Annex A. Model construction under GLD 2016 environment

In this document we show the details concerning the inputs made in GLD software in order to correctly model our GSHP system. It is especially useful for people wanting to exploit the same capabilities of the software but it represents a too much detailed set of information that could deviate the attention from the main focus of the presented work.

Ground and borehole field characteristics

We must first select the Design Method as "Fixed length", since we consider an existing facility. Normally, when a project is started, it is the "Fixed Temperature" mode the preferred option in order to estimate the length of the borehole field (see Figure 1).

Step 1: Definition of fluid characteristics and flow rate.

In our case, we must consider pure water as the circulating brine. It is recommended to check the "Automatic entry mode" and work with the thermophysical parameter values calculated by GLD. **Otherwise, the simulation results may differ from different runs.**
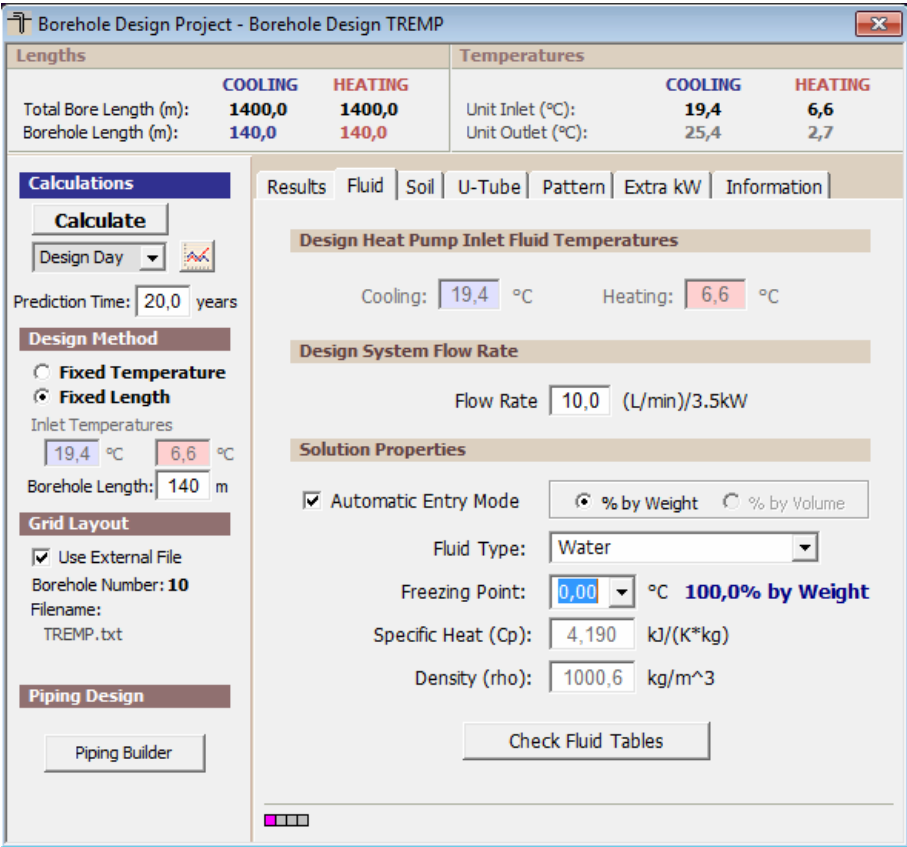

Figure 1. Snapshot of the borehole design module showing the "Fluid" panel.

Step 2: Design system flow rate.

