White Paper

Savings Calculations for Ground Source Heat Pumps with Central Pumping

For application in Multi-Family Buildings and Other Commercial Applications

March 9, 2020
Revised March 2021

Prepared for

NYSERDA and
NYS Department of Public Service

Prepared by:

Hugh I. Henderson, Jr. , P.E.
Owahgena Consulting, Inc.
Cazenovia, NY 13035

Overview

This white paper is intended to provide the background and basis for energy savings calculations for larger ground source heat pump (GSHP) systems that have centralized loop pumping.

GSHPs are often used in larger buildings where separate GSHP units are applied to meet heating and cooling loads for individual zones. Each heat pump is typically 1 to 4 tons, though larger units are possible on bigger spaces (e.g., a school gymnasium). The result can be hundreds of HPs all tied together on a common ground loop. Each HP is controlled by a thermostat in the local zone. This type of system arrangement can be used in multi-family buildings, schools, offices, etc. In this type of system, the central loop pumps must run continuously throughout the year.

Centralized Loop Pumping

If the central pumps are sized to meet the design flow requirements and run all the time, then pumping energy can exceed the energy use of all the GSHP units[1]. Therefore, the central loop pumps in a large building are usually controlled to vary the flow depending on the number of HPs that are operating. In order to have flow control on the central pumps, each HP is typically installed with two-way valve[2] that only allows water flow through the HP unit when it is activated. Otherwise, the valve is closed and the flow requirements for the central pumping system are reduced.

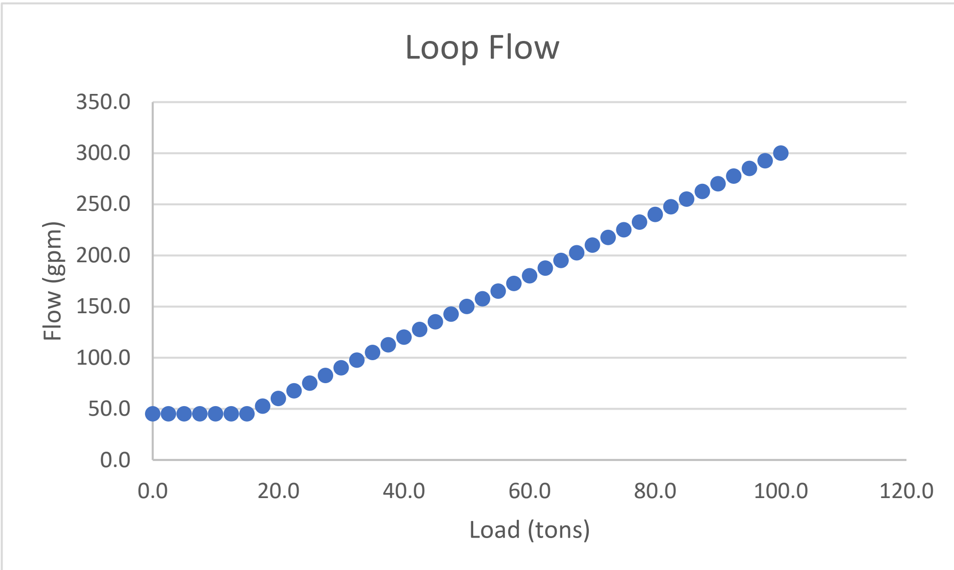
There are three common methods to vary flow

- Install a variable speed drive (VSD) to vary pump speed in order to maintain the required pressure difference across all the heat pumps
- Install a two-speed motor that can operate at a lower speed (usually 60% of full speed). Usually the change in speed is driven by a pressure difference measurement in building loop.
- Use a variable speed pump with internal controls to modulate speed to maintain a constant pressure difference across a range of flows. These controllers use an “open loop” or sensorless control approach that attempts to mimic differential-pressure control without requiring a pressure sensor out in the building loop. The controller infers the pressure difference (at the pump) from measured current and speed. These pumps are common in small and medium applications up to 300-400 gpm.

