# Improving Ground-Source Heat Pump Efficiency: Optimizing Pumping Control and Configuration of Closed-Loop Heat Pump Systems in Smaller Commercial Buildings

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## ABSTRACT

Ground-source heat pumps (GSHP) have proven to be an efficient alternative to standard Heating, Ventilating, and Air-Conditioning (HVAC) systems. Yet GSHP commercial market penetration has been slow in the Pacific Northwest. Eugene Water & Electric Board (EWEB), a moderate sized public utility, chose a GSHP system for a major renovation of a multi-use, 18,000 square foot historical building. The paper reviews lessons learned regarding ground loop size and pumping flow. Analysis focuses on the energy impact of various pumping configurations and control strategies.

The 14 commercial water-source heat pumps (HP) and 2 circulation pumps were monitored during peak heating and cooling seasons in this pilot project. The data allowed analysis of part-load operation in a smaller building. The part-load data was used to investigate improvements to pumping configuration and pumping control optimization. Four pumping configurations were analyzed with various control strategies to determine the optimum HVAC energy use while maintaining the simplicity necessary for small commercial buildings. The analysis shows that the pumping energy share can be reduced from 39% to 10% of total HVAC energy use. The authors recommend a decoupled system with individual HP water pumps for the building loop and simple temperature controls for the ground-loop pump. Six of the fifteen options analyzed were found to be cost-effective.

## Background

In 1998 EWEB renovated an historic building on their property. During design, staff found that mechanical and electrical systems needed replacement. This provided an opportunity to install a ground-source heat pump (GSHP) pilot. The installation included monitoring of ground-loop temperatures and heat pump (HP) status.

A literature review found that research to date focused on sizing the ground-loop heat exchanger and minimizing loop pressure drop for pumping selection (Cane and Clemes 1995; Kavanaugh 1995; Rafferty 1995). Several studies reported high pumping energy use and provided simplified analysis of variable or controlled pumping systems (Phetteplace 1998; Woller 1994). Energy impacts of specific pumping configurations combined with various control strategies have not been reported. For small- to mid-sized commercial buildings, general recommendations on various control strategies are expected to prove useful.

This is the first commercial GSHP project installed in Eugene. The closed-loop ground-coupled HP system has proven to be an efficient system, appropriate for small to medium commercial buildings. As the project started, there was no local design or contractor experience with a GSHP system, and no local closed-loop bore drillers. With this lack of local experience, we expected this pilot project to be a source of "lessons learned."

### Scope

The purpose of this study is to determine optimum pumping configuration and pumping control for ground-source heat pump systems. Monitored data is used to develop a model of ground-coupled heat exchanger performance and predict the energy impact of various flows, pumping configurations, and control strategies. Of many possible multiple-pump configurations, the focus here is on simple systems that are appropriate for the small commercial building.

Prior work (Price 1999) verified water-source heat pump performance and provided life cycle cost justification for ground-source heat pumps versus other system types in the monitored building. Part-load data for an actual building provides a good basis for analysis of various pumping configuration and control options. During the study period, internal loads in the monitored building were low and the overall Heating, Ventilating, and Air-Conditioning (HVAC) energy use index (EUI) may be lower than some retail or office occupancies. While study results may not transfer to all similar sized buildings, the focus here is making relative comparisons between the options available to reduce pumping energy in GSHP systems. Those relative rankings should apply to other similar sized buildings.

## Monitored Building and System Description

The building covers 18,000 square feet with multiple occupancies. The renovated historical building houses a multi-utility storefront for residential customer energy awareness, a classroom, a fitness center, and a day care center. A large portion of the upper floor is available as build-out tenant space that was unoccupied during the study period.

The newly installed HVAC system has 14 extended-range water-source heat pumps, ranging in size from 3 to 5 tons. A vertical-bore geothermal field located under the parking lot has 20, 300-foot deep bores and serves as the heat source and sink. During the study, 4 of the 14 heat pumps served unoccupied areas. The usage profiles for the 10 active heat pumps were applied to the unoccupied areas. Data was collected from June 1999 through February 2000. Actual data was used for peak cooling and heating seasons. Fall data was substituted for similar degree day months in spring to develop an annual load profile.

### HVAC Operation and Heat Pump Sizing

Analysis of the data shows that the heat pump compressors seldom operate. Figure 1 shows both total coil energy and hours of operation at various heating and cooling block loads.[1] The installed capacity is 42 tons; however, the peak heating hour uses only 60% capacity and peak cooling only 27%. This building contains a classroom and large museum area, both with low average loads. Apparent HP unit over-sizing is typical and expected for

[1] The block load is the total building cooling or heating load measured at the heat pump coils in a given hour. The simultaneous block loads are usually much less than the sum of individual heat pump installed capacities, because not all heat pumps operate at the same time under actual diversified load conditions.

these systems. Design safety factors are added and allowances are made for future occupancies with higher internal heat gains. In construction, the next larger size heat ~~pump is~~ often substituted, adding to the installed capacity. Pumping design is affected by the installed capacity being much greater than the average required heating or cooling load.

