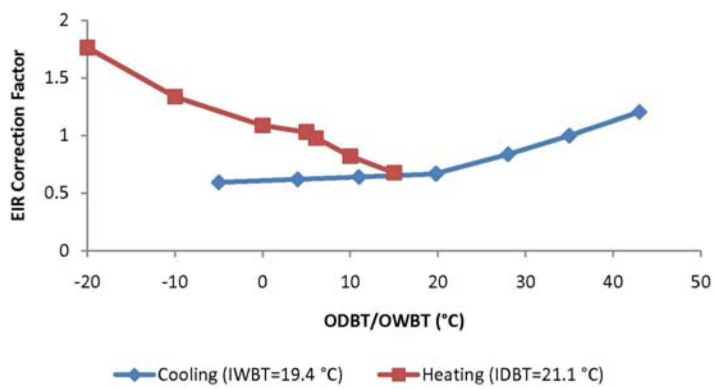

Fig. 9 VRF Cooling and heating EIR correction factor based on the temperature

As defined in the AHRI standard 1230-2010 [8], the cooling rating condition is 35 °C ODBT and 19.4 °C IWBT. All the total cooling capacities were normalized by the rated total cooling capacity. The same methodology was applied to generate the heating performance curves with the heating rating conditions, which are 21.1°C IDBT and 6.1 °C OWBT. Fig. 8 shows the performance curves of the cooling and heating capacities corresponded to various outdoor air drybulb and wetbulb temperatures. For the total cooling capacity, the correction factor keeps constant in the low temperature zone and decreases as the ODBT increase in the high temperature zone. However, the correction factor (CF) for the heating capacity presents the opposite trend. In the low temperature zone, CF increases with the OWBT and remain the same within the high temperature zone.

Fig.9 shows the performance curves of cooling and heating energy input ratio (EIR) corresponded to various outdoor air drybulb and wetbulb temperatures. For the whole temperature range, the cooling EIR increases with the ODBT and the heating EIR decrease with the OWBT. In the low temperature zone, the cooling EIR increases slower than the one in the high temperature zone.

The EIR is also a function of the part load ratio as illustrated in Fig. 10. There are two ranges of part load ratio which are low PLR (0-1) and high PLR (1-1.6). For the low PLR, the cooling EIR CF shows a U-shape curve in Fig. 5. As PLR is larger than 1, it is a constant with the value of 1. In the low PLR, the heating EIR CF almost increases linearly with PLR and then decreases as PLR is larger than 1.

For VRF system, the air-cooled outdoor condenser was directly connected to the multiple indoor terminal units. The indoor terminal units were controlled by the thermostat cooling and heating schedules to meet the sensible cooling or sensible heating loads. The VRF system can simultaneously cool and heat multiple zones. Only one indoor terminal unit was chosen as the master thermostat, which determined the VRF system operating mode based on the total zone load. The heat pump will operate in cooling mode, and provide waste heat to zones with a heating load, when the dominant load among all zone terminal units is cooling. The heat pump will operate in heating mode, and

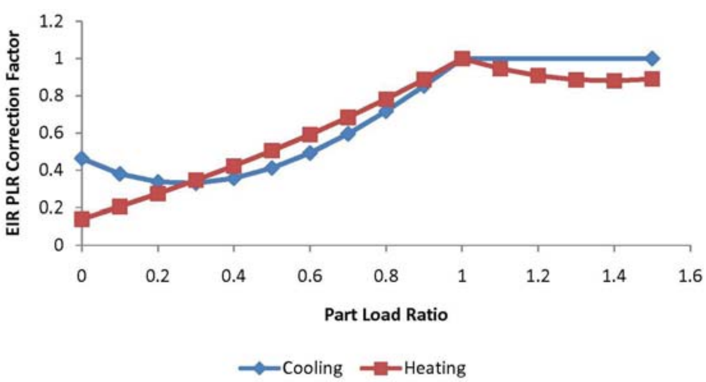

Fig. 10 VRF Cooling and heating EIR correction factor based on the part load ratio

Table 4 Test conditions for the rated cooling and heating capacities

Mode	Entering drybulb temp. (EADB)	Entering wetbulb temp. (EWAB)	Entering fluid temp.
Cooling	27°C	19°C	25°C
Heating	20°C	15°C	0°C

