INNO-U-33

A PARAMETRIC SENSITIVITY STUDY INTO BOREHOLE PERFORMANCE DESIGN PARAMETERS

Henk Witte

Groenholland Geo-Energysystems, Valschermkade 26, 1059CD Amsterdam, Netherlands,
Phone: 31-20-6159050, e-mail: henk.witte@groenholland.nl

1. Introduction

To use the ground as a source for the heat pump with closed loop systems, a ground heat exchanger is designed and constructed. The purpose of the ground heat exchanger is to exchange heat with the surrounding ground, where the flow of heat is driven by temperature differences. One major factor affecting the performance of the heat pump is the difference between evaporator and condenser temperature. The relationship between the evaporator and condenser temperatures and the Coefficient of Performance (COP, the thermal output divided by the electrical input) is shown in figure 1. If the evaporating temperature is decreased by one Kelvin, or condensing temperature is increased by one Kelvin, on average the thermal performance of the heat pump will drop by about 3%. Controlling the operational temperatures is therefore important to achieve high performance rates.

Partly these temperatures depend on the characteristics of the system and properties of the ground reservoir. On the building side the heating and cooling temperatures selected are usually under the designers control and sophisticated control strategies can be implemented to both reduce the total capacity needed as well as the operational efficiency. Examples of such strategies are the use of temperature compensated set points, variable flow systems with control on return temperature and the use of hybrid systems with additional capacity provided by conventional boilers or chillers. On the other hand, the ground properties cannot be controlled but can be investigated using geological, geo-hydrological and geo-physical methods, amongst which the Thermal Response Test (Austin 1998, Gehlin 1998, van Gelder et al 1999, Witte et al 2002).

The second important factor in the overall performance of the ground source system is the pressure loss, which equates to pumping energy required to move the carrier fluid around the heat exchanger.

In the design process, the effect of the actual ground coupling system, the Borehole Heat Exchanger, although calculated, is often overlooked in the performance optimization of a ground source heat pump system. Moreover, the thermal and hydraulic properties of the ground coupling system are normally considered only at one set of conditions, while the actual conditions in the system during operation will vary (flow conditions, temperature conditions). In the more advanced models that we are developing to allow full optimization of ground source heat pump design a dynamical consideration of these effects is important.

The effect on the performance of the borehole heat exchanger can be best understood by considering the total effective thermal resistance of this system, the relationship between the thermal resistance, thermal power and fluid temperature can be expressed as follows:

$$\Delta T = QR_{beff} \quad (1)$$

Where:

ΔT : Temperature difference between fluid and borehole wall to drive heat transfer (K).

Q : Specific heat rate (W/m).

R_{beff} : Borehole effective thermal resistance (K/(W/m)).

Using this linear relation it is not difficult to calculate the required temperature difference for a range of borehole resistances and specific heat rates. The relative impact on the expected performance can be estimated by assuming 3% loss of performance for each degree of temperature loss. Table 1 shows some typical values.

Figure 1.Coefficient of Performance (COP) as a function of condenser and evaporator entering water temperatures.

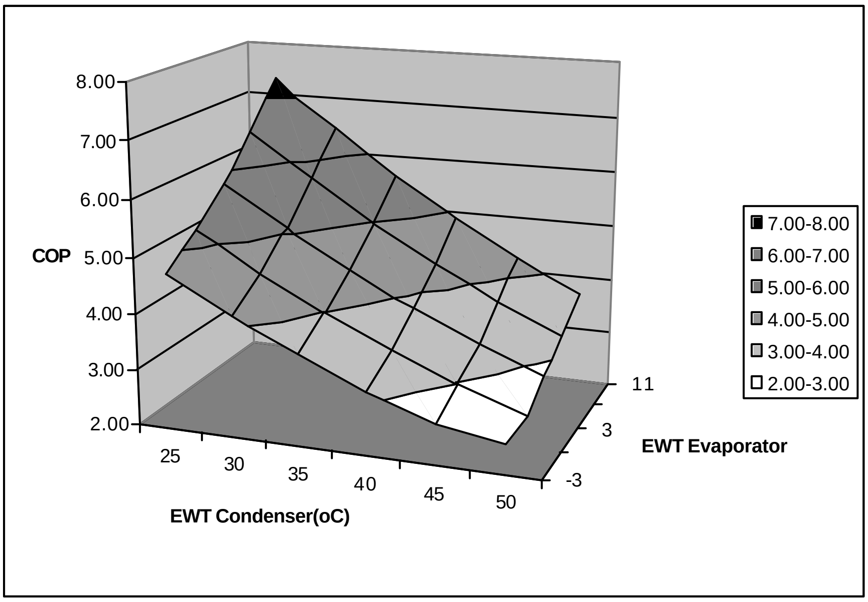


Table 1. Required temperature difference and resulting loss of performance for different heat rates and borehole resistances.