The advantage of variable speed pumping is that pumping power is substantially reduced at lower flow rates. According to the pump affinity laws, power should drop as the cube (third power) with flow. This translates to a decrease to 50% flow should theoretically result in power at 12.5%. In practice, this same level power reduction does not occur in a pumping system controlled to maintain a constant differential pressure. The power reduction with flow for a typical application is shown by Figure 1. While loop flow

[1] A high school in Boone, TN was monitored in detail in 1998-1999 and measured pumping power exceeded energy use of the HPs on an annual basis (Henderson and Walburger 1999).
[2] If a GSHP unit was dual stage or variable speed, then a modulating flow valve could also be used to proportionally limit water flow through the unit, but this option is rarely use in practice.

reduces proportionally with the building load, the pumping power decreases more rapidly. In this case a minimum flow that is 15% of the design flow is assumed. This minimum flow is maintained by installing some HPs without a two-way valve so that some loop flow still occurs when all the HP are off. This allows the variable speed pump to continue operating under all conditions across the year.



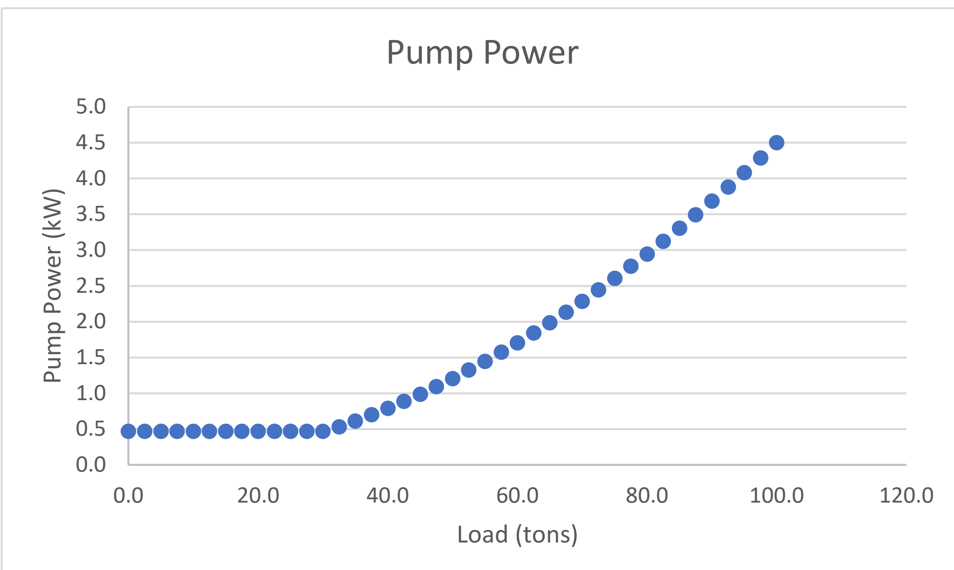


Figure 1. Trend of Flow and Power with Load in Hypothetical 100-ton GSHP application (exponent =2.2, 10.4% min)

The relationship between power and flow above is based on the equation below, where flow and power are expressed as a percent of the full load or design values.

$$[\%Power] = [\%Flow]^n \quad \text{where } n \sim 2.2$$

Another commonly use relationship is

$$[\%Power] = c0 + c1 \times [\%Flow] + c2 \times [\%Flow]^2$$

An example polynomial is given by Romberger (2014) from the Uniform Methods Project. Similar default curves are also available in DOE-2 and other simulation software. The figure below compares the polynomial and exponent curves. These two methods are in close agreement as long as minimum power value of 10.4% is applied to the exponent. We apply the exponent method for the remainder of this document.