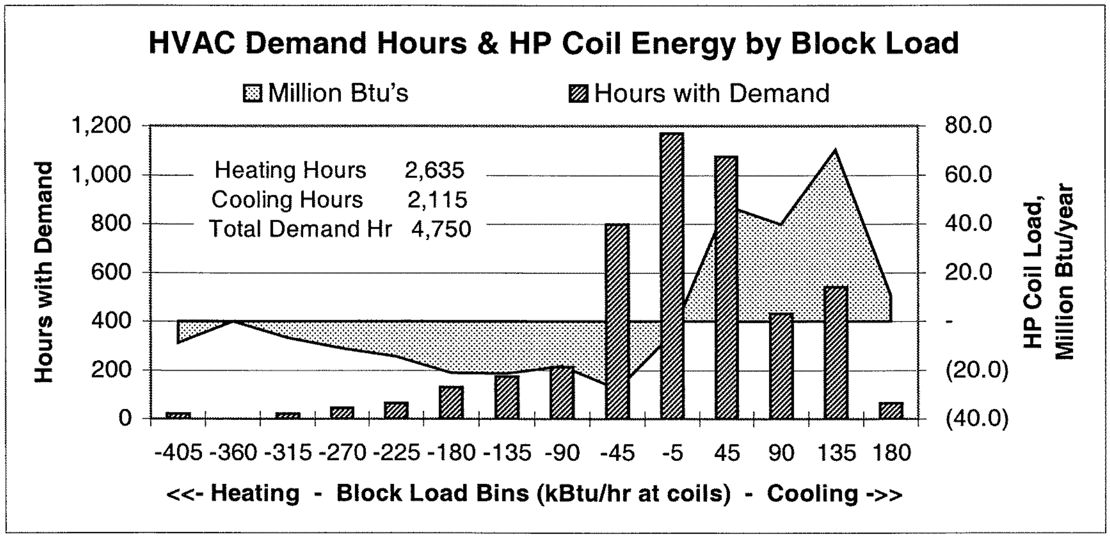


**Figure 1. HVAC System Hours of Operation at Various Loads**

## Adjustments to Bore Field Size and Loop Flow

While optimal field sizing and loop flow determination are beyond the scope of this paper, reasonable adjustments were made to the data from the installed system before analyzing control options. For a ground-loop system, apparent HP unit over-sizing can result in circulating pumps using a significant portion of the total HVAC energy if pumps are not designed for block loads. Higher design flows can also occur when trying to maintain adequate velocity through an oversized geothermal bore field.

As this was a new technology for the area, utility staff were trained to size the ground exchanger. Based on block loads from a DOE2 analysis and trials with two ground-loop sizing programs, 16, 300-foot bores were called for. The experienced geothermal driller advocated for 24 bores. In compromise, 20 bores were contracted for. With the benefit of actual operating data, a regression analysis of ground exchanger supply and return temperatures (shown in Figure 2) verified that 16 bores would have been more appropriate. Not only would the original 16-bore design save about $12,300 in field costs, but a 16 bore ground exchanger better matched system fluid flow. Similar over-sizing appears in the literature (Cane and Clemes 1995). A lesson learned is that trusting available analysis tools and avoiding over-sized bore fields is important in managing total system cost.

Furthermore, the installed pumping design was conservative; selecting individual heat pump unit flows at the maximum of 3.4 gpm per installed ton. Flow is less important to heat pump efficiency than entering water temperature (EWT), as can be seen in Figure 3. Selecting fluid flow below 2.0 gpm per installed ton significantly reduces pumping cost and saves energy overall. The analyzed baseline system uses 101 gpm or 2.4 gpm per installed ton and 3.0 gpm per block load ton. While even lower flows are possible, this seemed appropriate and matches current design recommendations (Kavanaugh and Rafferty 1997). The flows of installed, designed, and analyzed systems are compared in Table 1, with option numbers for later reference.

**Table 1. Loop Flows in GPM per Ton for Various Conditions**

| Description, Option Reference | Bores | GPM | Installed Capacity | Peak Heat Block Load | Managed Heat Block | Cooling Block Load |
|---|---|---|---|---|---|---|
| *Total Tons* | | | *42* | *34* | *23* | *15* |
| As Monitored (unbalanced): 1U, 1X | 20 | 185 | 4.4 | 5.4 | 8.2 | 12.3 |
| As Designed | 20 | 144 | 3.4 | 4.2 | 6.4 | 9.6 |
| As Analyzed: 1A through 4B | 16 | 101 | 2.4 | 3.0 | 4.5 | 6.7 |
| With Dual-pumping: 2E, 3E | 16 | 85 | 2.0 | 2.5 | 3.8 | 5.7 |

To determine baseline energy use, the SEER and COP shown in Figure 3 were used to adjust heat pump energy use to reflect reduced loop flow, a smaller bore field, and the wider range of loop temperatures that will occur when tenant spaces become occupied. The lesson (re)learned here (Kavanaugh and Rafferty 1997) is to base flow on block loads and not installed capacity. Energy savings of more than 40% are possible by reducing flow, while control and configuration improvements have a potential for about 30% additional savings.

