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Selecting HVAC Systems to Achieve Comfortable and Cost-effective Residential Net-Zero Energy Buildings

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Abstract

HVAC is responsible for the largest share of energy use in residential buildings and plays an important role in broader implementation of net-zero energy building (NZEB). This study investigated the energy, comfort and economic performance of commercially-available HVAC technologies for a residential NZEB. An experimentally-validated model was used to evaluate ventilation, dehumidification, and heat pump options for the NZEB in the mixed-humid climate zone. Ventilation options were compared to mechanical ventilation without recovery; a heat recovery ventilator (HRV) and energy recovery ventilator (ERV) respectively reduced the HVAC energy by 13.5 % and 17.4 % and reduced the building energy by 7.5 % and 9.7 %. There was no significant difference in thermal comfort between the ventilation options. Dehumidification options were compared to an air-source heat pump (ASHP) with a separate dehumidifier; the ASHP with dedicated dehumidification reduced the HVAC energy by 7.3 % and the building energy by 3.9 %. The ASHP-only option (without dedicated dehumidification) reduced the initial investment but provided the worst comfort due to high humidity levels. Finally, ground-source heat pump (GSHP) alternatives were compared to the ASHP; the GSHP with two and three boreholes reduced the HVAC energy by 26.0 % and 29.2 % and the building energy by 13.1 % and 14.7 %. The economics of each HVAC configuration was analyzed using installation cost data and two electricity price structures. The GSHPs with the ERV and dedicated dehumidification provided the highest energy savings and good comfort, but were the most expensive. The ASHP with dedicated dehumidification and the ERV (or HRV) provided reasonable payback periods.

Keywords

Net-zero energy building; Ventilation heat recovery; Dehumidification; Heat pump; Thermal comfort; Cost-effective

1 Introduction

Buildings accounted for 40 % of the total energy consumption in the U.S. in 2016, with residential buildings accounting for 21 % of the energy consumption [1]. Residential buildings accounted for approximately 38 % of the retail sales of electricity in 2016 [2]. This

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large amount of energy consumption resulted in the residential sector accounting for approximately 20 % of the total carbon dioxide emissions in the U.S. [1]. Therefore, advancements in the energy efficiency of residences could significantly reduce greenhouse gas emission [3].

It is toward this end that interest in net-zero energy buildings (NZEB) is increasing, where NZEB are defined here as buildings that produce at least as much energy as they use in a year when accounted for at the building site [4]. The US Department of Energy aims to achieve “marketable zero energy homes in 2020 and commercial zero energy buildings in 2025” [5]. The American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) assigned a target of market-viable NZEBs by 2030 [6].

Advances in NZEBs are mainly accomplished by minimizing the building energy demand and increasing on-site renewable energy generation.

Minimizing the building energy demand includes improving building design, using efficient appliances, integrating more efficient heating, ventilation, and air conditioning (HVAC) systems, using smart control technologies, and encouraging energy-efficient occupant behaviors [7, 8]. Improved building designs include enhanced thermal insulation, higher levels of airtightness, optimized orientation/shape, solar shading, etc. [9–11]. Adopting energy-saving lighting and efficient appliances (refrigerators, washing machines, dryers, etc.) can not only directly cut the electricity consumption but also lower the cooling load imposed on HVAC systems [12,13]. Alternative HVAC systems, such as ventilators with heat recovery [14], ground-source heat pumps (GSHP) [15], and integrated heat pumps (providing both space-conditioning and water heating) [16], can play an important role in energy efficiency. Using smart control technologies, such as smart meters and smart controllers, can also contribute greatly to the efficient operation of NZEBs [17]. Occupant behaviors, which affect indoor setpoints and load schedules, also have a great influence on the ability to achieve the NZEB goal [18].

Increasing on-site renewable energy generation includes the advances in solar, wind, and biomass [7]. Elkinton et al. [19] addressed the feasibility of wind power for NZEB developments in U.S. and found that wind turbines would be more cost beneficial in large-scale installations. Marszal et al. [20] compared a biomass/biofuel micro-combined heat and power (CHP) system with other renewable resources, and found that the micro-CHP systems were a rather costly technology and had a relatively short lifetime. On-site electricity generation with a solar photovoltaic (PV) system is currently the dominant technology [21] due to its easy integration with building roofs and facades, and the fact that there is an insignificant marginal cost difference between small and large-scale installations [20]. Furthermore, PV production is improving through the adoption of advanced photovoltaic technologies [22].

The literature review indicated that it was important to explore efficient HVAC technologies for NZEBs. Even with advanced envelope construction, the HVAC system remains responsible for the largest share of energy use in residential buildings, greater than 40 % [23]. Although prior research has analyzed the advantages and disadvantages of different

HVAC systems [14, 24–26], the studied HVAC options were limited, and the evaluation criteria mainly focused on energy performance or economic benefit without considering thermal comfort.

In this work, we studied various HVAC configurations for a residential NZEB constructed on the National Institute of Standards and Technology (NIST) campus in Gaithersburg, Maryland, USA (ASHRAE Climate Zone 4, mixed-humid [27]). Using a validated building energy model, the house and each HVAC subsystem were considered in detail, including the ventilation, dehumidification, and heat pump equipment. We compared three commercially-available options for each subsystem based on energy performance, thermal comfort, and economic performance. The goal of this study was to identify the relative merits of alternative HVAC technologies and provide recommendations for HVAC system design, in support of broader implementation of residential NZEBs in North America.

2 Description of the residential NZEB

2.1 General information

The studied residential NZEB, which is the basis for the model described in Section 3, was constructed on the NIST campus in Gaithersburg, Maryland, USA in 2012 (Fig. 1). The single-family house had two stories (251 m²) of living area and a full conditioned basement (135 m²). The first floor comprised the kitchen and dining room, a family room, an office, and a full bathroom. The second floor included a master bedroom with adjoining bathroom, two additional bedrooms and two bathrooms. The house featured a wide array of energy efficient technologies. Increased insulation, double-paned windows, and very tight construction (0.63 air change per hour measured by a blower-door test at 50 Pa) were used in the building envelope to reduce heating and cooling loads [28].

The house was unoccupied, but the activities associated with a family of four were emulated by computer activating appliances, plug loads, lighting, water draws, and devices that generate sensible and latent heat. These loads followed a weekly schedule [29], which was based on standard user profiles developed for the U.S. Department of Energy Building America program [30]. Each occupant was assumed to generate a constant 70 W of sensible heat and 45 W of latent heat (averaged for adults and children), which were emulated using resistance heaters and ultrasonic humidifiers. The weekly loads for lighting, plug, and appliances were approximately 8.5 kWh, 47.2 kWh, and 36.4 kWh. The total water volume used in a week was approximately 2229 L.

During the first year of operation, the NIST NZEB exceeded the net-zero energy goal by generating 484 kWh more electrical energy than it consumed. Detailed specifications and performance data can be found in [10, 28, 31, 32].

2.2 Electricity flow

Figure 2 maps the electricity flow of the house. The house used thirty-two 320 W positively-grounded monocrystalline silicon PV modules for a total rated capacity of 10.24 kW. The modules were mounted approximately 14 cm above the surface of the roof facing south at an angle of 18.4°, and were arranged in four rows of eight. Two inverters with a nominal

efficiency of 95.5 % were used to maximum power track the modules and convert their direct current to 60 Hz alternating current. The net electricity imported from or exported to the utility grid was determined by accounting the electricity generated by the PV inverters and the electricity consumed by all the loads within the house.

