Effects of Ground Heat Exchanger Design Flow Velocities on System Performance of Ground Source Heat Pump Systems in Cold Climates

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ABSTRACT

Industry guidelines for ground source heat pump (GSHP) systems provide recommendations for minimum and maximum flow rates, so as to keep the ground loop fluid flow rate sufficiently high to minimize the convective heat transfer resistance at the inside wall of the U-tube, while at the same time keeping the flow rate sufficiently low to minimize the pumping power. The same industry guidelines treat the convective heat transfer resistance as negligible if the flow is turbulent. The system flow and flow regime vary with viscosity, which varies with temperature, and in heating-dominated climates where heat carrier fluid temperatures go low in wintertime and require increasing antifreeze concentrations, the industry guideline recommendations for flow rates and head loss are difficult to meet. This paper describes a simulation-based study of design flow velocities for a ground-source heat pump system in a heating dominated climate – Sioux Falls, South Dakota. The effect of pipe diameter and Reynolds number are investigated and the effects on system performance and head loss are analyzed. An experimentally-validated ground heat exchanger model is used and pumping energy, heat pump energy and backup electric resistance heating energy are all accounted for in the study.

INTRODUCTION

The design of ground source heat pump (GSHP) systems necessarily involves a tradeoff between keeping the flow rate high, so as to minimize the convective heat transfer resistance at the inside wall of the U-tube, and keeping the flow rate low, so as to minimize the pumping power. Typical industry recommendations (IGSHPA 2009) for residential systems suggest a minimum flow rate keeping the Reynolds number at 2500 or higher and a maximum flow rate preferably giving a head loss of 1-3 ft of head/100 feet of pipe (1-3 m/100 m) but, if necessary, a head loss of 4 ft of head/100 feet of pipe (4 m/100 m) is acceptable. Kavanaugh (2011) warns that high fluid velocities may result in high pumping power with little thermal and even less economic advantage over non-laminar flow during part load conditions. Mescher (ASHRAE 2011) suggests that a properly designed borefield should be kept to less than 25 ft (7.6 m) of head loss with a total system pressure drop of maximum 50 ft (15.2 m).

In heating-dominated climates it becomes increasingly difficult to stay within the above limits as the heat carrier fluid temperature decreases and the required antifreeze concentrations increase. For example, at ethanol concentration of 20% by weight and a fluid temperature of 23°F (-5°C), a Re=2500 flow in ¾” SDR-11 (I.D. 21.83 mm) tubing will have a head

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loss of 10.0 ft/100 ft (10.0 m/100 m). At the same conditions, a Re=2500 flow in 1"SDR-11 (I.D. 27.33 mm) tubing will have a head loss of 5.1 ft/100 ft (5.1 m/100 m). Both of these values exceed the design recommendations for head loss.

The same industry guidelines treat the convective heat transfer resistance as negligible if the flow is turbulent. This approach goes back at least 30 years (Bose et al. 1985). But several complicating factors suggest that this approach be revisited: the system flow and flow regime (laminar, turbulent) will vary with viscosity, which varies with temperature; at Reynolds numbers around 2500, the flow may be laminar or in the transition region but will not be fully turbulent; and, all other things being equal, lower flow rates give more favorable approach temperatures into the heat pump.

In this paper an experimentally-validated ground heat exchanger model is used to investigate the effects of pipe diameter and flow rate, on system performance and economics for a ground-source heat pump system in Sioux Falls, South Dakota. Pumping energy, heat pump energy and backup electric resistance heating energy are all accounted for.

METHODOLOGY

In this study system Reynolds number, head loss, SCOP and electricity consumption for heat pump, electric resistance heating and circulation pump at varying design flow rates and pipe diameters are studied for a typical GSHP system for a house in a heating-dominated climate. A simulation tool using heating and cooling loads modeled in Energy Plus, a heat pump model featuring a North American type water-to-air with backup electric-resistance heating, and a ground heat exchanger model, was used to investigate the effect of changing viscosity and resulting variations in the Reynolds number due to temperature variations over the year. The simulation tool used is described in detail by Spitler et al. (2014a), and has been used in previous studies by Gehlin & Spitler (2014 a & b) and Spitler et al. (2014b).