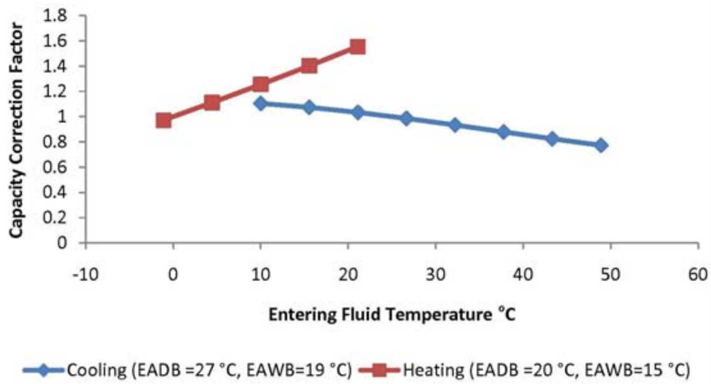

Fig. 11 GSHP heating and cooling capacity correction factor based on the entering fluid temperature

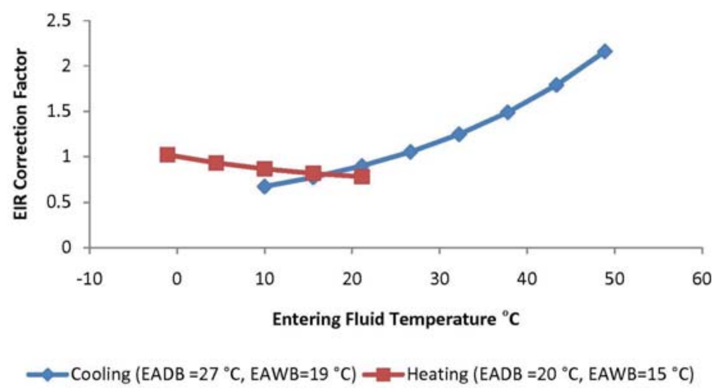

Fig. 12 GSHP heating and cooling EIR correction factor based on the entering fluid temperature

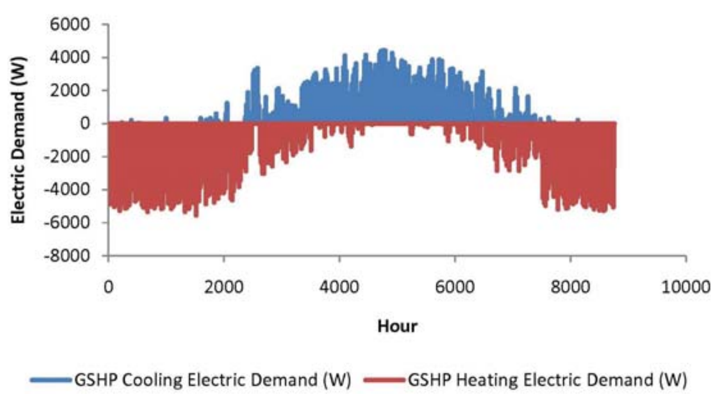

Fig. 13 Hourly Chicago cooling and heating demands for GSHP system

absorb heat from zones with a cooling load, when the dominant load among all zone terminal units is heating.

3.2 GSHP System

The variables or inlet conditions that influence the water-to-air heat pump performance are load side inlet water temperature, source side inlet temperature, source side water flow rate and load side air flow rate [9]. As per the ASHRAE standard 13256-1[10], the rated conditions for GSHP system are listed in Table 4.

As shown in Fig. 11, the cooling capacity CF of the GSHP system decreases with the entering fluid temperature. The heating capacity CF increases with the entering fluid temperature. The cooling and heating EIRs present the opposite trend in Fig. 12. When the entering fluid temperatures drop, the cooling EIR increases but the heating EIR decrease correspondingly.

According to their zone thermostats, individual heat pump units extracted heat from or rejected heat to a common water loop. The water loop connected the heat pump units with the GLHX. The loop temperature floated with the load, the features of the GLHX, and the ground temperature. The water pump attached to the water loop and circulated the water between the GLHX and the condenser. It ran intermittently with the rated pump head of 179 kPa and the motor efficiency of 0.6. The water flow rate was autosized by EnergyPlus. As a key part of GSHP system, the GLHX was used as a heat source and sink to cool or heat the condenser water. For the current study, it had 13 boreholes in one straight line. The borehole depth was 76m with the radius of 0.06 m. The U-tube spacing was 0.0254 m. The pipe thermal conductivity was 0.39 W/m.K with the outside diameter of 0.0334 m. The ground thermal conductivity and heat capacity were 2.4 W/m.K and 2160 J/m3.K. The undisturbed ground temperatures were 10.5°C, 13.9°C and 16.7°C for Chicago, Baltimore, and Atlanta, respectively.