2.3 HVAC system

The HVAC system consisted of an ASHP and a heat recovery ventilator (HRV) (Fig. 3). The ASHP was a split-system with a two-stage compressor and a 10 000 W electric heater. The system provided three heating stages, two cooling stages, and two dedicated dehumidification stages. According to AHRI Certificate of Product Ratings [33], the rated cooling capacity was 7067 W with a cooling COP of 2.92, and the rated heating capacity was 8041 W with a heating COP of 3.78. The heat pump operation was both temperature- and time-triggered by the thermostat. The temperature setpoint was 20.5 °C in the heating mode and 23.9 °C in the cooling mode, and the relative humidity setpoint was 48 %. The first-stage heating turned on when the thermostat read a temperature 0.1 °C below the setpoint. The second stage turned on when the first stage had been running for 10 min or the temperature dropped 1.1 °C below the setpoint. The third stage used the compressor in second stage and an electric heater, and turned on when the second stage had been running for 40 min or the temperature dropped 3.3 °C below the setpoint. The first stage cooling turned on when the thermostat read a temperature 0.2 °C above the setpoint. The second stage turned on when the first stage had been running for 40 min or the thermostat read a temperature 2.8 °C above the setpoint. The dedicated dehumidification activated only if the cooling mode was not in operation and the relative humidity was greater than 50 %. The first-stage dehumidification operated in the cooling mode with a lower fan speed. The second-stage dehumidification activated when the first stage had been running for 6 min but the humidity setpoint was not achieved. The second-stage dehumidification also operated in cooling mode, but reheated the supply air (to avoid overcooling) using a portion of hot refrigerant vapor that bypassed the condenser.

The HRV acted as a sensible heat exchanger between the supply of outdoor air and the exhaust air from the indoors. The HRV provided approximately 195 m³/h (0.15 air change per hour) of outdoor air in compliance with ASHRAE Standard 62.2 [34]. The HRV ran continuously, but based on the manufacturers' recommendation it entered a defrost mode when the outdoor air temperature fell below −10 °C. During defrost, the unit recirculated the indoor air for 7 min, then returned to normal operation for 22 min.

2.4 Domestic hot water (DHW) system

The DHW system consisted of a solar hot water (SHW) subsystem and a heat pump water heater (HPWH) subsystem (Fig. 4). The SHW subsystem preheated the water, which was then heated to the setpoint by the HPWH, if necessary, before being distributed to its end use point. The SHW subsystem used a 303 L storage tank and two single-glazed flat plate solar collectors, with each collector having an aperture area of 2.1 m² and facing south at an 18.4° tilt. The solar collectors used a 50 % by volume propylene glycol and water solution. The pumps turned on when the temperature difference between the solar collector outlet and the water tank bottom exceeded 10 °C and turned off when the difference was less than 3 °C.

The pumps also turned off when the temperature at the tank bottom exceeded 71 °C. Water exited from the top of the SHW tank to enter a tempering valve (thermostatic mixing valve), which tempered the water to a maximum of 49 °C before it entered the HPWH tank. The HPWH subsystem used an integrated HPWH and tank with a 3800 W resistive heating element. After exiting the HPWH tank, the hot water was delivered to a manifold and then to the end-use fixtures. Hot and cold water was mixed at the fixtures to achieve use-specific temperatures including: 41 °C for sinks and showers, 43 °C for baths, and 49 °C for the dishwasher and clothes washer.

3 Methods

We studied various HVAC technologies using a validated model of the residential NZEB. Section 3.1 through Section 3.3 discuss the model details and validation, alternative HVAC options, and evaluation indices.

3.1 Model of the residential NZEB

3.1.1 Model description—The components and main parameters of different subsystems of the TRNSYS [35] model of the NZEB are provided in Table 1. The model is described briefly here; more detail can be found in [36]. TRNSYS Type 56 was used to model the building, with the layers of material used in the walls, roof, floors, and ceilings specified in accordance with the building plans [37].

The PV subsystem was modeled using two instances of TRNSYS Type 194b, a photovoltaic array and inverter model. Type 922 and Type 667 were used to model the ASHP and HRV, respectively. Three instances of Type 922 were used to simulate the first and second stage heating/cooling, the third stage heating, and the dedicated dehumidification mode (Fig. 3). We used the measured capacity and power data to formulate a TRNSYS performance map for each mode. The defrost cycle is neglected by the TRNSYS ASHP type, therefore the defrost feature was added using a custom type developed as part of this study based on the measured electrical energy during defrost time periods. The defrost cycle was activated after 90 min of accumulated compressor runtime while the outdoor temperature was below 1.7 °C. The heat pump's measured electrical energy per defrost cycle ($E_{defrost}$) was linearly correlated to the outdoor dry-bulb temperature (t_{out}) according to Equation (1):

$$E_{defrost} = -0.0186t_{out} + 0.4005 \quad (1)$$

To model the SHW subsystem, Type 1b, Type 534, Type 91, Type 114, and Type 31 were used respectively as solar collectors, solar hot water storage tank, heat exchanger, pumps and pipes. To model the HPWH subsystem, Type 938, Type 534, and Type 1226 were used to simulate the heat pump water heater, cylindrical storage tank, and auxiliary electric heater, respectively.

Weather data measured on-site (dry-bulb temperature, wet-bulb temperature, and solar irradiation) and at an airport (KGAI) located 6.4 km away from the house (atmospheric pressure, wind direction and speed, and total sky cover) were converted and read by Type 15.

A time step of one minute was used because the equipment controls operate at approximately this time scale.

3.1.2 Model validation—The model for the residential NZEB was validated with 1 year of measured data [31]. The measurement uncertainty associated with the HVAC system, as well as all other electrical/mechanical subsystems within the NZEB were described in detail [32]. The measured and predicted monthly integrated thermal load of the building agree within 4 % for heating and within 5 % for cooling (Fig 5(a)). The relative deviations of subsystem energy consumption were between 0.9 % and 7.7 % (Fig. 5(b)). The measured and predicted power generated by the PV system agreed within 3.1 % for the annual total, and agreed within an average of 6.6 % for the monthly totals (Fig 6(a)). The high deviation in February and March (Fig 6(a)) was caused by snow covering the actual PV panels, which was not included in the model. The difference between the predicted and measured monthly power consumption of the ASHP was 13 % for cooling and 10 % for heating (Fig. 6(b)). The predicted average water temperatures at the inlet and outlet of the SHW tank and HPWH tank matched the measured results closely (Fig. 7).

3.2 Alternative HVAC options

We evaluated the HVAC system with several options for each of the ventilation, dehumidification, and heat pump subsystems. For each subsystem study, the equipment in the other subsystems was held constant: (1) for ventilation options, we compared three options with the same dehumidification and heat pump subsystems; (2) for dehumidification options, we compared three options with the same ventilation and heat pump subsystems; (3) finally, for heat pump options, we compared three options with the same ventilation and dehumidification subsystems.

3.2.1 Ventilation options—This study compared a mechanical ventilation system (a) without heat recovery, (b) with an HRV and (c) with an ERV (energy recovery ventilator, i.e., sensible and latent recovery).

The system without heat recovery simply brought in outdoor air. It was assumed that this system used only one supply fan, which used half of the power required by the HRV. The ERV was like the HRV (same size and same defrost cycle) except the ERV core was porous to moisture. Both the ERV and the HRV cores were crossflow exchangers and had 10.2 m² of surface area. The ERV core was made of polymerized paper, while the HRV core was made of polypropylene. Similar to the HRV, the ERV was also modeled using Type 667b. Curve fits to the manufacturer's specifications represented the sensible effectiveness (η) and the power (P) as a function of the air flowrate (m):

$$\eta = \eta_0 \left[c_0 + c_1 \frac{m}{m_0} + c_2 \left(\frac{m}{m_0} \right)^2 \right] \quad (2)$$

$$P = P_0 \left[c_0 + c_1 \frac{m}{m_0} + c_2 \left(\frac{m}{m_0} \right)^2 \right] \quad (3)$$

where η_0 , P_0 , and m_0 are the rated sensible effectiveness, power, and mass flow rate, respectively; c_0 , c_1 , and c_2 are the coefficients. Table 2 lists the main parameters for each ventilation option.

3.2.2 Dehumidification options—The study compared the following dehumidification options: (a) a single-packaged separate whole-house dehumidifier, (b) an ASHP with no dedicated dehumidification (ASHP only), and (c) an ASHP with dedicated dehumidification.