Description of the House and GSHP System

A prototype house (Spitler et al. 2014) located in Sioux Falls, South Dakota was modeled in Energy Plus to determine the hourly heating and cooling loads for a typical weather year. Annual total heating loads are 37 MMBTU (10,778 kWh); annual total cooling loads are 5 MMBTU (1478 kWh). The building peak heating load is 22.9 MBTU/hr (6.7 kW) and building peak cooling load is 8530 BTU/hr (2.5 kW). The house is heated by a water-to-air heat pump with nominal cooling capacity of two tons (7.0 kW), equipped with a backup electric resistance heater.

The ground heat exchanger consists of two boreholes with a spacing of 15 ft (4.6 m). The ground thermal conductivity is 1.63 BTU/hr·ft·°F (2.82 W/m·K), and the volumetric heat capacity is 32 BTU/ft^3·°F (2160 kJ/m^{3}K). Ground temperature is set to 48.4°F (9.1°C). The boreholes are fitted with single U-tubes, and backfilled with standard bentonite grout. The U-tubes are installed without spacers and are assumed to have the tubes located so that the spacing between the two tubes is the same as the spacing between each tube and the borehole wall.

Heat Pump Model

The heat pump system model used is a simple equation fit model of a water-to-air heat pump model with backup electric-resistance heating as might be used in the northern part of North America. The heating capacity and energy consumption (including compressor and intermittently-operated fan) of the heat pump is determined with an equation fit to source-side entering fluid temperature and coefficients are fit for a specific flow rate. The electric resistance heating is activated any hour the heating demand exceeds the available heat input. The model is an extended version of the model described by Spitler (2000) and was used and described in Spitler et al. (2014a).

Pumping/Piping Model

For this work, with a large number of cases, several simplifying approximations have been made. These include assuming a fixed mass flow rate and calculating the pump energy based on the calculated head loss, flow rate, hourly run-time fraction, and an assumed circulating pump efficiency of 25%. That is we assumed that such a pump could be obtained. Kavanaugh et al. (2003) measured circulating pump efficiencies for small circulators and maximum efficiencies were around 28%, though they could be considerably lower depending on the operating point.

Dissipation of pumping energy can be problematic in cooling-dominated applications with poorly designed pumping/piping systems. For heating-dominated applications, it should be less problematic. For the model shown here, we have only calculated the required pumping energy and not included the effect of dissipated pumping energy on the system performance. It is expected that it may reduce some of the required resistance heating energy and heat pump heating energy, but we have not accounted for it here.

Ground Heat Exchanger Model

The ground heat exchanger model used in this study is an extension to that originally described by Yavuzturk and Spitler (1999) and several validations have been previously described, e.g. Gentry et al. (2006). This version allows for time-varying borehole resistance and accounts for short-circuiting. G-functions are used to calculate the temperature response of the ground heat exchanger to a series of heat rejection/extraction rates. The heat extraction and rejection loads on the ground heat exchanger are devolved into a series of step inputs, then the g-function is used to determine the response due to each step input, and the temperature responses are superimposed to determine the evolution of borehole temperatures with time.

The fluid-to-borehole-wall resistance and the fluid-to-fluid short-circuiting resistance are calculated at each time step using Hellström's (1991) first-order multipole method. (See Chapter 8 of reference). The convective resistance is calculated using the Gnielinski (1976) correlation for turbulent flow at Re> 3000. For laminar flows at Re<2300 a fixed value of Nusselt number (4.36) is used. For the presumed transition region, a linear interpolation is used, as shown in Figure 1 for Prandtl number of 50, approximately corresponding to the value for a 20% by weight ethanol antifreeze mixture at a temperature of 32°F (0°C).