This value provides the necessary flow rate (per borehole) in order to exchange 3.5kW of power with a certain temperature gradient (assuming total heat transfer between the brine and the load). This indicator was conceived originally within the British unit system as the necessary heat flow rate in order to melt 1 ton of ice in 1 day (12000 Btu/h, which is equivalent to 3.5kW in the metric system). The required brine flow can be determined from the following expression:

$$\dot{V} = \frac{\dot{Q}}{c_p \rho \Delta T}$$

Where $\dot{Q}$=3.5 kW; c_p = 4184J/kgK; and ρ = 999.6kg/m³ in our case. The value of the temperature gradient ΔT is extracted from the temperature difference between the inlet and outlet of the evaporator at the heat pump, which can be closely approximated to the temperature difference between the inlet and outlet brine flow towards/from the ground when both compressors are ON. In Figure 2 it is shown the time evolution of the storage vessel, the brine inlet and brine outlet, during a heat production period occurred the 28th of March of 2017. The difference is ~5ºC, so we take this value for our calculation.
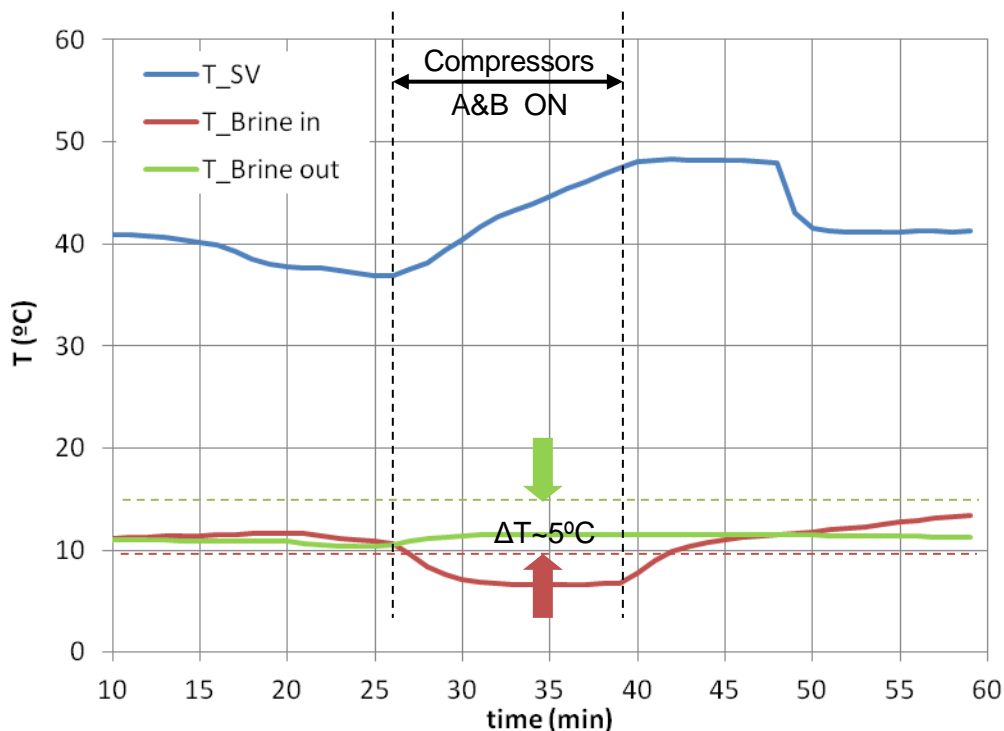

Figure 2. Temperature evolution of the storage vessel (T_SV), inlet and outlet flows of the brine towards/from the ground (T_Brine in and T_Brineout, respectively) during heat production, the 28th of March of 2017. The difference between T_Brine in and T_Brine out is ~5ºC.

Introducing all the data into the above expression, we obtain a flow rate value of 10.04 l/min/3.5 kW. Although we already have data about the actual flow rate, this cannot be input into the model because as mentioned above, the software is oriented towards design, not modelling. In fact, the flow rate expressed in units of l/min/3.5 kW provides an intensive variable that allows calculating the specific flow rate (in l/min) according to the actual thermal load. However, through this methodology, we will be able to check how close is the "design" flow rate w.r.t. to the actual one (16.7 l/min during the cool season, and 9.4 l/min during the warm season)

Step 3: Soil characteristics

Here we just input the data we know from the TRT report about the soil characteristics:

- $T_0 = 14.9\ ^{\circ}C$
- $\lambda = 2.1\ W/mK$
- $\alpha = 0.0434\ m^2/day$).