The separate dehumidifier used the existing ASHP ducts and only operated when the ASHP was off. The unit delivered air that was drier but warmer than the return air since the condenser heat was rejected to the supply airstream. To model the dehumidifier, Type 921, a residential/commercial air conditioner was used. Measured power and capacity data from two days during which the dehumidifier ran for substantial amounts of time were used to develop the performance maps as functions of the return air relative humidity. The separate dehumidifier was sized to provide (together with the ASHP) similar thermal comfort to the ASHP with dedicated dehumidification (as discussed in Section 4.2.2). Type 921 was unable to both dehumidify and reheat the air, so the dehumidifier was modeled as having only latent capacity. An auxiliary air heater (Type 121b) was used to heat the dry air exiting the dehumidifier to correspond with measurements (by adding back the latent load plus compressor power); the energy use of this auxiliary heater was not counted towards building energy consumption. The rated cooling capacity of the dehumidifier was 733 W, and the rated power was 1300 W.

3.2.3 Heat pump options—Although an ASHP has been used for the 1-year measurements, the residential NZEB was also equipped with three vertical 45 m boreholes for use with a GSHP. We considered the GSHP integrated with both 2 and 3 to boreholes to understand the marginal differences in initial cost and operating efficiencies with varied borehole length. Table 3 lists the model components and their main parameters.

The studied “mid-price” two-speed, two-ton GSHP unit had cooling and heating capacities of 6686 W and 4962 W and COPs of 4.4 and 3.5, respectively, at rated conditions specified by AHRI/ISO 13256-1 [38]. The GSHP was modeled using Type 919, a water-source heat pump model with normalized performance. The performance under various airflows, air temperatures, and water temperatures from the manufacturer specifications were used to generate performance maps for the GSHP in TRNSYS. The GSHP model did not have a dedicated dehumidification mode; to provide a fair comparison with the baseline ASHP, this capability was added in the simulation. The model of the GSHP dedicated dehumidification assumed that the ratio of dehumidification capacity to cooling capacity (as well as for power) is the same as that of the ASHP in each stage.

The vertical boreholes were spaced 6.4 m on average, each reaching a depth of 45 m. They were modeled using the duct ground heat storage model (Type 577) [39]. The thermal conductivity and heat capacity of the ground were computed from a thermal response test and discussed in [37, 40]. The heat transfer fluid used in the GSHP was a solution of 75 % water, 20 % ethanol, and 5 % isopropanol (percentages are by mass). The fluid properties were taken from the manufacturer's catalog.

3.3 Evaluation criteria

3.3.1 Energy performance—The building total energy consumption ($E_{building}$) was the sum of each subsystem consumption, including lighting ($E_{lighting}$), plug loads (E_{plug}), appliances ($E_{appliance}$), DHW (E_{DHW}), heat pump (E_{HP}), and ventilation ($E_{ventilation}$):

$$E_{building} = E_{lighting} + E_{plug} + E_{appliance} + E_{DHW} + E_{HP} + E_{ventilation} \quad (4)$$

ASHP and GSHP energy consumption (E_{HP}) were calculated by summing the consumption of heating ($E_{heating}$), defrost ($E_{defrost}$), cooling ($E_{cooling}$), dehumidification ($E_{dehumidification}$), and standby mode ($E_{standby}$):

$$E_{HP} = E_{heating} + E_{defrost} + E_{cooling} + E_{dehumidification} + E_{standby} \quad (5)$$

For the heat pump option investigation, we compared the COPs of ASHP and GSHP, with the COP calculated as capacity divided by electricity consumption:

$$COP = \frac{Q_{sensible} + Q_{latent}}{E_{HP}} \quad (6)$$

where $Q_{sensible}$ and Q_{latent} are the sensible and latent capacity, respectively.

The energy saving ratio (ESR) of the alternative option, compared to the baseline option, was defined as:

$$ESR = \frac{E_{baseline} - E_{alternative}}{E_{baseline}} \times 100 \% \quad (7)$$

where $E_{baseline}$ and $E_{alternative}$ are the energy use of the baseline option and alternative option.

The annual net energy (E_{net}) was the PV generation (E_{PV}) minus the total building energy consumption:

$$E_{net} = E_{PV} - E_{building} \quad (8)$$

For energy analysis, the installed PV capacity of each HVAC option was kept the same as the value shown in Table 1. Note that, for economic analysis, the PV capacity was adjusted to maintain the same net energy, as discussed in the results.

3.3.2 Thermal comfort—The predicted mean vote (PMV) and predicted percentage of dissatisfied (PPD) were used to evaluate the thermal comfort performance of the various HVAC options [41]. The PMV index predicts the mean value of the votes of a large group of persons on a seven-point thermal sensation scale, while the PPD index is an estimation for the number of persons who are dissatisfied with the thermal conditions [42]. The PMV thermal sensation scale is defined as follows: +3 (hot), +2 (warm), +1 (slightly warm), 0 (neutral), -1 (slightly cool), -2 (cool) and -3 (cold). The acceptable thermal environment for general comfort is $PPD < 10\%$, corresponding to $-0.5 < PMV < +0.5$ [42]. Thermal comfort calculation was based on Fanger's PMV-PPD model [41–43]:

$$\begin{aligned} PMV = & (0.303e^{-0.036M} + 0.028)\{(M - W) \\ & - 3.05 \times 10^{-3}[5733 - 6.99(M - W) - p_a] - 0.42[(M - W) - 58.15] \\ & - 1.7 \times 10^{-5}M(5867 - p_a) - 0.0014M(34 - t_a) \\ & - 3.96 \times 10^{-8}f_{cl}[(t_{cl} + 273)^4 - (\bar{t}_r + 273)^4] - f_{cl}h_c(t_{cl} - t_a)\} \end{aligned} \quad (9)$$

$$\begin{aligned} t_{cl} = & 35.7 - 0.028(M - W) - I_{cl}\{3.96 \times 10^{-8}f_{cl}[(t_{cl} + 273)^4 - (\bar{t}_r + 273)^4] + f_{cl}h_c(t_{cl} - t_a)\} \\ & (10) \end{aligned}$$

$$h_c = \begin{cases} 2.38(t_{cl} - t_a)^{0.25}, & 2.38 \times |t_{cl} - t_a|^{0.25} > 12.1\sqrt{V_{ar}} \\ 12.1\sqrt{V_{ar}}, & 2.38 \times |t_{cl} - t_a|^{0.25} < 12.1\sqrt{V_{ar}} \end{cases} \quad (11)$$

$$f_{cl} = \begin{cases} 1.00 + 1.290 \times I_{cl}, & I_{cl} \leq 0.078 \text{ m}^2 \cdot \text{K/W} \\ 1.05 + 0.645 \times I_{cl}, & I_{cl} > 0.078 \text{ m}^2 \cdot \text{K/W} \end{cases} \quad (12)$$

where M is the metabolic rate, W/m^2 ; W is the external work, W/m^2 (equal to zero for most activities); p_a is the water vapor partial pressure, Pa; t_a is the indoor air temperature, $^{\circ}\text{C}$; f_{cl} is the ratio of surface area of the body with clothes to the surface area of the body without clothes; t_{cl} is the clothing surface temperature, $^{\circ}\text{C}$; $\bar{t}_r$ is the mean radiant temperature, $^{\circ}\text{C}$; I_{cl} is the clothing thermal resistance, $\text{m}^2 \cdot ^{\circ}\text{C/W}$; V_{ar} is the relative air velocity, m/s; and h_c is the convective heat transfer coefficient, $\text{W}/(\text{m}^2 \cdot ^{\circ}\text{C})$. In this study, the clothing level was set to 0.36 clo (1 clo = $0.155 \text{ m}^2 \cdot ^{\circ}\text{C/W}$) for May-Sept and 0.6 clo for the other months; the activity

level was set to 98.9 W/m² (nominally 1.7 met); and the air velocity was assumed to be 0.05 m/s [41, 42].