Description of Test Cases

In order to examine the effects of flow velocity independent from other factors, a series of numerical experiments were run for three basic configurations: two 230 ft (70 m) boreholes in parallel, two 230 ft (70 m) boreholes in series, and two 262 ft (80 m) boreholes in series. The series configuration (and a sufficiently small pipe diameter) would be needed at the design flow rate if turbulent flow were required to always be maintained under all temperatures. (For reasons that will be demonstrated below, this is not a good idea). All simulations utilized the same heat pump, the same building loads, the same connecting piping, the same antifreeze (20% by weight ethanol) and the same flow rate of 6 GPM (0.38 l/s). The heat pump manufacturer gives a catalog data range of 3 GPM (0.19 l/s) to 6 GPM (0.38 l/s); we chose the highest value to make it easier to obtain turbulent flow. All simulations were done for a two-year period and results are presented for the 2nd year, for which the performance is deemed to be typical of other years, given that only one or two boreholes are used.

For each of the three configurations, the numerical experiments involved simulations of eleven different pipe sizes, all of DR-11 (that is the ratio of the outer diameter to wall thickness is 11). The smallest pipe diameter corresponds to ¾"SDR-11 and the largest corresponds to 1 ½"SDR-1. In between, the pipe diameter was varied in uniform 0.866" (2.2 mm) increments. This means that most of the pipe sizes are not commercially available, but it has the advantage of holding everything constant except for the flow velocity, resulting convective resistance and the thermal resistance of the grout. Neither 1" nor 1 ¼"SDR-11 pipes have been included but the diameter of 1"SDR-11 is just 1% larger than the case labeled "O.D. 33.1mm" and the diameter of 1 ¼"SDR-11 is less than 1% larger than case labeled "O.D. 41.8mm", so the interested reader can take those cases as proxies for the commercially available pipe sizes.

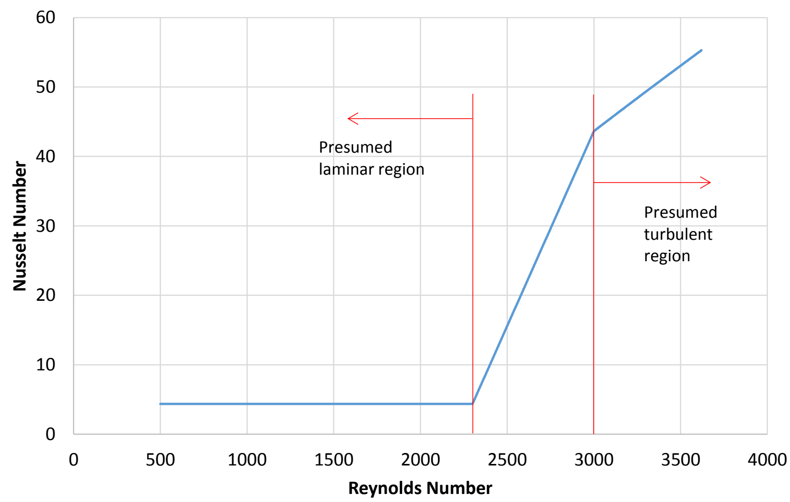

Figure 1 Interpolation between Re=2300 and Re=3000 for Prandtl = 50.

RESULTS

Figure 2 shows how Reynolds number varies due to viscosity changing with temperature over the course of the second year of operation for ¾" and 1 ½"SDR-11 and at an intermediate DR-11 pipe with O.D. of 1.48"(37.5 mm) for both the serial and parallel configurations. For the serial configuration, flow conditions in the pipes are turbulent throughout the year for the smallest pipe size. However, for the larger pipe sizes, Reynolds number falls into the laminar region for a significant number of hours. For the parallel configuration, all pipe sizes lead to laminar flow for most of the hours of the winter.

The resulting electricity use during the second year of operation for the two serial configurations is shown in Figures 3 and 4. As indicated in the legend, the electricity usage is broken down by category. As expected, the pumping energy is quite high for the serial configurations with small-diameter pipe required to maintain all turbulent flow, all the time. And, though the resistance heating energy and heat pump energy used for heating increase with increasing pipe size, this increase is more than outweighed by the decrease in circulating pump energy. Thus, the total electrical energy required decreases with increasing pipe size, even though the flow is laminar for increasing number of hours. As can be seen, the optimum is relatively flat, with little reduction in total electricity consumption for pipe diameters around 1"SDR-11 and larger.