We also consider a prediction period of 20 years (Figure 3).

Figure 3. Snapshot of the borehole design module showing the "Soil" panel.

Step 4: U-Tube characteristics

This section is relevant for the evaluation of the borehole thermal resistance R_b (Figure 4), which is evaluated following the procedure described in [1], through the following expression:

$$R_b = \frac{R_f + R_t}{2} + R_{grt} = \frac{\frac{1}{\pi D_i h_i} + \frac{1}{2\pi k_t} \ln\left(\frac{D_0}{D_i}\right)}{2} + \frac{1}{k_{grt}\beta_0 \left(\frac{D_b}{D_0}\right)^{\beta_1}} \qquad (Eq.\ 1)$$

Where:

- R_f is the resistance of the fluid interface with the tube wall.
- R_t is the thermal resistance of the tube.
- R_{grt} is the thermal resistance of the grout defined by the space between the tubes and the bore walls filled with thermo-concrete.
- D_b is the diameter of the borehole
- D_0 and D_i are the outer and inner diameter of the tube, respectively
- h_i is the heat transfer coefficient from the fluid to the inner wall of the tube (considering forced convection), which depends on the fluid characteristics and the Reynolds number (the higher the Reynolds number, the higher the h_i)
- k_t and k_{grt} are the thermal conductivity of the tube and the grout material
- β_0 and β_1 are shape factors (non-dimensional) that account for the location of the tubes w.r.t. to the borehole walls.

In our case, for the estimation of the borehole resistance, we make the following choices within GLD:

- *Pipe size* (D_0): 1 ¼"

NOTE:We cannot establish manually the dimensions of the pipe. For this reason we must choose the closest one from the available list, which corresponds to 1 ¼ in size.

- *Pipe type:* SDR11
- *Flow type*: Laminar

NOTE: We make this choice taking into account the resulting Reynolds number of the fluid according to the flow (10m³/h max) and pipe characteristics (DN40, SDR11) (see Table 1).

T (ºC)	T (K)	Density (kg/m3)	Cp (J/kgK)	Thermal cond. (W/mK)	Dynamic viscosity (Pa·s)	Prandt number	Tfreeze (ºC)	Reynolds number	**Flow type**
0	273,15	1003,51	4212,43	0,55	0,20	1524,12		54,26	**laminar**
5	278,15	1002,67	4205,90	0,56	0,22	1628,52		49,61	**laminar**
10	283,15	1001,71	4200,08	0,58	0,24	1741,15		45,41	**laminar**
15	**288,15**	1000,61	4194,97	0,59	0,26	1862,42	273,15	**41,60**	**laminar**
20	293,15	999,37	4190,57	0,60	0,28	1992,78		38,16	**laminar**
25	298,15	998,01	4186,88	0,60	0,31	2132,71		35,04	**laminar**
30	303,15	996,51	4183,91	0,61	0,33	2282,77		32,21	**laminar**

Table 1. Thermophysical properties of water.

- *U-tube configuration:* Single
- *Radial pipe displacement:* Average

NOTE: The radial pipe displacement is chosen arbitrarily as "average", since we do not have data available on this.

- *Borehole diameter (D_b):* 130 mm
- *Thermal conductivity of the backfill (k_{grt}):* 1.5 W/mK

NOTE: The value chosen for the thermal conductivity is somehow arbitrary, but at the same time it is a moderate and common value (GLD software database offers a list of different backfill materials and their corresponding thermal conductivity), and is in agreement with the information of the project, which states that $k_{grt} > 1W/mK$.

Figure 4. Snapshot of the borehole design module showing the "U-tube" panel.