The thermal comfort analyses were based on predicted zone average indoor temperatures and relative humidities, without accounting for local variation and air speed that might be impacted by air distribution and other effects. The zone mean radiant temperature was assumed to be the area weighted mean surface temperature of all the surface of a zone [35] (each floor was a zone):

$$\bar{t}_r = \frac{t_1 A_1 + t_2 A_2 + \dots + t_n A_n}{A_1 + A_2 + \dots + A_n} \quad (13)$$

where, t_n is the surface temperature of area n , °C, which was calculated in TRNSYS based on the heat transfer between the indoor air and surface in the zone; A_n is the surface area of area n , m², which was also calculated in TRNSYS based on the input geometry of the house.

The PPD index can be computed from the PMV value by the following relation [42]:

$$\text{PPD} = 100 - 95 \exp(-0.03353 \text{PMV}^4 - 0.2179 \text{PMV}^2) \quad (14)$$

3.3.3 Economic analysis—For each HVAC system, the installed PV capacity was adjusted to maintain the same net energy. Since the cost for other components of the NZEB was not changed by HVAC options, we used the summed cost of PV and HVAC to evaluate the difference between the various HVAC systems:

$$C_{PV+HVAC} = C_{PV} + C_{ventilation} + C_{dehumidification} + C_{HP} \quad (15)$$

where C_{PV} , $C_{ventilation}$, $C_{dehumidification}$, and C_{HP} are the cost of the PV, ventilation, dehumidification, and heat pump, respectively.

The installed cost of PV depends on its installed capacity, with the installed price for residences in different states presented in Fig. 8 [44]. The median price of \$3.9/W in Maryland in 2015 was used in this study. Table 4 lists the installed cost estimates of the HVAC equipment options (including ductwork) for the NZEB [45] and the median installation cost of the ground loops [46, 47]. The median ground loop cost was \$49.02/m in 2013 [46] and was inflated using an annual rate of 2.65 % derived from reference [47], which resulted in a cost of \$53.02/m in 2016. Note that no tax credits, tax deductions, rebates, or any other financial incentives have been applied to the costs.

The electricity price of \$0.1530/kWh for import and \$0.0866/kWh for export was used in this study [48]. Two electricity metering policies were used: (1) “net-metering”, where the net electricity was summed annually and surplus was credited at the export price; (2) “instantaneous markup”, where all imported and exported electricity was accounted

instantaneously at the corresponding prices. This analysis showed the sensitivity of the economic conclusions to the electric metering policy. Under “net-metering”, all HVAC options had the same annual electricity cost (or credit) since the PV capacity was adjusted to achieve the same net annual energy; therefore, the options were compared based on initial cost alone. Under “instantaneous markup”, the HVAC options all had different annual electricity costs because of unique ratios of imported/exported electricity; the options were compared based on payback period, relative to a baseline system. The current local Maryland utility uses “net-metering”. The “instantaneous markup”, with reduced incentive for PV production, represented a utility that was less willing to bear the costs of managing the exported power.

4 Results and discussions

4.1 Comparisons of ventilation options

This section compares the ventilation system performance with the HRV and ERV against performance without heat recovery.

4.1.1 Energy performance—The three ventilation systems introduced different sensible and latent loads (Fig 9), and led to different ASHP energy consumption. The thermal load is the monthly cumulative thermal energy required to meet the indoor parameters. We used kWh as the unit to be consistent with the electricity consumption. Both the HRV and ERV greatly reduced the sensible load through exhaust heat recovery. The sensible load reduction was especially significant in winter when the temperature difference between indoor and outdoor air was the greatest. Compared to the ERV, the HRV introduced a slightly lower sensible load due to the marginally higher sensible effectiveness. The latent loads (Fig. 9(b)) indicated that ERV introduced the least latent load in summer, when there was a greater difference between the indoor and outdoor air relative humidity.

Fig. 10 depicts the thermal loads and energy consumption of the ASHP under the three ventilation options. Note that the negative value denotes the heating load in winter, while the positive value denotes the cooling load in summer. Without heat recovery, the load on the ASHP was the largest, particularly in winter. The HRV and the ERV reduced the ASHP load by a similar amount in winter, but the ERV provided more benefit in summer than the HRV because of its latent capacity.

Table 5 lists the annual electricity consumption and generation under different ventilation options. Compared to the ventilation without heat recovery, the electricity consumption with HRV was reduced from 7381 kWh to 6143 kWh (1238 kWh reduction) for the ASHP and increased from 226 kWh to 440 kWh (214 kWh increase) for ventilation. Electricity consumption with the ERV was reduced by 1578 kWh for the ASHP and increased by 255 kWh for ventilation. Put another way, the HRV and ERV saved HVAC energy by 13.5 % and 17.4 %, and saved building electricity consumption by 7.5 % and 9.7 %, respectively. These reductions in energy use resulted in an increase in net electricity generation of 1022 kWh for the HRV and 1322 kWh for the ERV.

4.1.2 Thermal comfort—The analyses of thermal comfort included PMV and the cumulative number of “uncomfortable” hours where $PPD > 10\%$ ($|PMV| > 0.5$) [42] (Fig. 11 and Table 6). Ventilation without heat recovery had a slightly superior comfort related to better air circulation to the 2nd floor due to longer heat pump run times. Nevertheless, the thermal comfort indexes for the three options were quite close. All options exhibited a few outliers with PPD above 20 % in the shoulder seasons (spring and fall) when the thermostat is frequently switching between heating and cooling setpoints, but the simulated occupants were not making a commensurate clothing adjustment (the model adjusted clothing based only on season).

4.1.3 Economic analysis—Using the option without heat recovery as a baseline, the PV capacities were reduced from 10.2 kW to 9.5 kW and 9.2 kW with the application of HRV and ERV respectively, while maintaining the same net energy. The costs of different ventilation options are compared in Table 7. For the HRV, the PV cost was reduced from \$39 780 to \$36 908 (\$2872 reduction) at the price of \$3736 more HVAC cost. As for the ERV, the PV cost was reduced from \$39 780 to \$36 063 (\$3717 reduction) with an additional \$4327 paid for the HVAC system. With “net-metering”, the NZEB obtained \$43/yr income (negative operation cost) due to the surplus electricity for all the options. With the “instantaneous markup” policy, the HRV and the ERV saved the operation cost by \$57 and \$68, with a simple payback of 15.4 and 9.0 years, respectively.

4.2 Comparisons of dehumidification options

This section compares the residential NZEB with the separate dehumidifier, the ASHP only, and the ASHP with dedicated dehumidification.

4.2.1 Energy performance—The three options had significantly different energy removal and energy consumption (Fig. 12) in the dehumidification mode. The difference between the options peaked in July, during the time with the highest outdoor humidity, and there was little difference between the options in winter when the dehumidification mode was not active. The dehumidification energy was only counted when the equipment operated specifically to dehumidify, i.e., the ASHP in dehumidification mode, or the separate dehumidifier. Latent capacity of the ASHP operating in cooling mode was not accounted for in Fig. 12, which is why the “ASHP only” configuration had no dehumidification energy. The ASHP in dedicated dehumidification mode had both sensible and latent capacity, and therefore the dehumidification energy removal and consumption was the highest (Fig. 12). This sensible capacity offset part of the load for the ASHP in cooling mode.

Fig. 13 and Table 8 present the thermal loads and energy consumption of the ASHP under the three dehumidification options. The ASHP with the separate dehumidifier consumed the most energy, 6659 kWh, because all the heat rejection from the dehumidifier became indoor heat gain. On the contrary, the heat generated due to the ASHP dedicated dehumidification was rejected outdoors through the ASHP condenser, so the ASHP energy consumption was smaller at 6143 kWh, which was 7.3 % less HVAC energy and 3.9 % less total building energy than with the separate dehumidifier. The ASHP electricity consumption under dedicated dehumidification was lower than with the separate dehumidifier by 516 kWh

(6659 kWh versus 6143 kWh), leading to 512 kWh higher net generation after a whole year operation. Though the ASHP-only option consumed the least energy, 5097 kWh, the house had higher humidity and PPD (as discussed in Section 4.2.2).