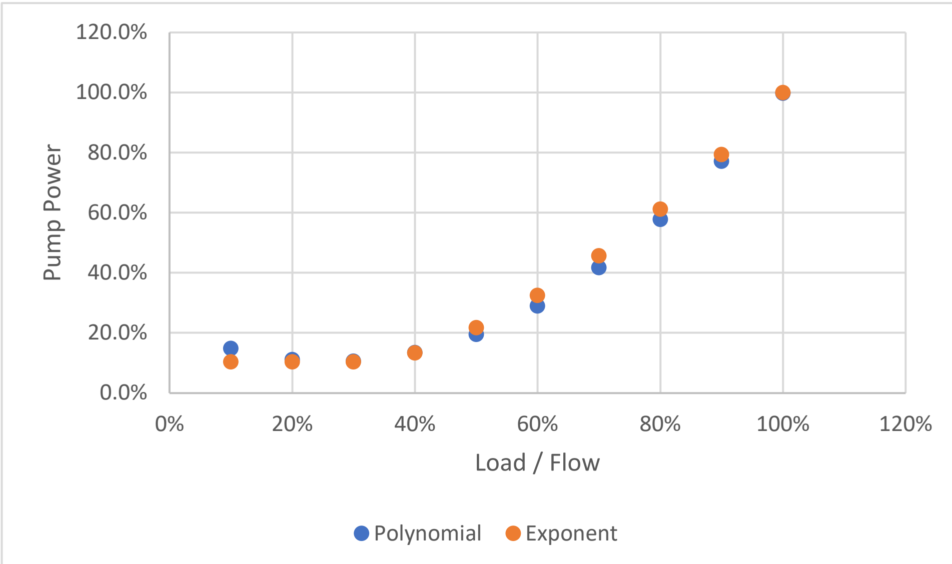


Figure 2. Comparing Polynomial and Exponent Curves to Predict Pumping Power (exponent is 2.2 and minimum power is 10.4%)

Bin Analysis to Predict Annual Pumping Power

The bin analysis relates the building load trends to outdoor temperature. Then the loop flow and pumping power are predicted based on the building loads. In commercial buildings, each HP is sized to meet the zone loads therefore the loads are not coincident. As a result, only faction of the heat pumps are typically operating at the design temperatures. In this case, we assume that 90% of the HPs are operating at the heating design temperature and 80% at the cooling design temperature.

The temperature bin hours are used to predict the annual average power percentage for both VSD case and for the dual stage pump case. The sensorless VSD case will not be as precisely controlled so the pump settings will have to be conservatively set. Therefore, we have assumed that the average power for the sensorless VSD will be 5% higher than the traditional case (Niu et al 2016).

Figure 3 shows the trend of load, loop flow and pump power for the VSD and staged scenarios.

Table 1. Bin Analysis to Determine Average Pumping Power Using Albany TMY Data

Temp	Albany Hours	Heating Load (% HPs)	Cooling Load (% HPs)	Loop Flow (%)	VSD Pump Power (%)	Staged Pump Power (%)
-29	0	128%	0%	100%	100%	100%
-27	0	125%	0%	100%	100%	100%
-25	0	122%	0%	100%	100%	100%
-23	0	119%	0%	100%	100%	100%
-21	0	117%	0%	100%	100%	100%
-19	0	114%	0%	100%	100%	100%
-17	0	111%	0%	100%	100%	100%
-15	0	108%	0%	100%	100%	100%
-13	0	105%	0%	100%	100%	100%
-11	0	103%	0%	100%	100%	100%
-9	0	100%	0%	100%	99%	100%
-7	0	97%	0%	97%	93%	100%
-5	0	94%	0%	94%	88%	100%
-3	5	91%	0%	91%	82%	100%
-1	10	89%	0%	89%	77%	100%
1	16	86%	0%	86%	71%	100%
3	20	83%	0%	83%	66%	100%
5	14	80%	0%	80%	62%	100%
7	55	77%	0%	77%	57%	100%
9	58	75%	0%	75%	53%	100%
11	76	72%	0%	72%	48%	100%
13	80	69%	0%	69%	44%	100%
15	152	66%	0%	66%	40%	100%
17	154	63%	0%	63%	37%	100%
19	179	61%	0%	61%	33%	100%
21	168	58%	0%	58%	30%	33%
23	132	55%	0%	55%	27%	33%
25	214	52%	0%	52%	24%	33%
27	265	50%	0%	50%	21%	33%
29	256	47%	0%	47%	19%	33%
31	255	44%	0%	44%	16%	33%
33	369	41%	0%	41%	14%	33%
35	229	38%	0%	38%	12%	33%
37	278	36%	0%	36%	10%	33%
39	243	33%	0%	33%	10%	33%
41	124	30%	0%	30%	10%	33%
43	226	27%	0%	30%	10%	33%
45	210	24%	0%	30%	10%	33%
47	282	22%	0%	30%	10%	33%
49	281	19%	0%	30%	10%	33%
51	455	16%	0%	30%	10%	33%
53	259	13%	0%	30%	10%	33%
55	300	10%	0%	30%	10%	33%
57	298	8%	0%	30%	10%	33%
59	202	5%	4%	30%	10%	33%
61	341	2%	10%	30%	10%	33%
63	294	0%	15%	30%	10%	33%
65	322	0%	21%	30%	10%	33%
67	275	0%	27%	30%	10%	33%
69	393	0%	32%	32%	10%	33%
71	245	0%	38%	38%	12%	33%
73	235	0%	44%	44%	16%	33%
75	199	0%	49%	49%	21%	33%
77	97	0%	55%	55%	27%	33%
79	141	0%	60%	60%	33%	100%
81	123	0%	66%	66%	40%	100%
83	97	0%	72%	72%	48%	100%
85	63	0%	77%	77%	57%	100%
87	37	0%	83%	83%	66%	100%
89	15	0%	88%	88%	76%	100%
91	9	0%	94%	94%	87%	100%
93	3	0%	100%	100%	99%	100%
95	5	0%	105%	100%	100%	100%
97	1	0%	111%	100%	100%	100%
99	0	0%	116%	100%	100%	100%