The annual total electricity use and the contributions from heat pump energy and pump energy for heating and cooling mode respectively, and electricity for resistance heating, are shown at various pipe sizes in Figures 3 and 4. As pipe size increases, total electricity use decreases due to decreasing use of electricity for the circulation pump. The optimum is relatively flat, at the larger borehole dimensions (from O.D. of 1.48" (37.5 mm) and larger). A slight increase in use of resistance heating as the degree of turbulence decreases with larger pipe size, but this increase is well compensated for by the decrease in pumping electricity.

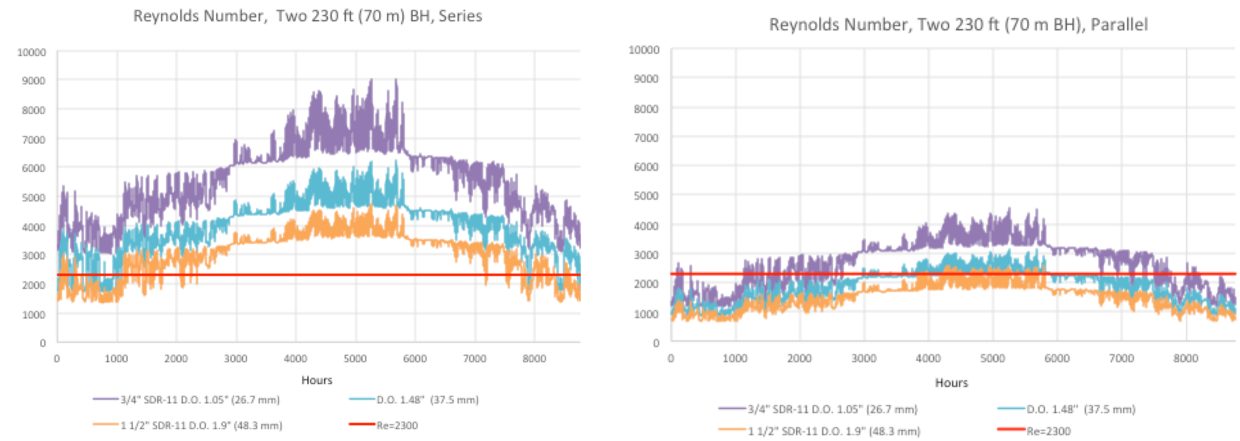

Figure 2 Reynolds Number for boreholes in serial and parallel configurations

Comparing Figures 3 and 4, deeper boreholes are favorable for the overall system performance as long as the pipe size is not so small that the increase in pumping electricity exceeds the effect of the more favorable temperatures given by the deeper borehole. For pipe sizes of O.D. 1.39" or larger, the total electricity use is roughly 120 kWh less for the deeper boreholes.