Rb (K/(W/m))	**Temperature gradient (K) for heat rate (W/m)**				**Percentage lower COP for specific heat rate (W/m)**		
	30	**40**	**50**		**30**	**40**	**50**
0.08	2.4	3.2	4.0		7.2	9.6	12.0
0.12	3.6	4.8	6.0		10.8	14.4	18.0
0.16	4.8	6.4	8.0		14.4	19.2	24.0
0.20	6.0	8.0	10.0		18.0	24.0	30.0

Especially for higher heat rates the effect of borehole resistance is important, between 10 and 25% lower COP's can be expected for a badly designed or constructed borehole. For moderate heat rates the effect of poor borehole design can still reach almost 20%. In this paper I will present the first results of a sensitivity study performed with a TRNSYS (Klein et al 1976) model component developed to calculate the borehole resistance.

2. Methods

To investigate the borehole performance sensitivity to design parameters several model components were developed that calculate the different thermal resistances and pressure loss under varying conditions. This model, developed as a set of TRNSYS (Klein et al 1976) components programmed in Fortran, was then used to calculate the sensitivity of the main performance terms to a range of realistic conditions. This sensitivity analysis was carried out by the GenOpt programme (Wetter 2004). The different model components have been validated by comparison with published data and alternative calculation methods. Due to space limitations we do not present a complete overview of the background to the model components, a full account of the component models is given in Witte (2010).

The overall performance of the ground heat exchanger can be described by two main terms: the pressure loss and thermal resistance.

The pressure loss is a function of fluid flow rate, temperature dependent fluid properties (especially viscosity) and the heat exchanger geometry and material properties (especially roughness). The pressure loss is calculated by solving Colebrook's (1939) equation with Segherides (1984) method.

The thermal resistance, is more complex but can be expressed as a single resistance term, the effective borehole resistance R_{beff} (K/(W/m)). Hëllström (1991) gives a comprehensive discussion of the different terms in the borehole resistance, his detailed equations have been implemented for both U-loop type heat exchangers and concentric heat exchangers. The main question which arises in the resistance calculations is which Nusselt calculation to use. The model uses Merker (1987) for laminar conditions (Reynolds < 2300), Dittus Boelter (McAdams, 1942) for Reynolds between 2300 and 10000 (and differentiating between heating or cooling mode). Petuhkov (1970) correlation is included for completeness for Reynolds > 10000.

The final formula for the effective borehole resistance for a U-tube heat exchanger (Hëllström 1981) is:

$$R_{beff} = R_{btot} + \left(\frac{1}{3}\right) * \left(\frac{1}{R_a}\right) * \left(\frac{L}{C_f q_f}\right)^2 \qquad 2$$

Where

R_{beff}	:	Effective borehole resistance	(W/(m/K))
R_{btot}	:	Total borehole resistance	(W/(m/K))
R_a	:	Internal borehole resistance	(W/(m/K))
L	:	Length of the borehole heat exchanger	(m)
C_f	:	Volumetric heat capacity of the fluid	(J/m^{3}K)
q_f	:	Flow rate of the fluid	(m/s)

R_{btot} is calculated as the combination of the convective fluid-to-pipe wall resistance (using the equations presented by Hëllström) and R_a, the internal borehole resistance, which is calculated by the multipole method of Bennet et al (1987).

An important difference between the standard practice, where the borehole resistance is calculated at constant flow conditions, temperatures and fluid properties, in the model component the borehole resistance can be calculated at varying conditions, this allows a better incorporation of the borehole resistance in model calculations where operating conditions vary as a function of flow rate or temperature.

3. Results

Using the model components described above, the sensitivity of the borehole resistance has been calculated for a range of typical values (table 2) to study the behaviour and relations between the main parameters. The flow rate is in fact also representative of different pipe dimensions (diameter), therefore the pipe diameter itself was not used as an additional parameter.