4. Result and Discussion

For Chicago, Baltimore, and Atlanta, the hourly building cooling and heating electric demands are shown in Fig.13~18 for both systems. The results indicate that the cooling and heating electric demands vary with the locations and weather conditions. Also, the HVAC system can

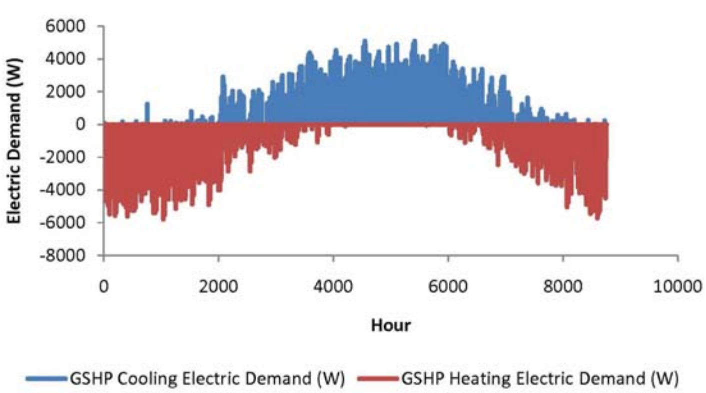

Fig. 14 Hourly Baltimore cooling and heating demands for GSHP system

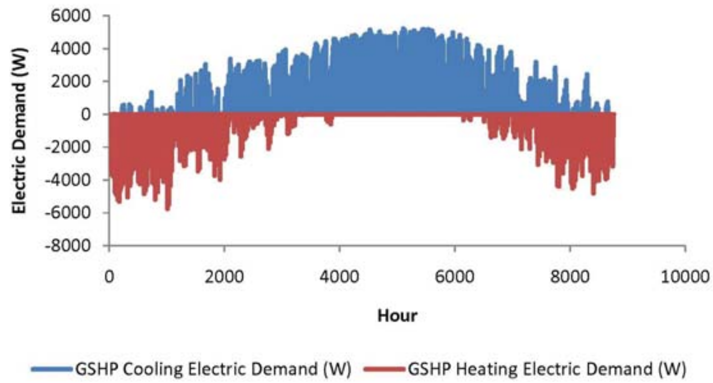

Fig. 15 Hourly Atlanta cooling and heating demands for GSHP system

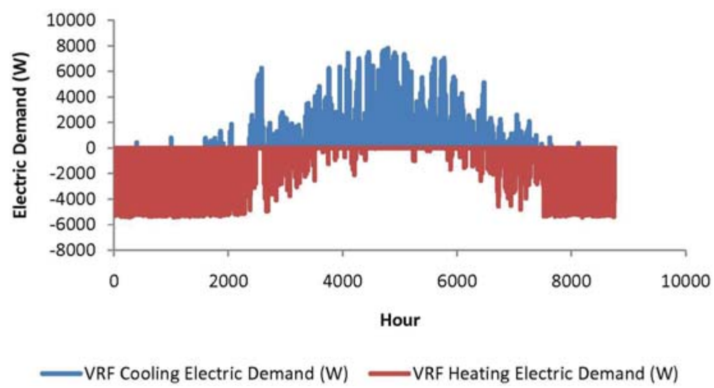

Fig. 16 Hourly Chicago cooling and heating demands for VRF system

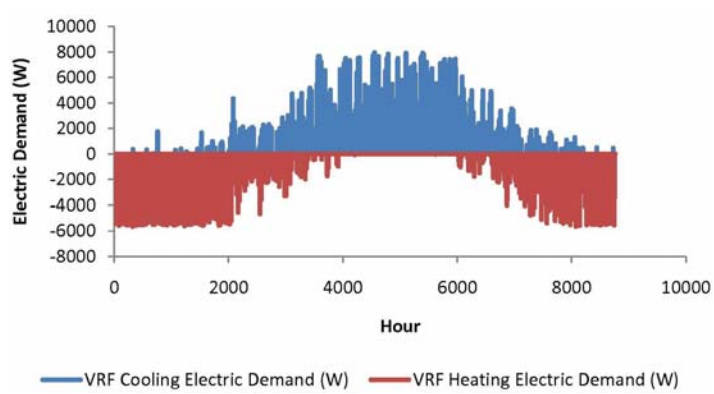

Fig. 17 Hourly Baltimore cooling and heating demands for VRF system

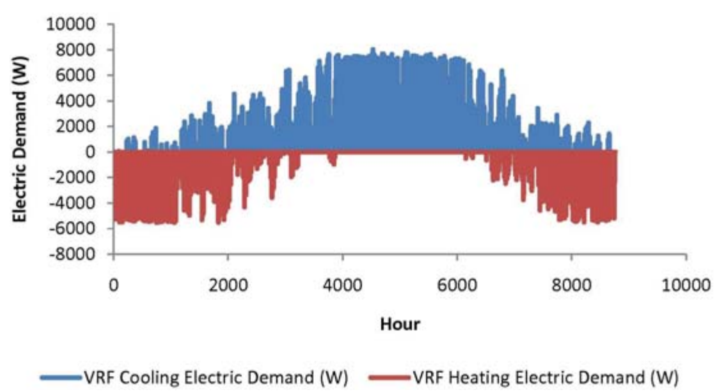

Fig. 18 Hourly Atlanta cooling and heating demands for VRF system

alter the hourly air-conditioning electric demand. GSHP system reduced the hourly HVAC air-conditioning demand as compared with VRF system.

The typical hourly electric demands for indoor fan and water pump are presented in Fig.19 and Fig. 20. Fig. 21 shows the annual HVAC electric consumptions for GSHP and VRF systems in the locations of Chicago, Baltimore and Atlanta. Table 5 presents the annual HVAC electric consumptions with different end uses for GSHP and VRF systems. Overall, GSHP system shows around 20% annual electric saving as compared with air-source VRF system in the three locations. For the locations with the substantial heating loads such as Chicago and Baltimore, the energy savings are a litter bit higher than Atlanta due to the “free heat” from the ground. GSHP system also is benefited by the low ground temperature in the hot summer where the outdoor dry-bulb and wet-bulb temperatures are much higher.