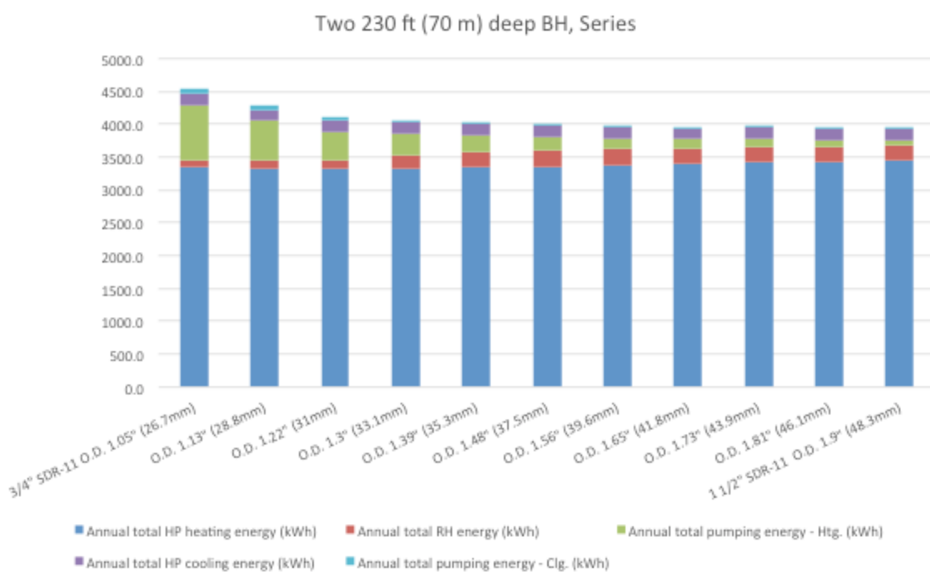

Figure 3 Electricity Use for 230 ft (70 m) Boreholes in Series. DR-11 Pipes ¾" to 1 ½" (O.D. 26.7-48.3 mm).

The electricity usage during the second year of operation for the parallel case is shown in Figure 5. As pipe size increases, total electricity use decreases very slightly due to decreasing use of electricity for the circulation pump. In fact, it is somewhat remarkable that the electricity usage varies so little (only about 120 kWh) over the range of pipe sizes. However, since most of the heat pump energy consumption is during the winter, most of the hours for all pipe sizes will be in laminar flow where the Nusselt number is approximately fixed. For a fixed Nusselt number the convective resistance will remain constant, so the only difference will be in the small number of hours with turbulent flow and the small changes in the grout resistance as the pipe size changes.

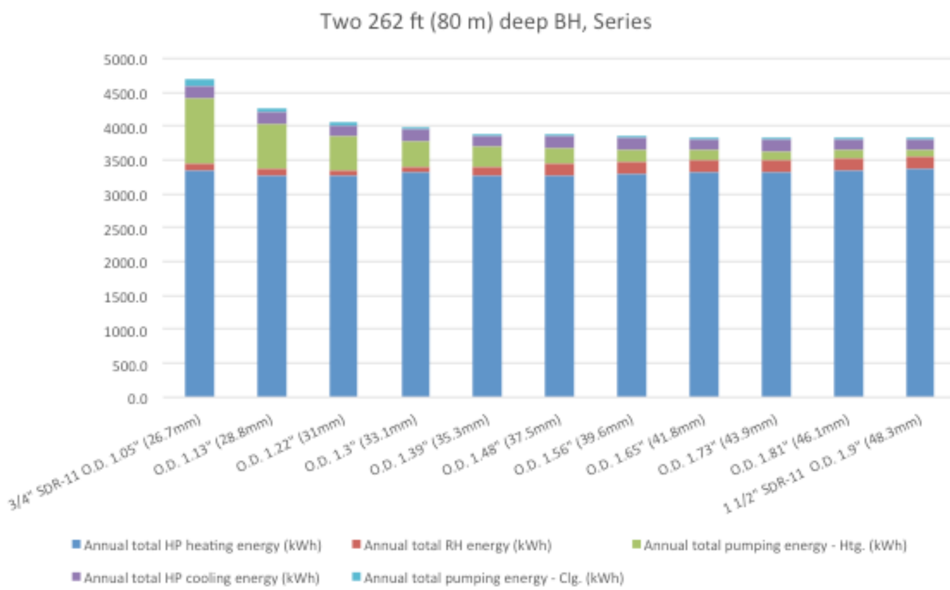

Figure 4 Electricity Use for 262 ft (80 m) Boreholes in Series. DR-11 Pipes ¾" to 1 ½" (O.D. 26.7-48.3 mm).