Table 2. Parameters and ranges of parameter values used in this study.

Parameter		Values or value range	
Carrier fluid	Type	Water @ 20 °C	30% MPG @ -5 °C
	Density (kg/m^3)	998.3	1037.7
	Heat capacity (J/kg/K)	4182	3810
	Conductivity (W/mK)	0.600	0.422
	Viscosity (mPa.s)	1.003	8.140
Total length (m)		50, 100, 150, 200	
Diameter borehole (m)		0.12, 0.17, 0.22	
Shank spacing (m)		0.05, 0.075, 0.10	
Pipe material conductivity (W/mK)		0.2, 0.4, 0.6	
Backfilling conductivity (W/mK)		0.5, 1.5, 2.5	
Soil conductivity (W/mK)		0.5, 1.5, 2.5	
Flow (m^3/hr)		0.1; 0.6; 1.1, 1.6, 2.1	

The full set of parameters requires 9720 model evaluations. To visualise the results I present, for each parameter the effective borehole resistance R_{beff} of all evaluations in a series of box plots (box-and-whisker diagram, McGill et al 1978), one for every value of the parameter.

As there is a large difference between the effective borehole resistance for laminar and turbulent conditions, and it is of interest to see how the other parameters behave under turbulent and laminar flow conditions, the box plots were constructed separately for these two cases.

Figure 2. Relation between flow rate (m/s) and effective borehole resistance (K/(W/m)) for water (left) and MPPG (right).

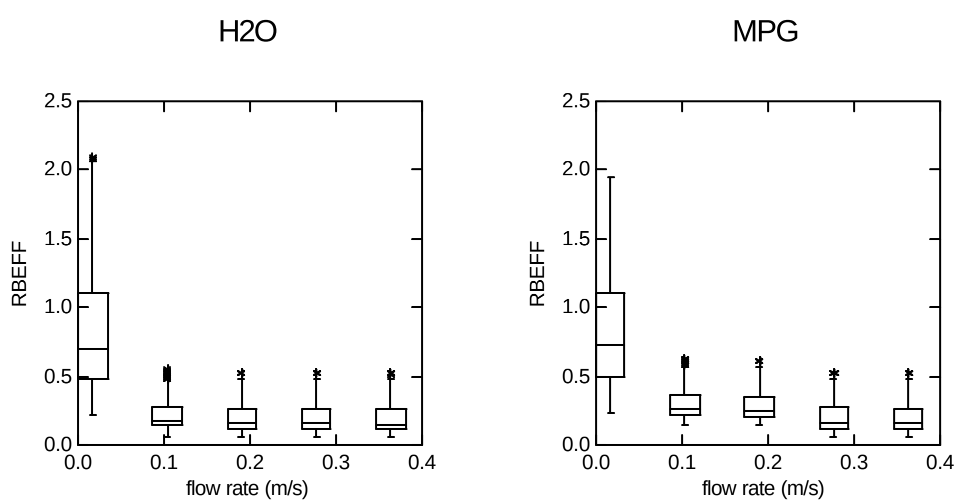
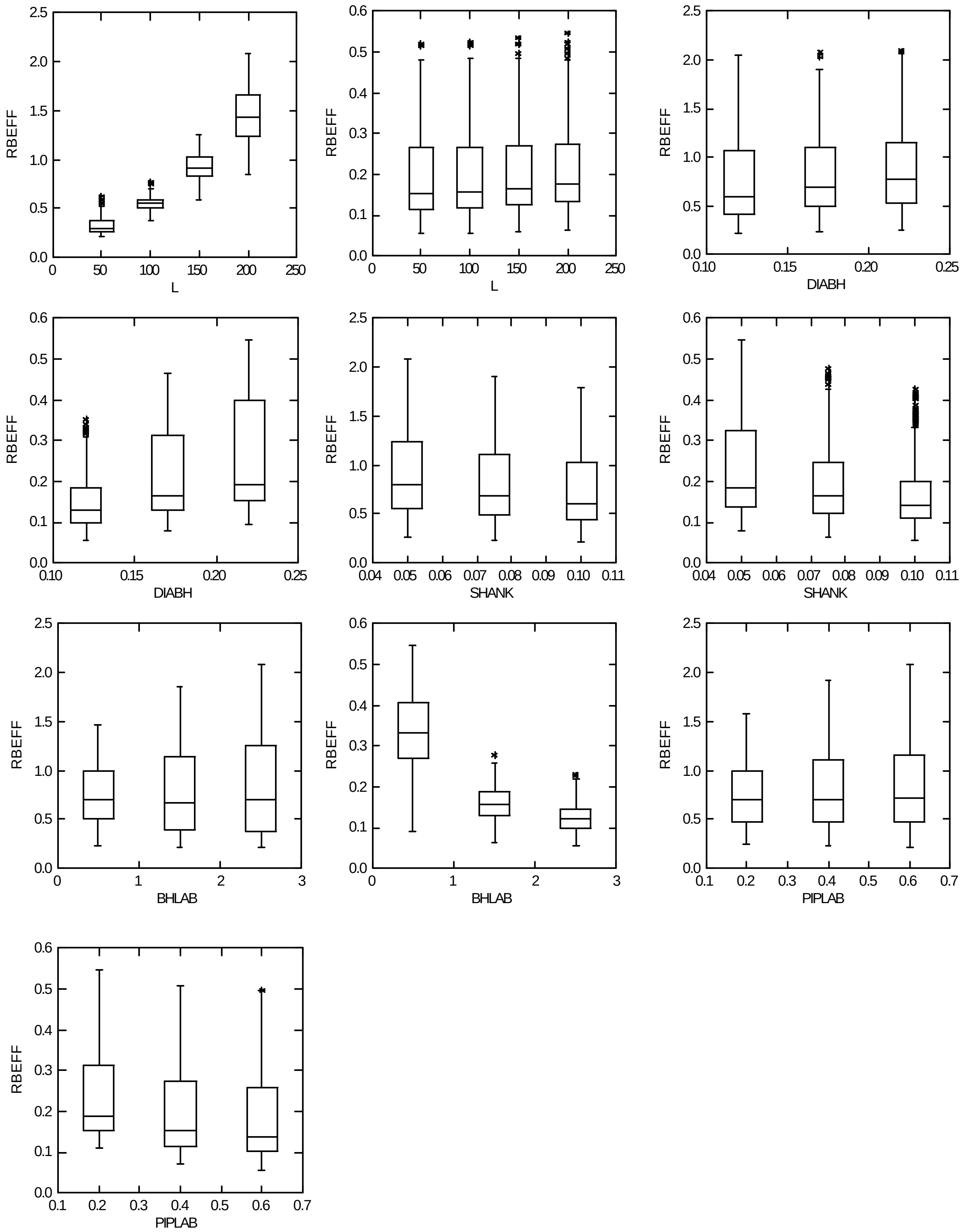