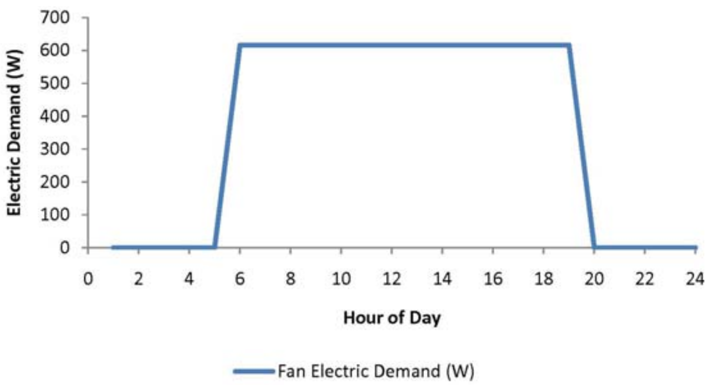

Fig. 19 Typical hourly Fan electric demand

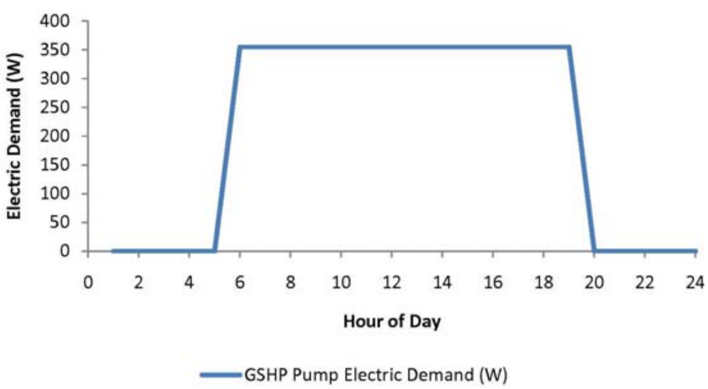

Fig. 20 Typical hourly GSHP water pump electric demand

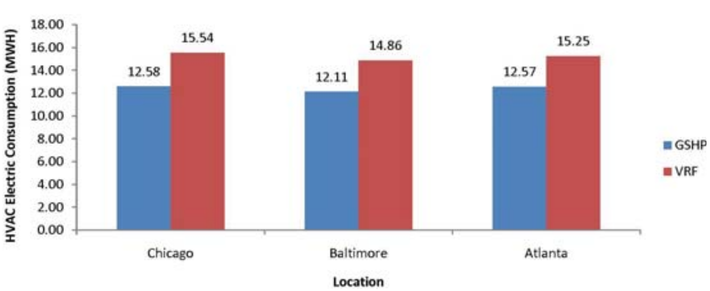

Fig. 21 Annual HVAC Electric Consumptions for GSHP and VRF Systems

Table 5 Annual HVAC Electric consumptions for GSHP and VRF systems

Location	End Use	GSHP	VRF	HVAC Electric Savings
Chicago	Heating	5.01	7.77	
	Cooling	2.86	4.60	
	Pumps	1.55	0.00	
	Fans	3.17	3.17	
	HVAC Total	**12.58**	**15.54**	**19.0%**
Baltimore	Heating	3.50	5.63	
	Cooling	3.98	6.09	
	Pumps	1.48	0.00	
	Fans	3.15	3.15	
	HVAC Total	**12.11**	**14.86**	**18.5%**
Atlanta	Heating	2.16	3.64	
	Cooling	5.81	8.43	
	Pumps	1.41	0.00	
	Fans	3.19	3.19	
	HVAC Total	**12.57**	**15.21**	**17.6%**

Table 6 HVAC Electric Peak Demands for GSHP and VRF Systems

Location		GSHP	VRF	Peak Demand Reduction
Chicago	Time of Peak	17-JUL-09:00:00	17-JUL-09:00	
	Cooling	3910	7583	
	Pumps	358	0.00	
	Fans	619	619	
	HVAC Total	**4887**	**8201**	**40%**
Baltimore	Time of Peak	09-JUL-09:00	09-JUL-09:00	
	Cooling	4512	7853	
	Pumps	355	0.00	
	Fans	616	616	
	HVAC Total	**5483**	**8468**	**35%**
Atlanta	Time of Peak	19-AUG-09:00	08-JUL-09:00	
	Cooling	4686	7632	
	Pumps	346	0.00	
	Fans	624	624	
	HVAC Total	**5657**	**8255**	31%

There has been a marked increase over the past few years in efforts to rely on energy efficiency as a utility system resource in meeting customer energy demands and keeping system costs down. Besides decreasing the annual HVAC electric consumptions, GSHP system also can reduce the Electric Peak demand in the range of 31% and 40% as shown in Table 6 in the summer which can really benefit the utilities company and their clients.