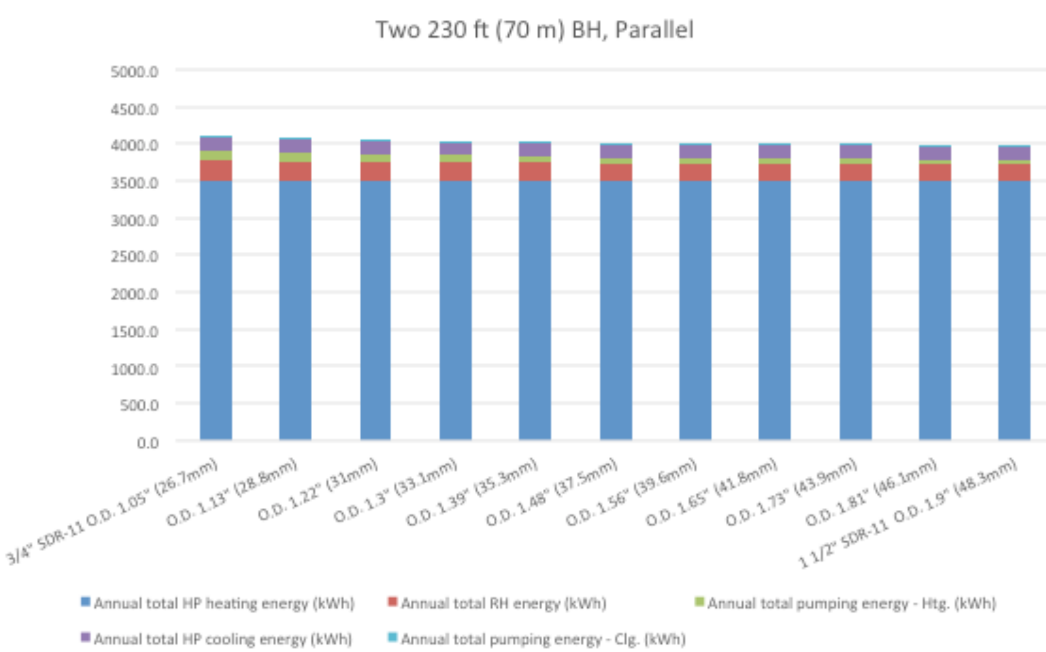

Figure 5 Electricity Use for 230 ft (70 m) Boreholes in Parallel. DR-11 Pipes ¾" to 1 ½" (O.D. 26.7-48.3 mm).

One figure of merit for the system is the seasonal or annual system coefficient of performance for heating. We refer to this as SCOP and it includes pumping energy in heating mode, resistance heating energy, and heat pump energy in heating mode. Figure 6 shows heating SCOP and head loss at 32°F (0°C) for the three borehole configurations at the range of pipe sizes between ¾" SDR-11 and 1 ½" SDR-11. It is clear that from a system performance point of view, deeper boreholes in series and larger pipe sizes are the better choices. Furthermore, the best system performance is only obtained for cases where there is a substantial number of hours with laminar flow. Although more research covering more cases can always be done, we caution the reader against choosing pipe dimensions and flow rates solely on the basis that the flow always be turbulent at all temperatures.

An alternative criterion given by Mescher (ASHRAE 2011) to meet the "A" grade for pumping suggested by Kavanaugh and Rafferty (1997) is that the total system head loss should be no larger than 50 ft (15.2 m) and the ground heat exchanger head loss should be no larger than 25 ft (7.6 m). Figure 6 (right side) shows ground heat exchanger head losses, including some horizontal connecting piping, for the test cases. The red-dashed lines shows Mescher's criterion. For all cases, pipe sizes 1" SDR-11 or larger keep the ground heat exchanger head loss below the limit. However, further improvements in the seasonal SCOP of 3-4% are possible for the series cases by increasing the pipe size to 1 ¼". This increase in pipe sizes brings the head loss down to well below Mescher's criterion, so with further research, it may be desirable to decrease the suggested maximum value. Mescher was discussing commercial and institutional buildings and we are looking at a single-family house but nevertheless, the criterion gives generally good results.