Figure 3. Box plots sensitivity study: Water, laminar (left) and turbulent flow (right). L: length borehole (m); DIABH: diameter borehole (m), SHANK: shank spacing (m), BHLAB: thermal conductivity backfilling material (W/mK), PIPLAB: .thermal conductivity backfilling material (W/mK).



The effective borehole resistances, as shown in figure 2 for the different flow rates, vary between 0.056 upto values exceeding 2.0. Typical values for a commercial borehole heat exchanger system would be expected to have a range not exceeding 0.05 and 0.5, for the sensitivity study a wider range has been allowed.

Although there are some differences between the calculations using water and monopropylene glycol, the main trends are the same. In view of the available space only the figures of water are shown (figure 3). The main results, the effective borehole resistance as a function of parameter values, are presented. For the different parameters the cases with turbulent and laminar flow are plotted separately. We may observe that the sensitivity to the parameters is bigger for the calculations with MPG that H2O, especially for the laminar flow conditions. This is partly attributable to the fact that there are more cases for MPG in the laminar flow regime.

The first parameter presented is the relation between the length of the borehole heat exchanger and the effective borehole resistance. For the laminar flow regime there is a very significant relationship between effective borehole resistance and length, the longer the BHE the higher the thermal resistance. This effect is bigger for the cases with water as working fluid.

The main reason for this is the fact that the effective resistance equation (eq. 2) multiplies the internal resistance R_a by a fraction including length in the numerator and flow rate in the denominator. The internal resistance is larger for low Reynolds numbers and it is multiplied with a large fraction (low flow rate). The flow rates for MPG where conditions are still laminar are much higher and thereby this effect is reduced for the cases with MPG as circulation fluid

The diameter of the borehole has a relatively small effect with laminar flow conditions, with turbulent flow conditions the effect is larger. Moreover the effect of borehole diameter clearly depends on the thermal conductivity of the backfill material, with low thermal conductivity the effect of borehole diameter is much bigger.