As addressed before, the outdoor air and ground loop temperatures significantly influence the performance of the VRF and GSHP systems. Fig. 22, Fig. 23 and Fig. 24 compare the hourly outdoor dry blub temperature and outlet water temperature from GLHX. For Chicago, the outlet water temperature from the GLHX was more stable as compared with the outdoor air bulb. The same trend also can be observed in Baltimore and Atlanta. Therefore, the GSHP system can achieve high efficiency and consume much less energy than the air-VRF system.

Besides the annual HVAC energy consumptions and Electric Peak demands, the unmet hours also can be used to evaluate the performance of the HVAC system. For this study, the tolerance is 0.2 °C. If the zone temperature is away from the cooling/heating setpoint by more than this value, the cooling/heating unmet hours will increment as appropriate. Table 7 lists the cooling and heating unmet hours in the three cities. The GSHP system has less unmet hours than air-source VRF system in the occupied hours.

5. Conclusion

A comparison of energy consumption between GSHP and air-source VRF systems was conducted using the whole building energy modeling program, EnergyPlus. For air-source VRF system, the EnergyPlus model used the performance curves in the EnergyPlus VRF template developed by FSRC. The performance data for GSHP system was provided by the equipment manufacturer. The results show that, for conditioning the same small office building, GSHP system uses much less energy than VRF system. For the three locations representing Mixed-Humid (Zone 4A), Cool-humid (Zone 5A) and Warm – Humid climates, GSHP system saves about 20% of annual HVAC electric energy compared with the ‘‘heat recovery’’ type VRF system. The electrical peak reduction is another advantage of GSHP system over air-source VRF system. Overall, GSHP system dramatically shrinks the Electric Peak demand by 31 to 40% as compared with the air-source VRF system. In addition to the reduction of annual energy usage and Electric Peak demand, GSHP system also shows the excellent capacity to meet the cooling and heating setpoint during the occupied hours as well as improve the thermal comfort of the occupants.