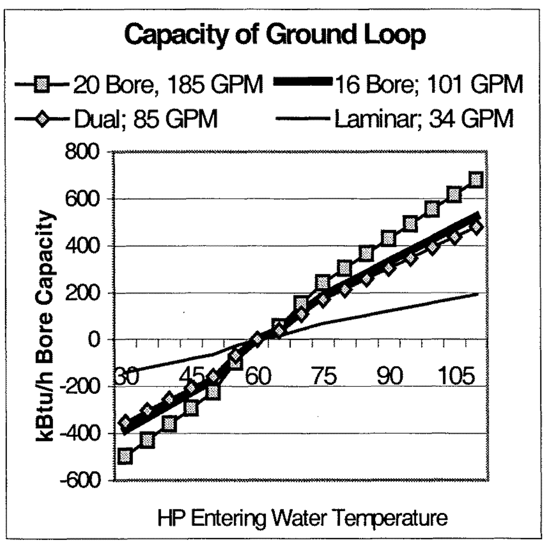


**Figure 2. Ground Exchanger Capacity**

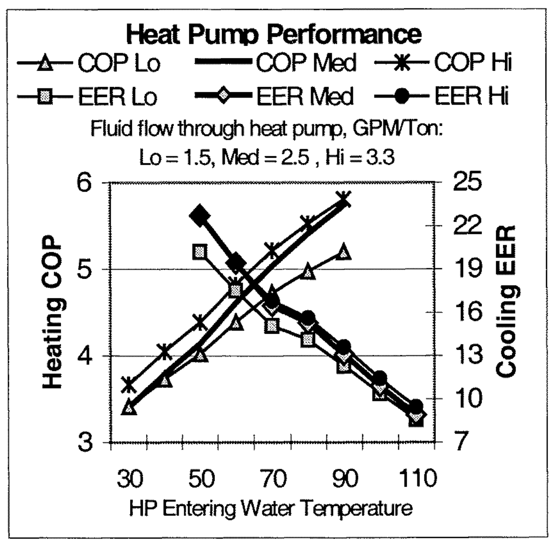


**Figure 3. GSHP Efficiency**

## More Efficient Pumping

An analysis of the installed condition (option 1U with high pumping flow) shows that with a DDC system operating the loop pump, 47% of the HVAC electric use was for pumping. With standard programmable thermostats, the loop pump typically operates continuously (option 1X), increasing pumping energy use to a 65% share. An evaluation of the system design using the "grading" proposed in the ASHRAE GSHP design manual

(Kavanaugh and Rafferty 1997) shows that by reducing to a flow of 101 gpm (the analyzed condition in Table 1), the system grade improves from "D" to "A."[2] With continuous lower flow, the baseline system (option 1A) results in 39% of HVAC energy in pumping. A grade of peak pumping power is a good place to start; however, hours of operation can be more important in evaluating pumping costs. In fact, option 4B received an "F" grade, yet resulted in about one-third the pumping energy use of the "A" graded baseline system (option 1A).

**Alternate Pumping Configurations**

To evaluate pumping efficiency, four alternate pumping configurations are combined with several different control strategies (that will be discussed separately). In all four pumping configurations, individual heat pumps are piped in parallel to a building supply and return loop. The closed-loop ground heat exchangers are piped in parallel on header piping. The differences occur in pump location and in how the two loops are connected. The four configurations are discussed below.

**Pumping Configuration 1: Single Loop.** In configuration 1 (see Figure 4) the building loop and ground loop are connected in series with a single active pump. Two pumps are typically installed to provide redundancy, since a pump failure will halt the entire building HVAC system. The pump must operate when any heat pump calls for heating or cooling.

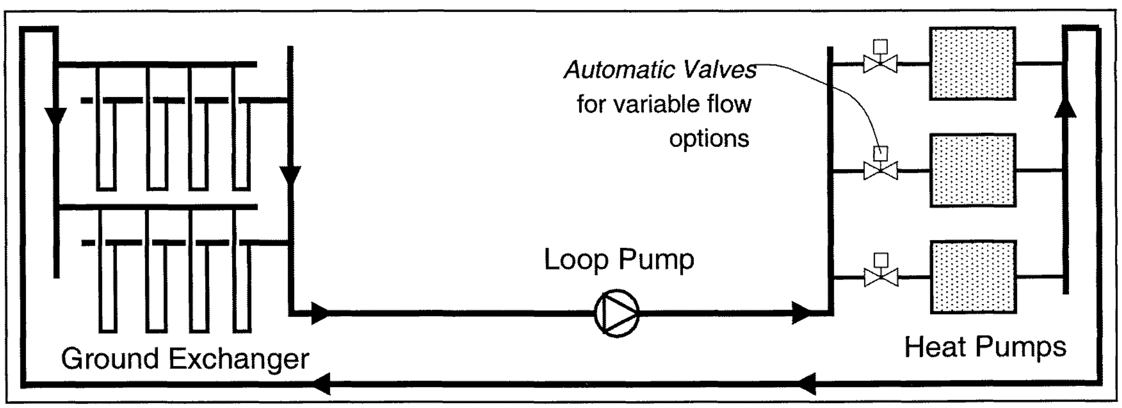


**Figure 4. Pumping Configuration 1: Single Loop**

**Pumping Configuration 2: Decoupled Pumping Loops.** Figure 5 shows configuration 2 where the building loop and ground loop have separate pumps with a bypass bridge that hydraulically decouples the two loops. Configuration 2 allows different flows in the ground loop and the building loop, although a consistent 101 gpm was used for analysis.[3] The building loop pump must operate when any heat pump calls for heating or cooling. The ground exchange loop pump must operate only when the load on the GSHPs drives the loop

[2] The grade reflects peak pumping power per cooling block ton and is influenced mostly by system flow, pipe sizing, heat pump selection, pump efficiency, and pump motor efficiency. Hours of pump operation or variable flow are not accounted for. Grades are based on the designed 34 block tons of cooling, even though monitored operation shows only 15 block tons of cooling for the building.