4.2.2 Thermal comfort—Fig. 14 and Table 9 show the thermal comfort for the three dehumidification options. The differences between the options were the biggest in the summer when outdoor humidity was the highest. The ASHP-only option had the highest number of hours PPD>10% in summer (1649 h) because humidity was not explicitly controlled, compared to those with the separate dehumidifier (855 h) and the dedicated dehumidification (809 h).

We sized the separate dehumidifier in such a way that the number of summer hours with relative humidity above 50 % was within 5 % deviation compared to the dedicated option. Simulations revealed that this number reached 2463 h for the ASHP-only option, 559 h for the separate dehumidifier option, and 584 h for the dedicated dehumidification option, with average relative humidities of 54.8 %, 47.4 %, and 48.0 %, respectively. Apart from thermal comfort, an indoor relative humidity between 30 % and 50 % is also desired to control hydrolysis (volatile organic compound emissions) and microbial (virus, bacteria, molds, mites, etc.) and to enable comfort in respiratory, skin and mucous membranes [49].

4.2.3 Economic analysis—Using the option with the separate dehumidifier as a baseline, the PV capacities for the ASHP only and the dedicated dehumidification were reduced from 10.2 kW to 9.1 kW and 9.8 kW while maintaining the same level of net electricity generation. Table 10 compares the cost of different dehumidification options. For dedicated dehumidification, the PV cost was reduced from \$39 780 to \$38 341 (\$1439 reduction), and the HVAC cost was decreased from \$34 629 to \$32 775 (\$1854 reduction). Therefore, the total cost of HVAC and PV was reduced from \$74 409 to \$71 116, a reduction of \$3293. With “net-metering”, the NZEB obtained \$87/yr income (negative operation cost) due to the surplus electricity for all the options. With “instantaneous markup”, the dedicated option saved the operation cost by \$19/yr. Although the ASHP-only option was the least expensive, the thermal comfort and humidity control was the worst as discussed in Section 4.2.2.

4.3 Comparisons of heat pump options

The studied heat pump options included ASHP, GSHP with 2 boreholes, and GSHP with 3 boreholes.

4.3.1 Energy performance—Fig. 15 shows the thermal loads and energy consumption of the three heat pump options. The GSHP with 3 boreholes consumed less energy than the GSHP with 2 boreholes, but the difference was not significant. Both systems consumed much less energy than the ASHP, with the highest differences of 443 kWh and 469 kWh in January.

Fig. 16 illustrates the hourly heat source and sink temperatures and heat pump COPs, from Jan 1 to Dec 31. The air temperature is shown for the ASHP, and the entering liquid temperature is shown for the GSHP. For the ASHP, the minimum and maximum temperatures were -17°C and 34°C , respectively. For the GSHP, the minimum and

maximum entering liquid temperatures were 2.4 °C and 31.7 °C with 2 boreholes, and 5.6 °C and 24.7 °C with 3 boreholes. The favorable source and sink temperatures for the GSHP, relative to the ASHP, resulted in an increased COP by up to 2.0 in winter and 1.0 in summer. Note that the outlier COPs below 1.0 occurred during operation in dedicated dehumidification mode.

The long-term performance of GSHPs depends on the soil heat balance and soil temperature [50]. The heat balance accounts for the energy that is extracted/rejected by the GSHP from/to the soil, and transferred to/from the storage volume from/to the surrounding air and soil; these heat transfers are outputs of the borehole model (Type 557). The annual summation of these is the energy imbalance that either raises or lowers the average soil temperature each year. The imbalance ratio (energy imbalance divided by energy in or out of soil, whichever is greater) was only 7 % for the GSHP with 2 boreholes and 2 % for the GSHP with 3 boreholes (Fig. 17), owing to balanced building loads in the studied climate and insignificant thermal interference between adjacent boreholes in this small-scale application. Since the imbalance and associated change in average soil temperature was small (beginning the year at 12.4 °C and ending the year at 12.2 °C for 2 boreholes and 12.4 °C for 3 boreholes), it was not necessary to simulate the GSHP for multiple years; rather, the first year of GSHP operation was considered representative of all operating years.

Table 11 shows the heat pump electricity consumption breakdown for the different options. The GSHP not only benefited from favorable heat source and sink temperatures, but also avoided using 360 kWh required by the ASHP for defrost operation. The energy savings in the heating season were greater than that in the cooling season, with the total heating energy consumption reduced by 996 kWh for 2 boreholes and 1091 kWh for 3 boreholes, and the total cooling consumption reduced by 725 kWh for 2 boreholes and 843 kWh for 3 boreholes. The annual electricity consumption reduction was 1721 kWh for the GSHP with 2 boreholes and 1934 kWh for the GSHP with 3 boreholes. The annual COP of the ASHP was 2.42, compared to 3.21 for the GSHP with 2 boreholes, and 3.38 for the GSHP with 3 boreholes.

Table 12 provides the annual electricity consumption and generation under different heat pump options. The GSHP with 2 boreholes and 3 boreholes had significantly lower electricity consumption than the ASHP, 4422 kWh and 4209 kWh versus 6143 kWh, with minimal change in the ventilation energy. This corresponds to HVAC energy savings of 26.0 % and 29.2 %, and building energy savings of 13.1 % and 14.7 %. These energy savings for the GSHP with 2 boreholes and 3 boreholes resulted in 3171 kWh and 3384 kWh annual net generation, which were 1691 kWh and 1899 kWh above the ASHP value, respectively.

4.3.2 Thermal comfort—The GSHP and ASHP provided similar comfort; the GSHP provided marginally better comfort due to longer run times, which was caused by its slightly lower capacity (Fig. 18 and Table 13).

4.3.3 Economic analysis—Using the ASHP option as a baseline, the PV capacities with the GSHP were reduced from 10.2 kW to 9.0 kW for 2 boreholes and 8.8 kW for 3

boreholes. For the GSHP with 2 boreholes, the PV cost was reduced from \$39 780 to \$35 028 (\$4752 reduction) with an increase of HVAC cost by \$13 036 (Table 14). As for the GSHP with 3 boreholes, the PV cost was reduced from \$39 780 to \$34 442 (reduced by \$5338) with additional \$15 620 paid on HVAC. With “net-metering”, the NZEB obtained \$128/yr income (negative operation cost) due to the surplus electricity for all the options. With “instantaneous markup”, the GSHPs saved the operation cost by \$186 (2 boreholes) and \$191 (3 boreholes), with a simple payback of 42.8 and 51.2 years, respectively.

5 Conclusions

This study compared the energy, comfort, and economic performance of commercially-available HVAC technologies for a residential NZEB, based on a TRNSYS model validated by a year’s operation data. We investigated three kinds of HVAC options: (a) ventilation (no heat recovery, HRV, and ERV), (b) dehumidification (separate dehumidifier, ASHP only, and ASHP with dedicated dehumidification), and (c) heat pump (ASHP, GSHP with 2 boreholes, and GSHP with 3 boreholes). The main conclusions are:

1. For different ventilation options, the HRV and the ERV reduced the HVAC energy by 13.5 % and 17.4 % and reduced the building energy consumption by 7.5 % and 9.7 %. Under a “instantaneous markup” policy the HRV and ERV had simple paybacks of 15.4 and 9.0 years, respectively, compared to ventilation without heat recovery. The thermal comforts across the three options were equivalent.
2. For different dehumidification options, the ASHP with dedicated dehumidification reduced the HVAC energy by 7.3 %, reduced the building energy consumption by 3.9 %, and reduced the initial cost by \$3293 compared to the separate dehumidifier option. The ASHP-only option was the least expensive; however, the thermal comfort was the worst (1649 hours PPD>10%) and the humidity in the summer was the highest (2463 hours with relative humidity above 50 %).
3. For different heat pump options, the GSHP reduced the HVAC energy by 26.0 % for 2 boreholes and by 29.2 % for 3 boreholes, and reduced the building energy consumption by 13.1 % for 2 boreholes and 14.7 % for 3 boreholes, compared to the ASHP. The annual COP of the ASHP was 2.42, compared to 3.21 and 3.38 for the GSHPs. The comfort with the ASHP and the GSHPs was similar. However, for the same net electricity generation, the GSHPs required \$7934 (2 boreholes) and \$9739 (3 boreholes) higher initial cost than the ASHP, which resulted in long payback periods under the “instantaneous markup” policy.
4. The highest energy savings and good comfort was provided by the GSHPs with the ERV and dedicated dehumidification, but these options were the most expensive. A larger scale application of GSHP could decrease the initial cost and improve the economy in the future.
5. The option with the lowest initial cost that met the net zero energy target with good comfort was the ASHP with dedicated dehumidification and ventilation

without recovery. Under the “instantaneous markup” electric rates, upgrading the ventilation system to include an ERV (or HRV) provided reasonable payback periods.