Assumptions

Albany	Design Temp (F)	Balance Pt (F)
Heating	-2	62.5
Cooling	86	57.5

Heating Load at Design: 90%
Cooling Load at Design: 80%
Minimum/Bypass Flow: 30%

Results

Annual Average Pump Power
VSD Pump Avg = 18.1%
Staged Pump Avg = 42.6%

Annual kWh =
[Avg Pump Pwr %] x [design pump kW] x [8760 hrs]

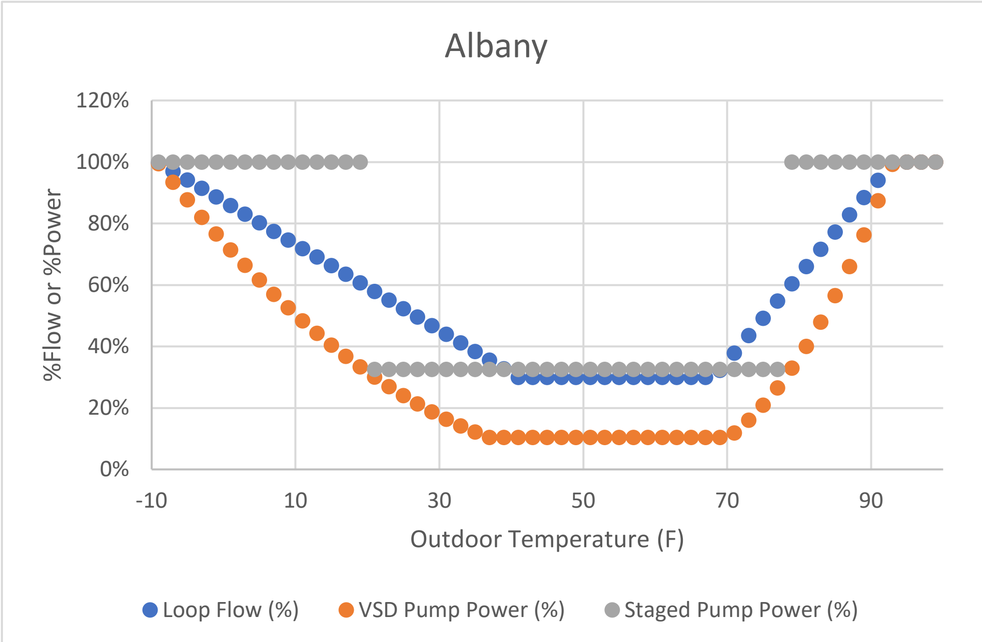


Figure 3. Trend of Flow and Power Used in Bin Analysis for Albany

Note that in case above, the minimum pump power use is 10% of design when the bypass flow is 30% of design. If raise the bypass flow to 40% of design, then the minimum power is 13% of design. At 50% bypass flow the minimum power becomes 22% and at 60% bypass it becomes 33%.

Another key assumption is the diversity of the heating and cooling loads. In the case above we assumed that 80% of the building HPs will operate in cooling at the summer design conditions (86°F for Albany) and 90% will operate in heating at the with design condition (-2°F for Albany). A study of four commercial buildings with multiple small water source and ground source heat pumps demonstrated that most buildings never have more than 50-60% of the heat pumps operating at one time (Henderson 1999). The amount of loads diversity affects the loop flowrate and therefore impacts the pumping power as well.

Similarly, a new multi-family/mixed use building in Syracuse has 185 tons of heat pumps installed in 114 apartments with a design building loop flow rate of 555 gpm (the max loop flow is 1000 pm to cover future retail tenants). In the first two years of operation, the measured loop flow has never exceeded 68% of the design flow, or 38% of the maximum available flow. The minimum loop pumping power has been less than 10% of design.