An increase in shank spacing decreases the effective borehole resistance. The shank spacing decreases the internal heat exchange in the borehole heat exchanger and thereby improves the performance. This effect is more pronounced for the turbulent flow regime. An important correlation can be found also between shank spacing and thermal conductivity of the borehole backfill material(figure 6). With a high thermal conductivity filling the effect of shank spacing is almost zero, while for a low conductivity backfilling the shank spacing has a very important effect.

Although the wall thickness of the pipes is not very big (about 3 – 4 mm typically) there still is a clear effect of the thermal conductivity of the pipe material. Especially for turbulent flow conditions the effective borehole resistance decreases with increasing thermal conductivity.

The conductivity of the borehole material (figure 5 and 6) can affect the sensitivity of other parameters, but also has an important effect by itself. The effective borehole resistance decreases markedly with increasing conductivity of the borehole backfilling material. Only for the laminar flow regime with water as medium there is no clear effect.

The flow rate and viscosity (depending on fluid type and temperature) have the largest overall effect on the effective borehole resistance. Especially the flow rate is, within limits, under the designers control as the number of boreholes and heat exchanger diameter can be selected by the designer. Even the flow required by the heat pump can be varied somewhat. Usually the main distinction is made between laminar (Reynolds < 2300) and turbulent (Reynolds > 2300), however from the parametric study it is clear that very low flow rates should be avoided especially for longer heat exchangers. Moreover, increasing the flow rate also increases the pressure loss, which equates to pumping power. Figure 4 shows the relation between flow rate and pressure loss. From this graph it is evident that the pressure loss increases rapidly with increasing flow rate. In general a flow rate between 0.2 and 0.3 m/s will give a reasonably low thermal resistance with a pressure loss that is not too great.

The soil thermal conductivity has no discernable effect on the borehole resistance.

A ranking of the effects of the different parameters, for the laminar and turbulent flow regime, is given in table 3. When one considers the absolute change in borehole resistance, apart from the flow regime (turbulent / laminar) the main effect is length of the borehole heat exchanger for the laminar flow regime. For the turbulent flow regime the largest change of effective borehole resistance is found for the borehole backfilling thermal conductivity. When the fractional changes are considered, which is the change in borehole resistance divided by the change in parameter, the picture is different. The largest effect is then found for the shank spacing followed by the borehole diameter for both the laminar and turbulent flow regimes.

So far we have considered the effects of the individual parameters. There are also parameters that are coupled, especially borehole diameter, borehole backfilling thermal conductivity and shank spacing. In figure 4 the effect of borehole diameter is shown for low and high thermal conductivity backfill, figure 5 shows the effect of shank spacing. Although there are differences between the turbulent and laminar cases and also between the cases with water or monopropylene glycol, the overall behaviour is the same and these cases are not shown separately. With a low borehole conductivity there is a clear effect of borehole diameter (increasing borehole resistance with increasing diameter) and shank spacing (lower borehole resistance with increasing spacing).

Table 3. Ranking of effects of different borehole construction parameters on effective borehole resistance for laminar and turbulent flow regime. Given are the absolute change in the median effective borehole resistance over the parameter range and the fractional change (change of Rbeff divided by the change of parameter).

Laminar flow regime			**Turbulent flow regime**		
Parameter	Absolute	Fractional	Parameter	Absolute	Fractional
Length	1.00	0.01	Length	0.01	0.00
Borehole diameter	0.15	1.50	Borehole diameter	0.08	0.80
Shank spacing	-0.20	-4.00	Shank spacing	-0.07	-1.40
Pipe therm. cond.	-0.05	-0.13	Pipe therm. cond.	-0.08	-0.20
Borehole thermal cond.	-0.22	-0.11	Borehole thermal cond.	-0.21	-0.11
Soil thermal cond.	0.10	0.05	Soil thermal cond.	0.00	0.00

Figure 4. Pressure loss (kPa) versus flow rate (m/s) for the different cases in the sensitivity study.