[3] Different flows were investigated during the analysis. The higher viscosity of antifreeze fluid can result in pump power increases at lower flows, compounded by lower flows requiring more antifreeze to meet peak loads.

temperature out of the desired range. Two pairs of central pumps with check valves are typically installed to provide redundancy.

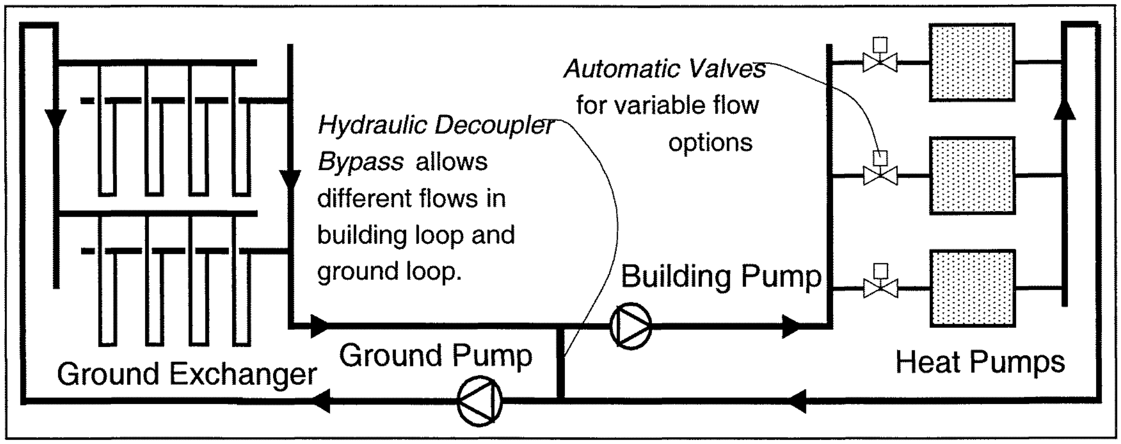


**Figure 5. Pumping Configuration 2: Decoupled Pumping**

**Pumping Configuration 3: Ground Pump with HP Pumps.** Figure 6 shows configuration 3 where the building loop and ground loop are also decoupled. A single pump (or redundant pair) serves the ground exchange loop. Each heat pump has a separate water pump and check valve for the building loop.[4] Each HP water pump operates only when that individual heat pump calls for heating or cooling. The ground exchange loop pump operates only when the building loop temperature is outside the desired range. The pumps are selected and balanced to maintain the minimum required heat pump flow when all heat pumps are operating.

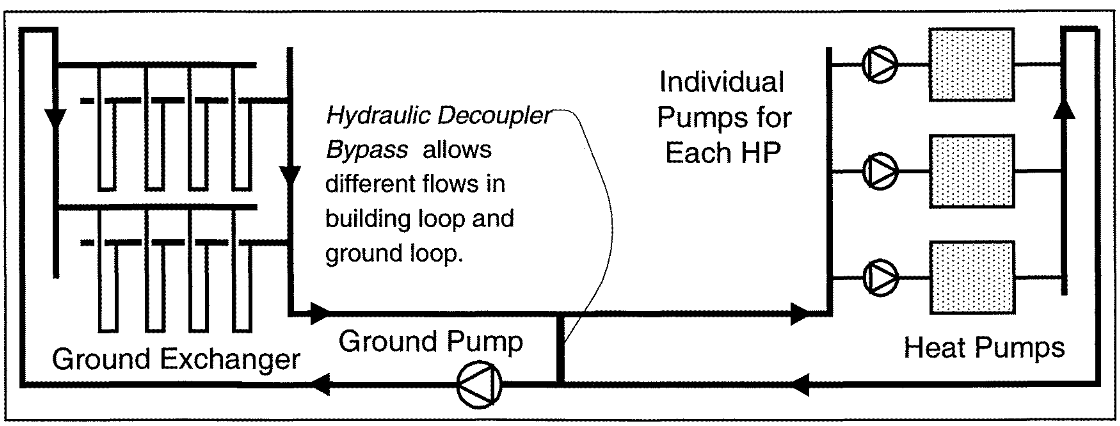


**Figure 6. Pumping Configuration 3: Ground Pump with HP Pumps**

**Pumping Configuration 4: Individual HP Pumps.** Note that in Figure 7 the configuration 4 loop piping is in series, similar to configuration 1. Each heat pump has an individual water pump like configuration 3. The water pumps operate only when the

[4] While the small pumps and motors are not very efficient, they only operate when needed. The operating time savings results in a lower overall energy cost than the other configurations. Individual water pumps are better matched to individual HP pressure drops than one building pump that must handle the largest pressure drop.

associated heat pump calls for heating or cooling. Pumps are selected and balanced to maintain the minimum required HP unit flow when all pumps are operating.[5] In option 4A, constant flow valves maintain the minimum flow at all times.[6] In option 4B, a higher flow is allowed when fewer pumps are operating and total system pressure drop is reduced.