Table 2 shows the average annual pumping power for each weather city using TMY data and the local design conditions. We show the average pumping power for 1) the traditional VSD, 2) the two-speed, staged pump, and 3) the sensorless VSD. Table also includes results for:

- four levels of bypass flow (and the corresponding minimum pumping power): 30, 40, 50 and 60%
- two levels of load diversity,
    - modest diversity (80% cooling & 90% heating) and
    - more diversity (60% cooling & 70% heating)

Table 2. The Average Pump Power Percentage for Each Weather City and With Other Assumptions

	Albany	Binghamton	Buffalo	Massena	New_York	Poughkeepsie	Syracuse
	Bypass Flow = 30% (10% min Power) & Diversity = 80%/90%						
Traditional VSD	18.1%	19.5%	19.5%	18.6%	19.2%	20.7%	19.1%
Two-Stage	42.6%	42.9%	43.1%	41.6%	40.9%	44.5%	43.0%
Sensorless VSD	19.0%	20.5%	20.5%	19.5%	20.1%	21.7%	20.0%
	Bypass Flow = 40% (13% min Power) & Diversity = 80%/90%						
Traditional VSD	19.7%	21.1%	21.1%	20.4%	20.7%	22.4%	20.6%
Two-Stage	42.6%	42.9%	43.1%	41.6%	40.9%	44.5%	43.0%
Sensorless VSD	20.7%	22.2%	22.1%	21.4%	21.8%	23.5%	21.7%
	Bypass Flow = 50% (22% min Power) & Diversity = 80%/90%						
Traditional VSD	25.6%	26.8%	26.7%	26.3%	26.2%	28.0%	26.2%
Two-Stage	42.6%	42.9%	43.1%	41.6%	40.9%	44.5%	43.0%
Sensorless VSD	26.8%	28.1%	28.0%	27.6%	27.5%	29.4%	27.5%
	Bypass Flow = 60% (33% min Power) & Diversity = 80%/90%						
Traditional VSD	34.3%	35.4%	35.2%	35.0%	34.7%	36.5%	34.8%
Two-Stage	42.6%	42.9%	43.1%	41.6%	40.9%	44.5%	43.0%
Sensorless VSD	36.1%	37.2%	37.0%	36.8%	36.4%	38.3%	36.5%
	Bypass Flow = 30% (10% min Power) & Diversity = 60%/70%						
Traditional VSD	13.0%	13.7%	13.7%	13.4%	13.4%	14.7%	13.4%
Two-Stage	34.0%	35.1%	35.1%	35.0%	34.2%	36.6%	33.8%
Sensorless VSD	13.7%	14.4%	14.4%	14.1%	14.1%	15.4%	14.1%
	Bypass Flow = 40% (13% min Power) & Diversity = 60%/70%						
Traditional VSD	15.2%	15.9%	15.8%	15.6%	15.5%	16.8%	15.5%
Two-Stage	34.0%	35.1%	35.1%	35.0%	34.2%	36.6%	33.8%
Sensorless VSD	16.0%	16.7%	16.6%	16.4%	16.2%	17.6%	16.3%
	Bypass Flow = 50% (22% min Power) & Diversity = 60%/70%						
Traditional VSD	22.4%	22.9%	22.8%	22.8%	22.6%	23.7%	22.6%
Two-Stage	34.0%	35.1%	35.1%	35.0%	34.2%	36.6%	33.8%
Sensorless VSD	23.5%	24.1%	23.9%	23.9%	23.7%	24.9%	23.7%
	Bypass Flow = 60% (33% min Power) & Diversity = 60%/70%						
Traditional VSD	32.6%	32.8%	32.7%	32.8%	32.7%	33.5%	32.7%
Two-Stage	34.0%	35.1%	35.1%	35.0%	34.2%	36.6%	33.8%
Sensorless VSD	34.3%	34.5%	34.3%	34.5%	34.4%	35.2%	34.3%

To determine annual pumping energy:
Pumping kWh = [Avg Annual Pumping Power, %] x [design pumping power, kW] x [8760 hours per year]

Another factor that affects pumping energy use is the differential pressure that must be maintained across the water-side of the heat pump units. Most HP units only require 4-6 psi to be maintained to achieve the required flow rate. If hydronic components such flow limiters and other valves are added to the system, then the necessary differential pressure is typically 6-10 psi. Mescher (2009) lists some of the hydronic components that should be considered. In some variable speed pumping systems, differential pressures as high as 18 psi have been required because of poor component/equipment selection (Henderson et al 2000). Pressure differentials over 10 psi limit the potential turn down in pumping power. In systems with excessive differential pressure requirements, the case with bypass flow equal to 60% (and a minimum pumping power of 33%) can be used represent the annual performance of this case.