As a more energy efficient HVAC technology, GSHP system contributes to annual energy savings, peak demand reduction and thermal comfort improvement. It can help lower the customer’s electric bill. For the utilities, it can reduce the load and stress induced on various points in the power distribution network. Furthermore, it also can enhance the security of the system by decreasing the likelihood of failure at those points in the system.

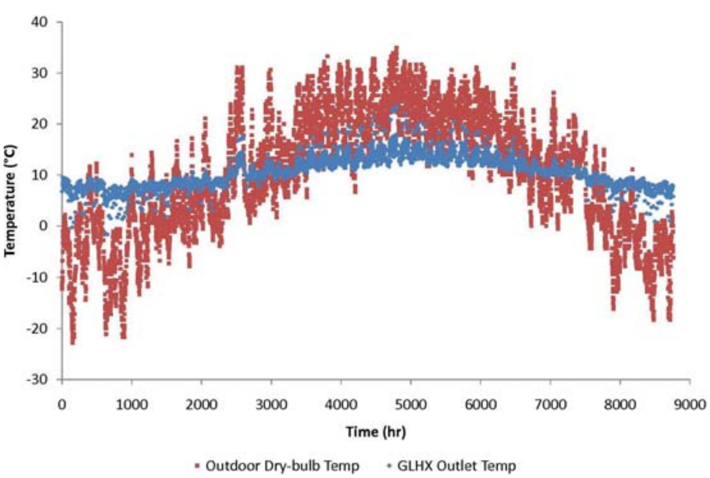

Fig. 22 GLHX Outlet and Outdoor Dry Bulb Temperatures (Chicago)

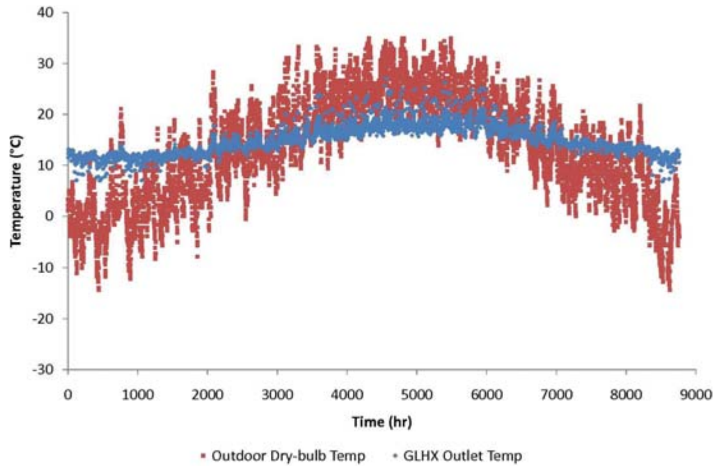

Fig. 23 GLHX Outlet and Outdoor Dry Bulb Temperatures (Baltimore)

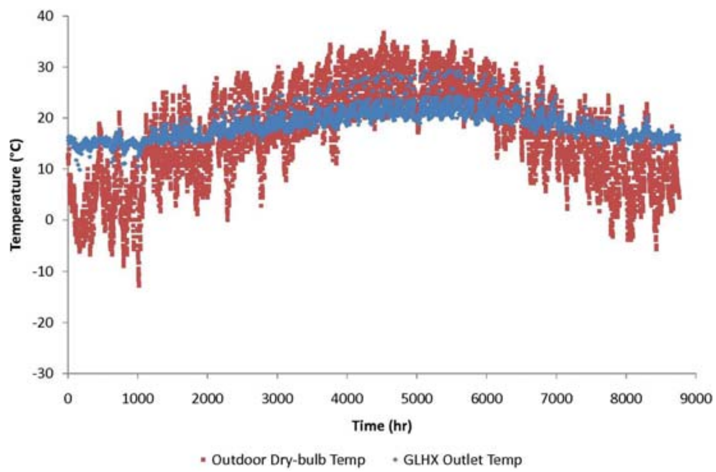

Fig. 23 GLHX Outlet and Outdoor Dry Bulb Temperatures (Atlanta)

Table 7 Time Setpoint Not Met During Occupied Hour (Tolerance: 0.2 °C)

	Chicago		Baltimore		Atlanta	
System	Cooling	Heating	Cooling	Heating	Cooling	Heating
GSHP	36	17	44	1	56	0
VRF	80	454	162	185	364	49

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