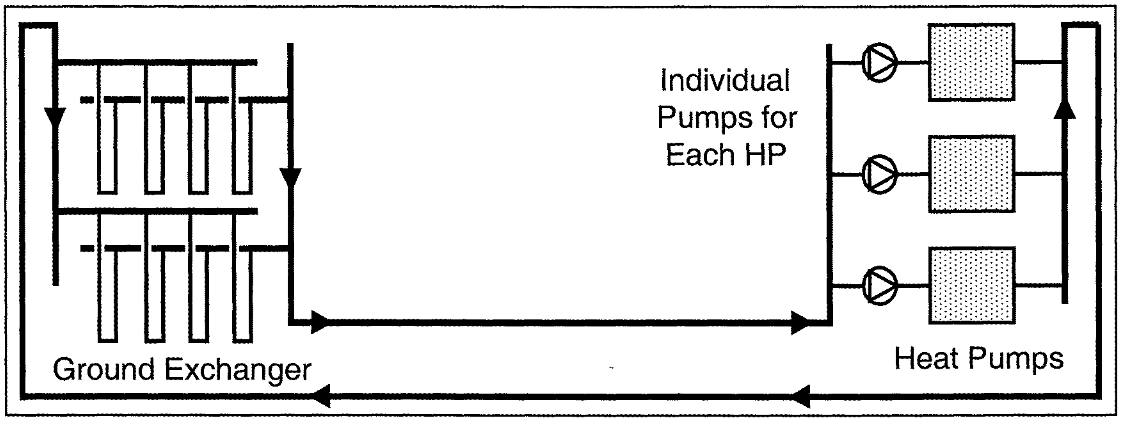


**Figure 7. Pumping Configuration 4: Individual HP Pumps**

## Analyzed Control Options

Several control strategies can be applied to either the building loop pump or the ground exchanger loop pump in each of the four pumping configurations. The range of options analyzed is listed in Table 2. Following the KISS theme (Cooper 1994), each option is assigned a subjective KISS index, from "1" for simple to "4" for complex. Each option is then compared based on energy use and cost. Each control strategy is discussed below.

**HP Control.** A DDC control system was used to provide monitoring in the pilot building. For the analysis, more simplified networked thermostats were used for "on request" control. Where pumps run continuously or have other controls, standard programmable thermostats are used for heat pump control.

**Continuous.** This pump is on a manual switch and operates 8,760 hours per year.

**Request.** The pump operates whenever there is any heat pump requiring heating or cooling. This can be accomplished either with a central DDC system, a set of electric relays, or a communicating programmable thermostat system that is networked to report heating or cooling calls.

**Loop Temp.** The pump is on whenever the loop temperature (building loop return after the heat pumps) goes outside the desired range. For analysis, the pump operates when the loop temperature was warmer than 80°F or cooler than 40°F. Between 50°F and 70°F, the ground

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[5] Pump selection can be more difficult than other configurations, with low-flow/high-head conditions often requiring series pumping or high rotational pump speeds.

[6] At low flows, the ground exchanger sees a higher temperature difference, resulting in more extreme heat pump operating temperatures and higher heat pump energy use. For option 4A, the high temperature difference at laminar flows requires an increase to 20% methanol.

pump is off and the building pump(s) continues to circulate. Control can be implemented with two aquastats (for heating and cooling) wired in parallel to the pump starter.

**Table 2. Pumping Configurations and Control Options**

| *Control Option* | *HP Control* | *Building Pump Control* | *Ground Pump Control* | *KISS Index* |
|---|---|---|---|---|
| 1U | DDC for monitoring | | Request | 3 |
| 1X | Prog Tstat | | Continuous | 1 |
| **Pumping Configuration 1: Single Loop** | | | | |
| 1A | Prog Tstat | | Continuous | 1 |
| 1B | Net T'stats | | Request | 3 |
| 1C | Prog Tstat | | VSD dPress | 4 |
| 1D | Net T'stats | | VSD dP/req | 4 |
| **Pumping Configuration 2: Separate Ground & Building Loops** | | | | |
| 2A | Prog Tstat | Continuous | Continuous | 1 |
| 2B | Net T'stats | Request | Loop Temp | 3 |
| 2C | Prog Tstat | VSD dPress | VSD Temp | 4 |
| 2D | Prog Tstat | Continuous | Loop Temp | 2 |
| 2E | Prog Tstat | VSD dPress | Dual Temp | 4 |
| 2F | Prog Tstat | VSD dPress | Loop Temp | 4 |
| **Pumping Configuration 3: Ground Pump with Separate HP Pumps** | | | | |
| 3A | Prog Tstat | with HP | Continuous | 1 |
| 3B | Net T'stats | with HP | Request | 3 |
| 3C | Prog Tstat | with HP | VSD dT | 4 |
| 3D | Prog Tstat | with HP | Loop Temp | 2 |
| 3E | Prog Tstat | with HP | Dual Temp | 3 |
| **Pumping Configuration 4: Individual HP Pumps** | | | | |
| 4A | Prog Tstat | with HP, Constant Minimum Flow | | 2 |
| 4B | Prog Tstat | with HP, Flow Increases if Fewer HP On | | 2 |