Other Possible Pumping Arrangements

The two variable speed pumping arrangements discussed above are shown in Figure 4. Other arrangements

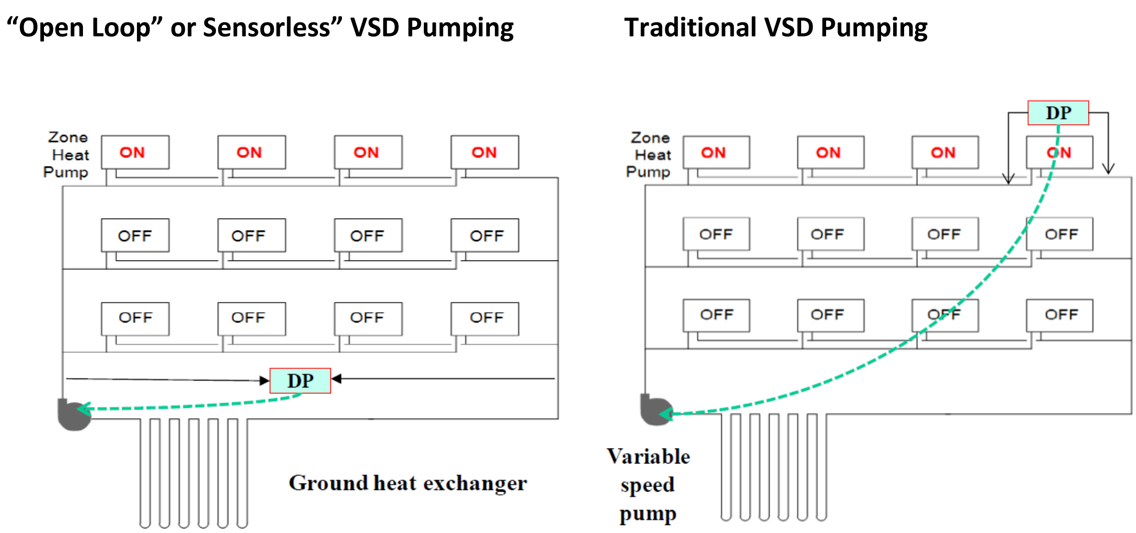


Figure 4. Schematic of Variable Flow Pumping Arrangements for GSHP Systems with Central Pumping (from Lui et al 2017)

Hart and Price (2000), Kavanaugh and McInerny (2001) and Mescher (2009) all point out another system and pumping arrangements that has been used to provide similar or better energy performance than traditional VSD pumping systems but with lower installed costs. In this arrangement (Figure 5) the two-way valve and other flow limiting components on each heat pump are replaced with individual cartridge pumps similar to those used in residential GSHP systems. The individual pumps in this case are only sized to overcome the pressure drop of the GSHP unit. The induvial pumps have lower pump efficiency and result in only a linear decrease in pumping power. While this approach sometimes results in annual pumping power just slightly more than the best-designed central VSD systems, it simplicity and lower cost can make it a reasonably energy efficient choice when the ground loop pump are variable speed. The speed of the ground loop pump can controlled based on a combination of loop return temperatures and the building side temperature difference