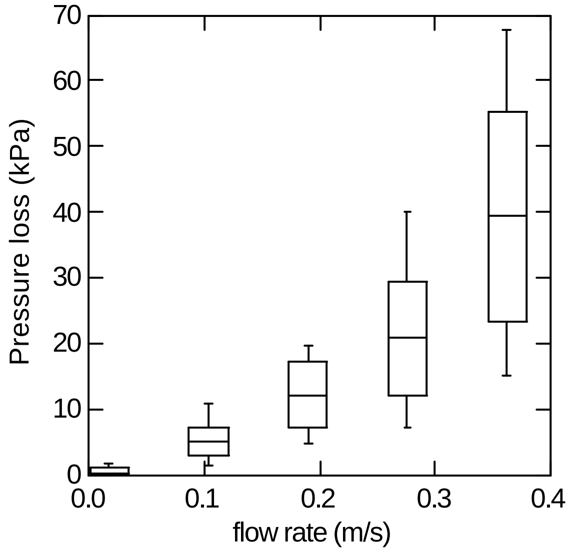


Figure 5. Effective borehole resistance for different borehole diameters. Left: a low borehole conductivity (0.5 W/mK), right: a high borehole conductivity (2.5 W/mK).

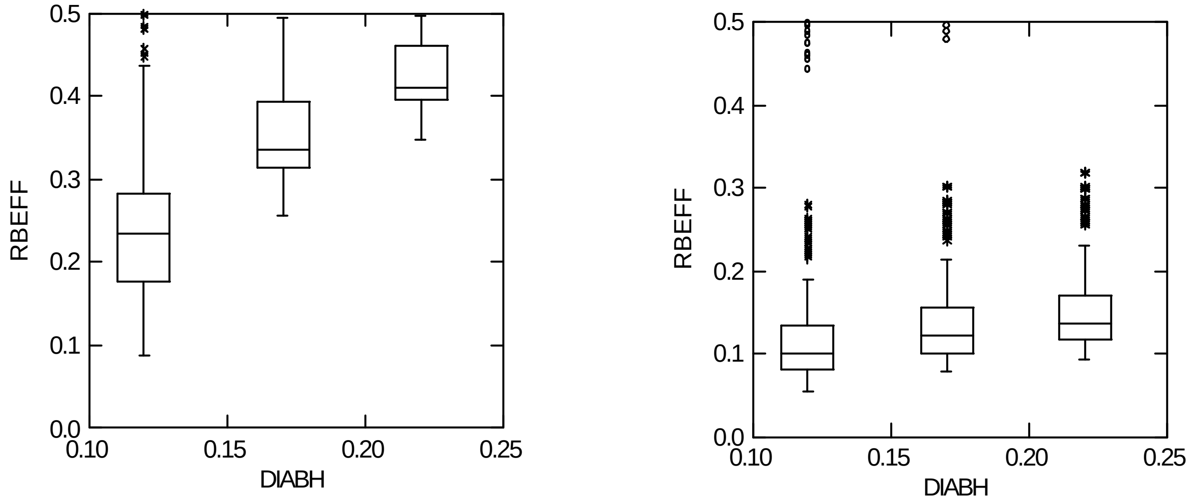
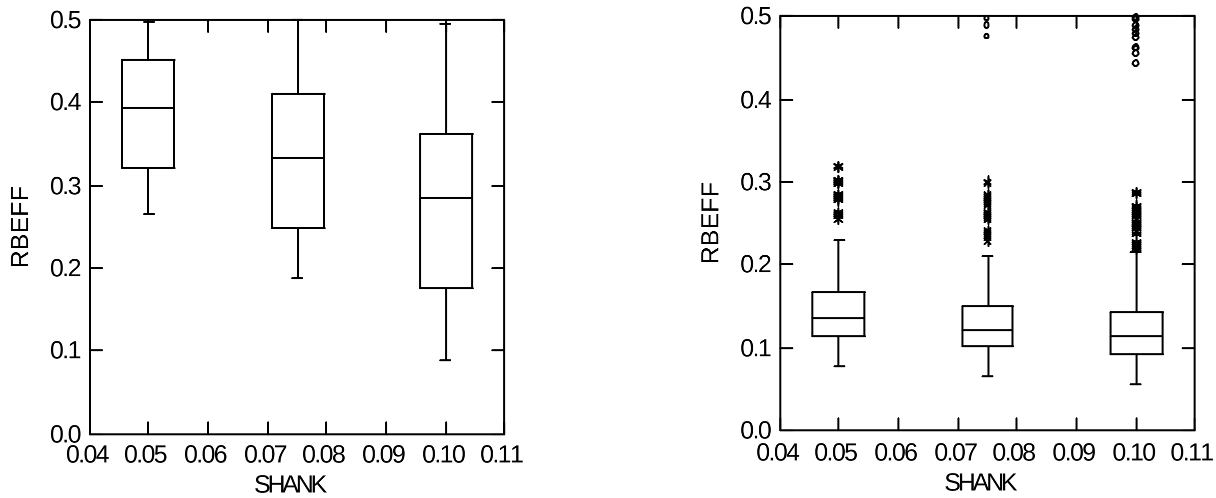