**VSD dPress.** In this control option the pump is equipped with a variable speed drive and controlled to maintain a set differential pressure between the building loop supply and return. The strategy requires a two-position control valve for each heat pump. The valve opens when the heat pump is on. This works well for configurations 2 and 3; however, in configuration 1 the ground loop will experience a large temperature difference, especially at laminar flow conditions. To avoid reduced heat pump efficiency or more antifreeze, option 1C is analyzed with a fairly high minimum flow setting.

**VSD Temp.** In this control option the pump is equipped with a variable speed drive and is controlled to maintain a set building loop return temperature. The ground pump will operate at full speed above 75°F or below 50°F and ramps to minimum speed in between. The VSD controller can better match load than a switching strategy, so the HP EWT is about 5°F closer to ground temperature and heat pump efficiency improves.

**Dual Temp.** Similar to "loop temp" except both pumps (a standby pump is typically installed) are used. The pumps are sized so both pumps operating in parallel meet design flow. The pumps are controlled based on building loop return temperature. The lead pump is operated as described under "Loop Temp" above, with the second pump operated whenever the loop temperature is warmer than 85°F or cooler than 35°F. Setting controls so one pump leads for heating and the other leads for cooling eliminates the need for a lead/lag controller. "Dual Temp" can be implemented with four aquastats installed in the building loop return.

## Analysis and Results

Analyzing the various control/configuration options produced the energy use results shown in Table 3. More efficient results are bolded. To find results, load profiles are matched with ground exchanger performance. Load profiles are generated from monitored 5 minute HP and pump amp draw readings selected for a typical weekday and Sunday for each month. Building loop Btu and pump operating requirements were found for each hour of the typical days. The load data was combined into annual hours of operation at various heating and cooling block load levels as previously shown in Figure 1.

**Table 3. Energy Use of Pump Configuration/Control Options**

| *Pump and Control Configuration* | | | *Pump* | *System* | *Pump* | *Pump* | *Annual Energy Use, MWh* | | | *EUI* |
|---|---|---|---|---|---|---|---|---|---|---|
| *Option* | *Bldg Loop* | *Ground Loop* | *kW* | *"Grade"* | *% Egy* | *kWh/Tonh* | *Pump* | *HP* | *HVAC* | *kWh/sf* |
| **1U** | | Request | 4.53 | D | 47% | 0.77 | 19.4 | 21.7 | 41.1 | 2.3 |
| **1X** | | Continuous | 4.53 | D | 65% | 1.57 | 39.7 | 21.7 | 61.4 | 3.4 |
| **1A** | | Continuous | 1.58 | **A** | 39% | 0.55 | 13.8 | 21.8 | 35.6 | 2.0 |
| **1B** | | Request | 1.58 | **A** | 24% | 0.27 | 6.8 | 21.8 | 28.6 | 1.6 |
| **1C** | | VSD dP | 1.58 | **A** | **12%** | **0.13** | 3.4 | 25.3 | 28.7 | 1.6 |
| **1D** | | VSD dP/req | 1.58 | **A** | **7%** | **0.08** | 1.9 | 25.3 | **27.2** | **1.5** |
| **2A** | Continuous | Continuous | 1.59 | **A** | 39% | 0.55 | 13.9 | 21.8 | 35.7 | 2.0 |
| **2B** | Request | Loop Temp | 1.59 | **A** | 17% | 0.18 | 4.5 | 22.7 | **27.2** | **1.5** |
| **2C** | VSD dP | VSD dT | 1.59 | **A** | **14%** | **0.14** | 3.5 | 22.2 | **25.7** | **1.4** |
| **2D** | Continuous | Loop Temp | 1.59 | **A** | 25% | 0.30 | 7.6 | 22.7 | 30.3 | 1.7 |
| **2E** | VSD dP | Dual Temp | 1.59 | **A** | **12%** | **0.12** | 3.1 | 23.2 | **26.3** | **1.5** |
| **2F** | VSD dP | Loop Temp | 1.59 | **A** | **13%** | **0.14** | 3.5 | 22.7 | **26.2** | **1.5** |
| **3A** | with HP | Continuous | 2.41 | B | 29% | 0.35 | 9.0 | 21.8 | 30.8 | 1.7 |
| **3B** | with HP | Request | 2.41 | B | 18% | 0.19 | 4.8 | 21.8 | **26.7** | **1.5** |
| **3C** | with HP | VSD dT | 2.41 | B | **11%** | **0.10** | 2.6 | 22.2 | **24.8** | **1.4** |
| **3D** | with HP | Loop Temp | 2.41 | B | **10%** | **0.10** | 2.6 | 22.7 | **25.3** | **1.4** |
| **3E** | with HP | Dual Temp | 2.18 | B | **9%** | **0.09** | 2.2 | 23.2 | **25.4** | **1.4** |
| **4A** | with HP, Constant Flow | | 5.22 | F | **9%** | **0.11** | 2.9 | 28.4 | 31.3 | 1.7 |
| **4B** | with HP, Increasing Flow | | 5.22 | F | 17% | 0.19 | 4.9 | 23.9 | 28.7 | 1.6 |