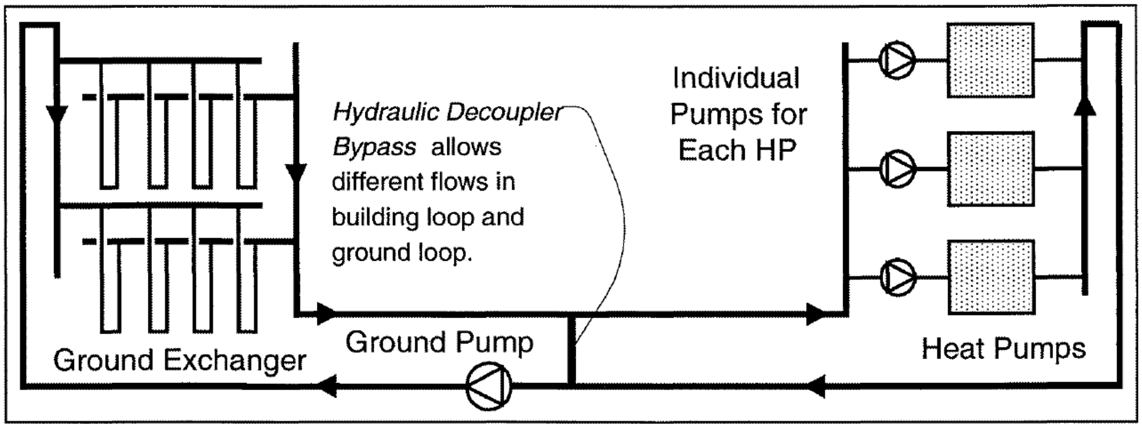


Figure 5. Schematic of Variable Flow Pumping Arrangements for GSHP Systems with Central Pumping and dedicated Cartridge Pumps on each HP (from Hart and Price 2000)

Loop Temperature for Heating and Cooling

The table below compares the average loop temperatures during the heating and cooling season for various building types. The average loop temperatures from 49 GSHP systems in Upstate NY are shown as are lightly loaded multi-family building and large dormitory at SUNY Albany. Generally, residences in Upstate NY have a bias towards heating heat rejection in the summer is 43% of heat extraction. The mixed-use building in Syracuse had even a lower percentage of heat rejection on a ground loop that is currently oversized due to unoccupied retail spaces. The large 500-bed dorm in Albany had a rejection-to-extraction ratio of 127%, which is more in line with other commercial buildings in NY. Large commercial office buildings and hotels in Zone 5 climates typical have ratios that are near 300% (Henderson 1999).

	Residential Upstate (from TRM)	**Iron Pier Multi-Family & Mixed Use, Syracuse (184,000 sq ft)**	**U Albany Dormitory (500 bed)**
Annual Heat Rejection-to-Extraction Ratio	0.43	0.21	1.27
Average Entering Water Temperature for Heating (°F)	40	48	54
Average Entering Water Temperature for Cooling (°F)	68	64	69

We recommend that the TRM request that average loop temperature in the heating and cooling mode be reported. If the average loop temperatures in heating and cooling ($EWT_{avg,h}$ and $EWT_{avg,c}$) are not available, we recommend that the following approximation be used:

$$EWT_{avg,h} = EWT_{min} + 10°F$$

$$EWT_{avg,c} = EWT_{max} - 10°F$$

Where EWT_{max} and EWT_{min} are the design temperatures from the ground loop sizing calculations.

References

Hart, R. and W. Price. 2000. Improving Ground-Source Heat Pump Efficiency: Optimizing Pumping Control and Configuration of Closed-Loop Heat Pump Systems in Smaller Commercial Buildings. ACEEE Summer Session Panel 10.

Henderson, H.I. 1999. Implications of Measured Commercial Building Loads on Geothermal System Sizing.' SE-99, ASHRAE Transactions, Vol. 106. Pt. 2. June.

Henderson, H. and A. Walburger. 1999. GEOEXCHANGE SYSTEM MONITORED PERFORMANCE DANIEL BOONE HIGH SCHOOL - FINAL REPORT. Prepared for the Geothermal Heat Pump Consortium, Washington, DC. Prepared by CDH Energy Corp., Cazenovia, NY. September.

Henderson, H.I., M. Khattar, S. Carlson, and A. Walburger. 2000. 'The Implications of the Measured Performance of Variable Flow Pumping Systems in Geothermal and Water Loop Heat Pump Applications.' MN-00, ASHRAE Transactions, Vol. 108. Pt. 2. June.

Niu, F. X. Liu, and Z. O'Neill. 2016. A Simulation-Based Study on Different Control Strategies for Variable-Speed Pumps in Distributed Ground-Source Heat Pump Systems. ASHRAE Transactions, St Louis. ST-16-018.

Romberger, J. 2014. "Chapter 18 - Variable Frequency Drive Evaluation Protocol" report as part of the The Uniform Methods Project: Methods for Determining Energy Efficiency Savings for Specific Measures. Subcontract Report NREL/SR-7A40-63166. November.

Liu, X. P. Hughes, A. Gehl, S. Hern, and D. Ellis, 2017. Advanced Controls for Ground-Source Heat Pump Systems. ORNL. CRADA final report for CRADA number NFE-13-04586. ORNL/TM-2017/302. June.

Mescher, K. 2009. Simplified GSHP Design: One-Pipe Geothermal Design. ASHRAE Jornal, Atlanta, GA. October