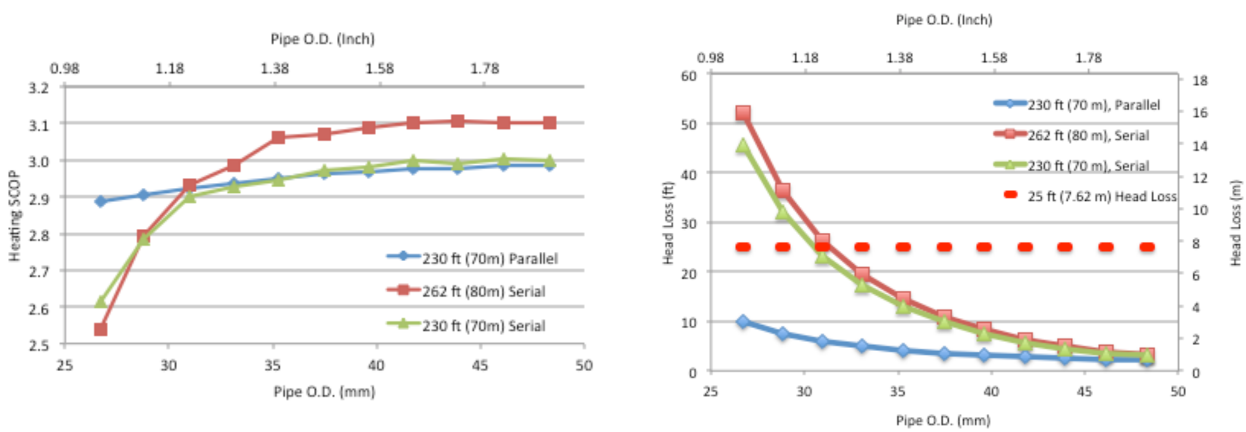

Figure 6 Heating SCOP (left) and ground heat exchanger head loss at 32°F (0°C) (right)

CONCLUSION

This paper has presented an investigation of the effect of design flow velocities on GSHP system performance for a house in South Dakota - a heating dominated area. The results suggest that keeping design pressure loss relatively low, as suggested by Mescher (2011) is more important than keeping the flow turbulent for any specific portion of the year. In fact, our simulations showed increased system performance as measured by seasonal system coefficient of performance as the number of hours with turbulent flow decreased. The systems that performed best had significant numbers of hours during the heating season with laminar flow.

More specifically, for this case with two boreholes 230-262 ft (70-80m) deep with a design flow rate of 6 GPM (0.38 l/s), the results suggest that from a system performance point of view, standard pipe sizes of 1" SDR-11 or larger be used to find an optimal balance between pumping energy and increased heat transfer as the Reynolds number varies with temperature over the year. For cases where the borehole is connected in series, using a pipe size of 1 ¼" SDR-11 instead of 1" SDR-11 gives a few percent increase in seasonal SCOP. The parallel configuration gives better performance at small pipe diameters and similar overall performance at larger pipe diameters to the serial configuration even though it operates for many more hours in laminar flow.

The results are consistent with the conclusions of Kavanaugh (2011) that high fluid velocities result in high pumping power with little thermal and even less economic advantage over non-laminar flow, and with the criterion given by Mescher that the borehole field head loss be kept to less than 25 ft (7.6 m) (ASHRAE 2011).

One aspect not included in the calculations is the dissipation of the pump energy in the fluid. Particularly for the cases with high pumping energy, the dissipated pumping energy will raise the borehole temperatures, presumably offsetting some of the resistance heating and, in that way, some pumping energy will be recaptured. Another aspect not included is the variation in flow rates over the year due to the system curve shifting with changes in viscosity due to changes in temperature. It's been our experience in a previous study that this effect is fairly small, so we expect it to have negligible effects on the results here.

This study had been limited to one type of geological and climatic conditions, in North America. It would be of interest to extend this study to include GSHP systems in other heating dominated climates, with other heat pump system designs and ground heat exchanger designs, e.g. water-to-water heat pumps with and without integrated domestic hot water production, and groundwater filled boreholes, as common in the Scandinavian countries. This would require an improved model of the ground heat exchanger to account for variations in borehole resistance due to convection in the borehole water, and an improved model of the heat pump that includes dynamic behavior of the domestic hot water production.

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