Monitored data of ground exchanger performance (ground-loop temperature differential vs. HP EWT) were graphed for peak winter and summer conditions. Regression analysis of the data was performed to extrapolate loop performance at higher temperature differentials. These factors were then adjusted at various flows and loads for flow characteristics, impact of 10% methanol, heat transfer characteristics, and vertical-bore short-circuiting. Figure 2 shows the results of the ground exchanger analysis.

For each option, the load information and ground exchanger performance are combined to determine the HP EWT that would result at each block load. Heat pump efficiency, pumping kW, and block load operating hours then determine the annual pumping and heat pump energy use. This is totaled as annual HVAC energy use in MWh.

The results shown in Table 3 indicate that neither peak installed pump kW, pumping to total HVAC ratio, nor grade are adequate to predict total HVAC energy use. The best— indicator of overall HVAC performance is the HVAC EUI per building area (kWh/square foot). While EUI is not usually determined during smaller building design, designers can select control and pumping options that are shown to be relatively more efficient.

## Cost-effectiveness Comparisons

Table 4 shows cost differences for controls and pumping on a typical heat pump system. Option 1A was used as the "uncontrolled" baseline. The combined control strategy and pumping configuration are used to find an incremental cost compared to baseline option 1A. While based on a particular building, this comparison gives a good idea of what to expect for the relative cost of pump adjustments, controls, and balancing for the various options.

Table 4. **Cost-effectiveness of Pump Configuration/Control Options**

| | *Pump Control* | | *Annual Savings* | | Pump/Control | | *Benefit-to-Cost Ratio* | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| *Option* | *Bldg Loop* | *Ground Loop* | *MWh* | *$@ .08* | *$ Cost* | *d$cost* | *pub/lo* | *priv/lo* | *pub/hi* | *priv/hi* |
| **1A** | | Continuous | Base | | 13,200 | base | **n/a** | **n/a** | **n/a** | **n/a** |
| **1B** | | Request | 7.0 | 564 | 19,600 | 6,400 | 0.91 | 0.44 | **2.53** | **1.21** |
| **1C** | | VSD dP | 7.0 | 556 | 19,000 | 5,800 | **0.99** | 0.48 | **2.76** | **1.32** |
| **1D** | | VSD dP/req | 8.4 | 672 | 25,200 | 12,000 | 0.58 | 0.28 | **1.61** | 0.77 |
| **2A** | Continuous | Continuous | (0.1) | (10) | 16,700 | 3,500 | **n/a** | **n/a** | **n/a** | **n/a** |
| **2B** | Request | Loop Temp | 8.4 | 671 | 24,300 | 11,100 | 0.63 | 0.30 | **1.74** | 0.83 |
| **2C** | VSD dP | VSD dT | 9.9 | 794 | 26,400 | 13,200 | 0.62 | 0.30 | **1.73** | 0.83 |
| **2D** | Continuous | Loop Temp | 5.4 | 428 | 19,800 | 6,600 | 0.67 | 0.32 | **1.86** | 0.89 |
| **2E** | VSD dP | Dual Temp | 9.3 | 744 | 25,300 | 12,100 | 0.64 | 0.31 | **1.77** | 0.85 |
| **2F** | VSD dP | Loop Temp | 9.4 | 752 | 24,900 | 11,700 | 0.67 | 0.32 | **1.85** | 0.89 |
| **3A** | with HP | Continuous | 4.8 | 387 | 19,900 | 6,700 | 0.60 | 0.29 | **1.66** | 0.80 |
| **3B** | with HP | Request | 9.0 | 718 | 26,200 | 13,000 | 0.57 | 0.27 | **1.59** | 0.76 |
| **3C** | with HP | VSD dT | 10.8 | 867 | 23,100 | 9,900 | 0.91 | 0.43 | **2.52** | **1.21** |
| **3D** | with HP | Loop Temp | 10.3 | 826 | 21,500 | 8,300 | **1.03** | 0.49 | **2.86** | **1.37** |
| **3E** | with HP | Dual Temp | 10.2 | 817 | 22,000 | 8,800 | **0.96** | 0.46 | **2.67** | **1.28** |
| **4A** | with HP, constant flow | | 4.4 | 349 | 18,200 | 5,000 | 0.72 | 0.35 | **2.01** | **0.96** |
| **4B** | with HP, flow increases | | 6.9 | 551 | 18,200 | 5,000 | **1.14** | 0.55 | **3.17** | **1.52** |

The incremental first cost and energy savings as compared to the baseline (1A) were analyzed using a benefit-to-cost ratio[7]. The options are analyzed for two customer types and two electric rates. Based on western US small commercial rates, the "lo" electric rate (including demand charges) was $0.045 per kWh, while the "hi" rate was $0.125 per kWh. Customers' economic horizons vary. The "pub" customer is a public or institutional entity

[7] A benefit-to-cost ratio (BCR) indicates if the economic benefits from a particular investment are greater than the cost. Benefits over the study period are discounted to determine the present value of benefits (PVB). The PVB is divided by the investment cost to find the BCR. A BCR greater than or near 1.0 indicates an acceptable investment. The discount rate used is the cost of money reduced to reflect an expected 1.5% escalation in electric savings above general inflation.

that looked at a 40 year building life with a 6% cost of money. The "priv" customer is a private developer or building owner that looked at a 15 year building life with a 9% cost of money.