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Abbreviations

ASHP	air source heat pump
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
CHP	combined heat and power
COP	coefficient of performance
DHW	domestic hot water
ERV	energy recovery ventilator
ESR	energy saving ratio
GSHP	ground source heat pump
HP	heat pump
HPWH	heat pump water heater
HRV	heat recovery ventilator
HVAC	heating, ventilation, and air conditioning
NIST	National Institute of Standards and Technology
NZEB	net-zero energy building
PMV	predicted mean vote
PPD	percentage of people dissatisfied
PV	photovoltaic
SHW	solar hot water
TRNSYS	Transient System Simulation Tool

Nomenclature

A	area, m ²
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C	cost, \$
E	energy consumption, kWh
f	ratio
h	convective heat transfer coefficient, $W/(m^2 \cdot ^\circ C)$
I	thermal resistance, $m^2 \cdot ^\circ C/W$
M	metabolic rate, W/m^2
m	mass flow rate, kg/s
P	power, W
p	pressure, Pa
Q	thermal load, kWh
t	temperature, $^\circ C$
V	velocity, m/s
W	work, W/m^2
η	effectiveness

References

1. U.S. Energy Information Administration. Monthly Energy Review. 2017
2. U.S. Energy Information Administration. Energy explained-use of electricity. 2017
3. Nejat P, Jomehzadeh F, Taheri MM, et al. A global review of energy consumption, CO2 emissions and policy in the residential sector (with an overview of the top ten CO2 emitting countries). *Renewable and Sustainable Energy Reviews*. 2015; 43:843–62.
4. Sartori I, Napolitano A, Voss K. Net zero energy buildings: A consistent definition framework. *Energy and Buildings*. 2012; 48:220–232.
5. US DOE. Building technologies program, planned program activities for 2008–2012. US: Department of Energy; 2008.
6. ASHRAE 2020 Ad Hoc Committee. ASHRAE vision 2020: providing tools by 2020 that enable the building community to produce market-viable NZEBs by 2030. 2008
7. Zhou Z, Feng L, Zhang S, et al. The operational performance of “net zero energy building”: A study in China. *Applied Energy*. 2016; 177:716–728.
8. Marszal AJ, Heiselberg P, Bourrelle JS, et al. Zero Energy Building—A review of definitions and calculation methodologies. *Energy and Buildings*. 2011; 43(4):971–979.
9. Kampelis N, Gobakis K, Vagias V, et al. Evaluation of the performance gap in industrial, residential & tertiary near-Zero energy buildings. *Energy and Buildings*. 2017; 148:58–73.
10. Fannee AH, Payne V, Ullah T, et al. Net-zero and beyond! Design and performance of NIST’s net-zero energy residential test facility. *Energy and Buildings*. 2015; 101:95–109.
11. Moran P, Goggins J, Hajdukiewicz M. Super-insulate or use renewable technology? Life cycle cost, energy and global warming potential analysis of nearly zero energy buildings (NZEB) in a temperate oceanic climate. *Energy and Buildings*. 2017; 139:590–607.
12. Thygesen R, Karlsson B. An analysis on how proposed requirements for near zero energy buildings manages PV electricity in combination with two different types of heat pumps and its policy implications—A Swedish example. *Energy Policy*. 2017; 101:10–19.

13. Lu Y, Wang S, Shan K. Design optimization and optimal control of grid-connected and standalone nearly/net zero energy buildings. *Applied Energy*. 2015; 155:463–477.
14. Ng LC, Payne WV. Energy use consequences of ventilating a net-zero energy house. *Applied Thermal Engineering*. 2016; 96:151–160. [PubMed: 26903776]
15. Li H, Xu W, Yu Z, Wu J, Sun Z. Application analyze of a ground source heat pump system in a nearly zero energy building in China. *Energy*. 2017; 125:140–151.
16. Fabrizio E, Seguro F, Filippi M. Integrated HVAC and DHW production systems for Zero Energy Buildings. *Renewable and Sustainable Energy Reviews*. 2014; 40:515–541.
17. Lu Y, Wang S, Shan K. Design optimization and optimal control of grid-connected and standalone nearly/net zero energy buildings. *Applied Energy*. 2015; 155:463–477.
18. Kim J, Hong T, Jeong J, et al. Establishment of an optimal occupant behavior considering the energy consumption and indoor environmental quality by region. *Applied Energy*. 2017; 204:1431–1443.
19. Elkinton MR, McGowan JG, Manwell JF. Wind power systems for zero net energy housing in the United States. *Renewable Energy*. 2009; 34(5):1270–1278.
20. Marszal AJ, Heiselberg P, Jensen RL, et al. On-site or off-site renewable energy supply options? Life cycle cost analysis of a net zero energy building in Denmark. *Renewable Energy*. 2012; 44:154–165.
21. Musall, E., Weiss, T., Lenoir, A., et al. Net Zero energy solar buildings: an overview and analysis on worldwide building projects. EuroSun Conference; Graz, Austria. 2010.
22. Good C, Andresen I, Hestnes AG. Solar energy for net zero energy buildings—A comparison between solar thermal, PV and photovoltaic–thermal (PV/T) systems. *Solar Energy*. 2015; 122:986–996.
23. Payne, WV. Annual Performance of a Two-Speed, Dedicated Dehumidification Heat Pump in the NIST Net-Zero Energy Residential Test Facility. 2016; ASHRAE Winter Conference; Orlando. 2016.
24. Mohamed A, Hasan A, Sirén K. Fulfillment of net-zero energy building (NZEB) with four metrics in a single family house with different heating alternatives. *Applied Energy*. 2014; 114:385–399.
25. Arroyo P, Tommelein ID, Ballard, et al. Choosing by advantages: A case study for selecting an HVAC system for a net zero energy museum. *Energy and Buildings*. 2016; 111:26–36.
26. AlAjmi A, Abou-Ziyan H, Ghoneim A. Achieving annual and monthly net-zero energy of existing building in hot climate. *Applied Energy*. 2016; 165:511–521.
27. ASHRAE Standard 90.1-2014. Energy Standard for Buildings Except Low-Rise Residential Buildings. American Society of Heating, Refrigerating and Air-conditioning Engineers; 2013.
28. Fanney AH, Payne V, Ullah T, et al. Net-zero and beyond! Design and performance of NIST's net-zero energy residential test facility. *Energy and Buildings*. 2015; 101:95–109.
29. Omar, F., Omar, F., Bushby, ST. Simulating Occupancy in the NIST Net-zero Energy Residential Test Facility. US Department of Commerce, National Institute of Standards and Technology; 2013.
30. Hendron, R., Engebrecht, C. Building America house simulation protocols. National Renewable Energy Laboratory, Building Technologies Program, U.S. Department of Energy; 2010.
31. [Accessed 6/9/2017] NIST Net-Zero Energy Residential Test Facility (NZERTF) website. <https://www.nist.gov/el/net-zero-energy-residential-test-facility>
32. Davis, M., Healy, W., Boyd, M., et al. Monitoring Techniques for the Net-Zero Energy Residential Test Facility. National Institute of Standards and Technology; 2014.
33. AHRI. Certificate of Product Ratings. Air Conditioning, Heating, and Refrigeration Institute; 2014.
34. ASHRAE Standard 62.2-2010. Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings. American Society of Heating, Refrigerating and Air-conditioning Engineers; 2010.
35. Klein, A., Beckman, A., Mitchell, W., et al. TRNSYS 17—a transient system simulation program. Madison: Solar Energy Laboratory, University of Wisconsin; 2011.
36. Balke, E. Master's Thesis. University of Wisconsin; Madison: 2016. Modeling, Validation, and Evaluation of the NIST Net Zero Energy Residential Test Facility.