Figure 6. Effective borehole resistance for different shank spacing. Left: a low borehole conductivity (0.5 W/mK), right: a high borehole conductivity (2.5 W/mK).



With higher borehole conductivity values there is only a small effect noticeable. With a low borehole backfilling conductivity the effective borehole resistance almost doubles with a larger borehole diameter, bigger shank spacing reduces the borehole resistance with 25%.

4. Conclusions

Borehole resistance is an important parameter in borehole heat exchanger design. The temperature difference needed to drive the heat transport between the ground and the fluid depends mainly on the borehole resistance. For borehole heat exchanger systems with lower specific heat extraction rates (mainly used for heating application), between 15 and 25 W/m, the effect on the performance of the heat pump is around 10% lower performance for a borehole with high thermal resistance. For systems with higher specific heat rates (used for cooling, or in high conductivity ground) the effect may be much bigger, up to 20% difference between the low resistance and high resistance borehole. In this parametric study the lowest resistance was slightly below 0.06 K/(W/m). These cases had in common a small borehole diameter, average shank spacing, high backfilling conductivity and turbulent flow conditions. The high thermal resistances (> 0.5 K/(W/m)) were found for very long boreholes (200 meters) with laminar flow (Reynolds < 1200), large borehole diameter, small shank spacing and low borehole conductivity values.

When designing a borehole heat exchanger the usual design criterion is the fluid temperature, for instance after 25 years. With this criterion the average or minimum performance of the system is fixed. However, the temperature change due to the borehole resistance is always needed and affects the performance during the complete operational life of the system. The designer therefore needs to carefully consider the different options with regard to the borehole construction.

From the information presented we can define some more general guidelines on borehole heat exchanger design, in order of importance:

1. Design flow rate should be between 0.2 and 0.3 m/s
2. For systems with laminar flow the flow rate should not be lower than 0.1 m/s
    a. The length of borehole with laminar flow should be limited
3. Large diameter low conductivity boreholes should be avoided.
4. The conductivity of the borehole backfilling material should be as high as possible, irrespective of flow regime.
5. The shank spacing should be as large as possible if the flow regime is laminar, for turbulent flow regime this is not so important.
6. The borehole diameter should be as small as possible, again mainly for the laminar flow regime.

Often shank spacers and double U-loop heat exchangers are proposed to enhance the thermal performance of a borehole heat exchanger. Given the cost and additional time needed to install these options it is important to note that, from the thermal performance point of view, this is useful only in situations where a relatively high specific heat rate is combined with a low conductivity backfilling.

Although the results presented here could be derived directly from the equations of borehole resistance as developed by Hëllsttröm (1981), it is not easy to obtain a quantitative assessment of the different parameters. The parametric sensitivity study presented here does give this quantitative assessment. In this first attempt we have selected parameters to be representative of the parameter values that can be expected in practice. To obtain both laminar and turbulent flow regime for the cases with water and monopropylene glycol as heat transfer fluid, a wide range of flow conditions was needed. Still, for the systems with water only the lowest flow rates had laminar flow conditions while for the cases with MPG only the highest flow rates achieved turbulent flow conditions. Although this allowed including the complete range of expected conditions, it also led to somewhat unrealistic combinations that resulted in very high borehole resistance values. It would therefore be interesting to repeat this analysis with a more restricted range of conditions. On the other hand, it would be interesting to include other types of heat exchangers (double U-loop, concentric heat exchangers) in the analysis.

The TRSNSYS types developed to carry out the calculations are useful for calculations where the borehole resistance changes as a function of operating conditions (heating in winter and cooling in summer, for instance) as the borehole resistance may change significantly.

5. References

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6. Acknowledgements

This study was supported under the FP7 programme "Advanced ground source heat pumps systems for heating and cooling in the Meditteranean climate" (GROUND-MED), contract number 218895.