The results in Table 4 are highly sensitive to electric rates and customer type: no control options are cost effective for the private customer at low rates (priv/lo) and all options are cost effective for the public customer at high rates (pub/hi). The other two cases give a better indication of the preferred options. Attractive options have good energy efficiency indicators and are also cost effective. Figure 8 displays the HVAC energy use indices and Figure 9 shows the average cost-effectiveness of the pub/lo and the priv/hi cases.

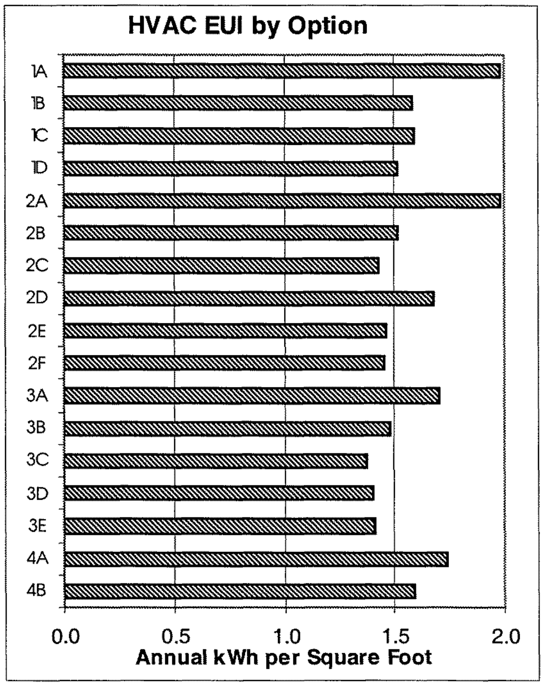


Figure 8. HVAC EUI by Option

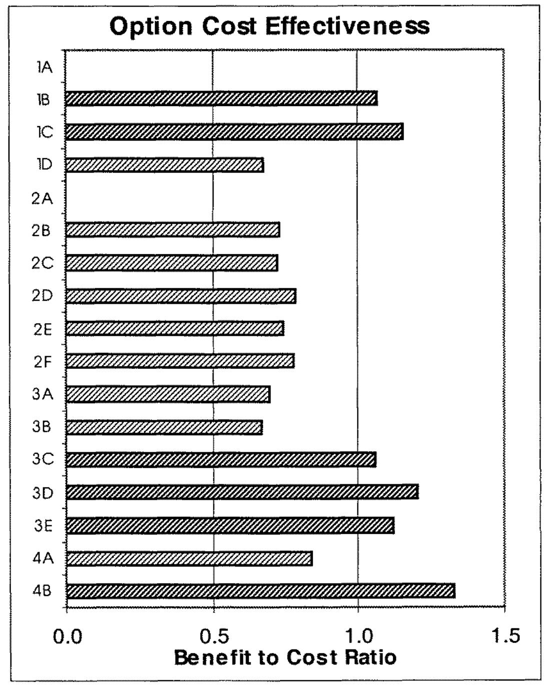


Figure 9. Option Cost-effectiveness

## Recommended Systems and Conclusions

The "winners" for both cost-effectiveness and low EUI are all in pumping configuration 3 with individual HP pumps and a decoupled ground-loop pump. They are 3C (VSDdT), 3D (Loop Temp), and 3E (Dual Temp). From the point of view of simplicity, option 3D with simple temperature control on the ground loop appears to be most appropriate for smaller commercial buildings.[8] Based on cost-effectiveness, other options that improve efficiency when compared to continuous pumping are options 1B (request), 1C (VSDdP), and 4B (with HP and increasing flow). For customers with low electric rates and a short

[8] Pumping configuration 3 also provides advantages to the designer. The HP pumps can be easily selected to match specific HP unit head at around 2.0 gpm per ton. The ground-loop pump can be separately selected based on block loads and ground-loop fluid velocity for a particular installation. The sometimes complex compromises that can be necessary to make one pump serve both loops (as in configurations 1 and 4) are avoided. The designer advantage given by hydraulic decoupling should offset any complexity introduced by requiring design of the building loop separate from the ground loop.

economic horizon, a continuously operating single loop pump (option 1A) may be the most appropriate option.

Control of GSHP water pumps in an appropriate pumping configuration can reduce total HVAC energy use by 30%. Operating hours of pumps should be considered in selecting a control strategy for a ground-coupled heat pump system. Before considering control options, proper bore field sizing and pump flow selection can provide up to 40% energy savings. Additional research, monitoring, and reporting of various control strategies, especially variable speed drives in seasonal operation, will help the industry determine the most cost effective approach to controlling the ground-coupled heat pump system.

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