37. Leyde, B. Master's Thesis. University of Wisconsin; Madison: 2014. TRNSYS Modeling of the NIST Net Zero Energy Residential Test Facility.
38. ISO Standard 13256-2:1998. Water-source heat pumps—Testing and rating for performance. International Organization for Standardization; 1998.
39. Hellström, G. Duct ground heat storage model, manual for computer code. Department of Mathematical Physics, University of Lund; Sweden: 1989.
40. Leyde BP, Klein SA, Nellis GF, et al. Determination of vertical borehole and geological formation properties using the Crossed Contour Method. *Geothermics*. 2017; 66:174–182. [PubMed: 28785125]
41. ISO Standard 7730:2005. Ergonomics of the Thermal Environment-Analytical Determination and Interpretation of Thermal Comfort Using Calculation of the PMV and PPD Indices and Local Thermal Comfort Criteria. International Organization for Standardization; 2005.
42. ANSI/ASHRAE Standard 55-2010. Thermal environmental conditions for human occupancy. American Society of Heating, Refrigerating and Air-conditioning Engineers; 2010.
43. Fanger, PO. Thermal comfort: analysis and applications in environmental engineering. Copenhagen: Danish Technical Press; 1970.
44. Galen, B., Naïm, D., Dev, M., et al. Tracking the Sun IX: The Installed Price of Residential and Non-Residential Photovoltaic Systems in the United States. US Department of Energy, Lawrence Berkeley National Laboratory; 2016.
45. Kneifel, J., O'Rear, E. Net-Zero Energy Residential Building Component Cost Estimates and Comparisons. US Department of Commerce, National Institute of Standards and Technology; 2016.
46. Battocletti, EC., Glassley, WE. Measuring the Costs and Benefits of Nationwide Geothermal Heat Pump Deployment. Bob Lawrence & Associates, Inc; 2013. DE-EE0002741. Available online at <http://www.osti.gov/geothermal/servlets/purl/1186828>. Database available online at <https://gdr.openei.org/submissions/180>
47. Kavanaugh S, Green M, Mescher K. Long term commercial GSHP performance: part 4: installation costs. *ASHRAE Journal*. 2012; 54(10):26–34.
48. Kneifel, J. Life-Cycle Cost Comparison of the NIST Net Zero Energy Residential Test Facility to a Maryland Code-Compliant Design. US Department of Commerce, National Institute of Standards and Technology; 2014.
49. Bean, R. Thermal Comfort and Energy. 2016 Humid Climate Conference; 2016.
50. You T, Wu W, Shi W, et al. An overview of the problems and solutions of soil thermal imbalance of ground-coupled heat pumps in cold regions. *Applied Energy*. 2016; 177:515–36.

Fig. 1.
Residential NZEB on NIST campus

Fig. 2.
Schematic diagram of the electricity flow

Fig. 3.
Schematic diagram of the HVAC system

Fig. 4.
Schematic diagram of the DHW system

Fig. 5.
Validation of building model

Fig. 6.
Validation of PV and ASHP model

Fig. 7.
Validation of DHW model

Fig. 8.
Installed price of residential PV systems in the U.S. [44]

Fig. 9.
Sensible and latent loads introduced by ventilation

Fig. 10.
Thermal loads and energy consumption of ASHP

Fig. 11.
Hourly thermal comfort of different ventilation options

Fig. 12.
Energy removal and energy consumption in dehumidification mode

Fig. 13.
Thermal loads and energy consumption of ASHP

Fig. 14.
Hourly thermal comfort of different dehumidification options

Fig. 15.
Thermal loads and energy consumption of heat pumps

Fig. 16.
Heat source and sink temperatures and heat pump COP

Fig. 17.
Heat balance and soil temperature of GSHP

Fig. 18.
Hourly thermal comfort of different heat pump options

Table 1

Components and main parameters of different subsystems of the TRNSYS model

Subsystem	Component	TRNSYS type	Main parameters
Building	Building	Type 56	living area: 251 m ² ; basement area: 135 m ² setpoint: 20.5 °C heating, 23.9 °C cooling, 48 % dehumidification
	Infiltration	Type 932	effective leakage area: 244 cm ²
PV	PV array	Type 194b	module area: 1.472×32 m ² ; orientation: south tilt angle: 18.4°; maximum power: 10.24 kW inverter efficiency: 95.5 %; frequency: 60 Hz
HVAC	1 st and 2 nd stage ASHP	Type 922	1 st stage rated air flow rate: 1060 m ³ /h 1 st stage rated capacity: 5425 W cooling, 5180 W heating 1 st stage rated power: 1440 W cooling, 1398 W heating 2 nd stage rated air flow rate: 1420 m ³ /h 2 nd stage rated capacity: 7067 W cooling, 8041 W heating 2 nd stage rated power: 2420 W cooling, 2126 W heating
	3 rd stage ASHP	Type 954	3 rd stage rated air flow rate: 1420 m ³ /h 3 rd stage rated heating capacity: 8041 W 3 rd stage rated heating power: 2126 W supplemental electric heating: 10 000 W
	Dehumidifying ASHP	Type 922	1 st stage rated air flow rate: 1025 m ³ /h 1 st stage rated cooling capacity: 5997 W 1 st stage rated cooling power: 1820 W 2 nd stage rated air flow rate: 933 m ³ /h 2 nd stage rated cooling capacity: 1653 W 2 nd stage rated cooling power: 1230 W
	HRV	Type 667b	outdoor air: 195 m ³ /h; defrost temperature trigger: -10 °C sensible effectiveness: 0.72; latent effectiveness: 0.01
	HRV fans	Type 111b	(each of the two fans) rated flow rate: 195 m ³ /h rated power: 27 W; motor efficiency: 0.9
DHW	Solar collector	Type 1b	collector area: 2.1 m ² ; intercept efficiency: 0.744 1 st order efficiency coefficient: 3.6707 W/(m ² ·K) 2 nd order efficiency coefficient: 0.00543 W/(m ² ·K ²)
	Solar storage tank	Type 534	volume: 0.189 m ³ ; height: 1.143 m loss coefficient: 0.59 W/(m ² ·K)
	Heat exchanger	Type 91	heat exchanger effectiveness: 0.44
	Pumps	Type 114	rated flow rate: 196 kg/h brine, 999 kg/h water rated power: 80 W glycol, 80 W water
	HPWH	Type 938	total air flow rate: 765 m ³ /h; blower power: 5 W rated heating capacity: 2025 W rated heating power: 794 W
	Electric heater	Type 1226	heating capacity: 3600 W; thermal efficiency: 1.0
	HPWH tank	Type 534	volume: 0.2953 m ³ ; height: 1.5939 m loss coefficient: 1.0 W/(m ² ·K)
	Pipes	Type 31	detailed diameters, lengths, and loss coefficients in [36]

Table 2

Parameters of different ventilation systems

Parameter	No heat recovery	HRV	ERV
Rated sensible effectiveness	0.00	$\eta_0 = 0.72$	$\eta_0 = 0.68$
Sensible effectiveness coefficients	Not Applicable	$c_0 = 1.273$	$c_0 = 1.103$
		$c_1 = -0.3246$	$c_1 = 0.0162$
		$c_2 = 0.0521$	$c_2 = -0.119$
Latent effectiveness	0.00	0.01	0.47
Flow rate (m ³ /h)	195	195	195
Rated power (W)	$P_0 = 27.0$	$P_0 = 54.0$	$P_0 = 60.0$
Power coefficients	$c_0 = 0.5656$	$c_0 = 0.5656$	$c_0 = 0.3098$
	$c_1 = -0.6726$	$c_1 = -0.6726$	$c_1 = -0.1082$
	$c_2 = 1.1477$	$c_2 = 1.1477$	$c_2 = 0.7972$

Table 3

Components and parameters of TRNSYS models for GSHP systems

Subsystem	Component	TRNSYS type	Main parameters
GSHP	1 st stage	Type 919	rated flow rate: 1020 m ³ /h air, 1.44 m ³ /h liquid rated capacity: 5273 W cooling, 3943 W heating rated power: 922 W cooling, 1048 W heating
	2 nd stage	Type 919	rated flow rate: 1360 m ³ /h air, liquid 1.80 m ³ /h rated capacity: 6686 W cooling, 4962 W heating rated power: 1520 W cooling, 1437 W heating
	3 rd stage	Type 919	rated flow rate: 1360 m ³ /h air, 1.80 m ³ /h liquid rated heating capacity: 4962 W rated heating power: 1437 W supplemental electric heating: 4800 W
Dehumidification	Dehumidifier	Type 921	rated air flow rate: 1025 m ³ /h rated cooling capacity: 1536 W rated cooling power: 2727 W
	Air heater	Type 121b	heating capacity is the heat rejection of dehumidifier
Ground loop	Borehole	Type 557	effective borehole radius: 4.88 cm; borehole depth: 45 m borehole spacing: 6.4 m; tube center to center: 6.83 cm tube diameter: 2.54 cm inner, 3.30 cm outer tube conductivity: 1.62 W/(m·K) soil conductivity: 2.43 W/(m·K) soil heat capacity: 2549 kJ/(m ³ ·K) fill conductivity: 2.617 W/(m·K) fluid density: 980.7 kg/m ³ fluid specific heat: 4.396 kJ/(kg·K)

Table 4

Cost of main equipment used in various HVAC options

Equipment	Cost
Ventilation without heat recovery	\$876
HRV	\$4612
ERV	\$5203
ASHP only	\$26 752
ASHP with dedicated dehumidification	\$28 163
Separate dehumidifier	\$3265 *
GSHP only	\$34 656
GSHP with dedicated dehumidification	\$36 067
Ground loop	\$2391 per borehole

* The separate dehumidifier was increased to 2.5 times the capacity of the original one to obtain thermal comfort similar to the ASHP dedicated dehumidification. The original cost for the separate dehumidifier was \$1306 [45], using a linear assumption, the cost of the separate dehumidifier was estimated to be: $\$1306 \times 2.5 = \3265 .

Table 5

Annual electricity consumption and generation under different ventilation options

Items	No heat recovery	HRV	ERV
DHW (kWh) *	1245	1247	1246
ASHP (kWh)	7381	6143	5803
Ventilation (kWh)	226	440	481
Lighting + plug + appliance (kWh)	4802	4802	4802
Total consumption (kWh)	13 654	12 632	12 332
PV generation (kWh)	14 154	14 154	14 154
Net generation (kWh)	500	1522	1822

* Slight differences in DHW energy between ventilation options are due to minor differences in indoor temperature

Table 6

Annual PMV and PPD of different ventilation options

Items	No heat recovery	HRV	ERV
Annual average PMV	0.11	0.14	0.14
Summer average PMV	0.34	0.38	0.37
Annual average PPD (%)	7.0	7.1	7.1
Summer average PPD (%)	8.3	8.5	8.5
Annual hours PPD>10%	985	1067	1053
Summer hours PPD>10%	792	809	795

Table 7

Economic analysis of different ventilation options

Cost	No heat recovery	HRV	ERV
HVAC (\$)	29 039	32 775	33 366
PV (before reduction) (\$)	39 780	39 780	39 780
PV (after reduction) (\$)	39 780	36 908	36 063
HVAC+PV (before reduction) (\$)	68 819	72 555	73 146
HVAC+PV (after reduction) (\$)	68 819	69 683	69 429
Operation (net-metering) (\$/yr)	−43	−43	−43
Operation (instantaneous markup) (\$/yr)	576	519	508
Simple payback (yr)	Baseline	15.4	9.0

Table 8

Annual electricity consumption and generation under different dehumidification options

Items	Separate	ASHP only	Dedicated
DHW (kWh) *	1243	1246	1247
ASHP (kWh)	6659	5097	6143
Ventilation (kWh)	440	440	440
Lighting + plug + appliance (kWh)	4802	4802	4802
Total consumption (kWh)	13 144	11 585	12 632
PV generation (kWh)	14 154	14 154	14 154
Net generation (kWh)	1010	2569	1522

* Slight differences in DHW energy between dehumidification options are due to minor differences in indoor temperature

Table 9

Annual PMV and PPD of different dehumidification options

Items	Separate	ASHP only	Dedicated
Annual average PMV	0.17	0.17	0.14
Summer average PMV	0.41	0.44	0.38
Annual average PPD (%)	7.5	7.6	7.1
Summer average PPD (%)	8.9	9.5	8.5
Annual hours PPD>10%	1275	1932	1067
Summer hours PPD>10%	855	1649	809

Table 10

Economic analysis of different dehumidification options

Cost	Separate	ASHP only	Dedicated
HVAC (\$)	34 629	31 364	32 775
PV (before reduction) (\$)	39 780	39 780	39 780
PV (after reduction) (\$)	39 780	35 398	38 341
HVAC+PV (before reduction) (\$)	74 409	71 144	72 555
HVAC+PV (after reduction) (\$)	74 409	66 762	71 116
Operation (net-metering) (\$/yr)	−87	−87	−87
Operation (instantaneous markup) (\$/yr)	492	418	473
Simple payback (yr)	Baseline	0 *	0

* A payback of 0 means both lower initial cost and lower operation cost

Table 11

Electricity consumption breakdown for different heat pump options

Items	ASHP	GSHP (2 boreholes)	GSHP (3 boreholes)
Heating (kWh)	2818	2195	2095
Defrost energy (kWh)	360	0	0
Heating standby (kWh)	138	125	130
Cooling (kWh)	1230	816	720
Dehumidification (kWh)	1477	1170	1147
Cooling standby (kWh)	120	116	117
Total heating (kWh)	3316	2320	2225
Total cooling (kWh)	2827	2102	1984
Total heat pump (kWh)	6143	4422	4209
Annual COP	2.42	3.21	3.38

Table 12

Annual electricity consumption and generation under different heat pump options

Items	ASHP	GSHP (2 boreholes)	GSHP (3 boreholes)
DHW (kWh) *	1247	1309	1310
ASHP (kWh)	6143	4422	4209
Ventilation (kWh)	440	450	450
Lighting + plug + appliance (kWh)	4802	4802	4802
Total consumption (kWh)	12 632	10 983	10 770
PV generation (kWh)	14 154	14 154	14 154
Net generation (kWh)	1522	3171	3384

* Slight differences in DHW energy between heat pump options are due to minor differences in indoor temperature

Table 13

Annual PMV and PPD of different heat pump options

Items	ASHP	GSHP (2 boreholes)	GSHP (3 boreholes)
Annual average PMV	0.14	0.12	0.13
Summer average PMV	0.38	0.34	0.34
Annual average PPD (%)	7.1	7.0	6.9
Summer average PPD (%)	8.5	8.1	8.1
Annual hours PPD>10%	1067	1008	999
Summer hours PPD>10%	809	772	762

Table 14

Economic analysis of different heat pump options

Cost	ASHP	GSHP (2 boreholes)	GSHP (3 boreholes)
HVAC (\$)	32 775	45 461	47 852
PV (before reduction) (\$)	39 780	39 780	39 780
PV (after reduction) (\$)	39 780	35 028	34 442
HVAC+PV (before reduction) (\$)	72 555	85 241	87 632
HVAC+PV (after reduction) (\$)	72 555	80 489	82 294
Operation (net-metering) (\$/yr)	−128	−128	−128
Operation (instantaneous markup) (\$/yr)	426	240	235
Simple payback (yr)	Baseline	42.8	51.2