Step 5: Define borehole pattern

GLD software allows to define simple arrays by introducing the number of rows and columns, the space between boreholes and borehole depth. Moreover, if more complex geometries are used, the user can define their own patterns by means of a .txt file where the coordinates of each borehole are input (Figure 5), which is actually our case. The software also allows visualizing the relative temperature variation of the borehole walls according to their geometry, characteristics and relative location.
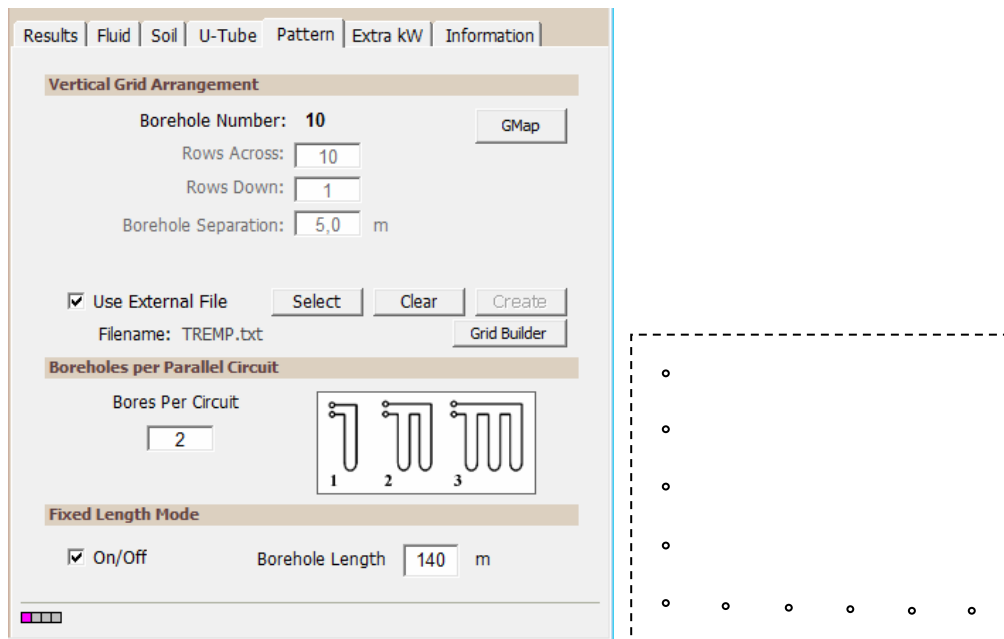
Figure 5. Snapshot of the borehole design module showing the "Pattern" panel (left). Borehole layout of the project. It consists of 10 boreholes arranged as shown in the figure, with a separation of 5m between boreholes (right).

Step 6: Define extra power consumption

Figure 6 shows the panel where the user can input extra power consumption attributed to circulation pumps, cooling tower, fans, or electric consumption of the heat pump itself. In our case, we only consider the extra power corresponding to the circulation pumps, since the data we have on electrical consumption account just for heat pump compressors and circulation pumps, indeed. This way, we will be comparing the same thing between the actual system and the model. The value we input here is taken from actual measurements. It does not really matter the real mechanical power and the electrical efficiency, as long as the final consumption value agrees with our experimental data. So, if we consider a required mechanical power of 522 W, and an electrical efficiency of $\eta_{e-m} = 30\%$, the resulting electrical power consumption is 1740 W, which corresponds with our measurements.

It is important to remark the fact that our particular heat pump comprises two twin modules with their corresponding compressors, and depending on whether the heat pump is operative with one or two of them, the circulation pump power is different. However, it is not possible to implement this kind of distinction in our model, because (as it will be explained later), the heat pump must be defined within GLD as a single module corresponding to its full capacity (60 kW, nominal). So its operation can only be considered as full-load or nothing, although actual heat pump allows to distinguish between 30 kW or 30+30 kW (because we have two twin modules operating in parallel). For this same reason, it is not possible to distinguish between two levels of circulation power consumption. In the end, this represents a source of inaccuracy in our model. Finally, there is the opportunity to input extra power consumption,

for example, corresponding to the stand-by mode. However we prefer not to input such value, since our measured values of $SCOP'$ and $SEER^*$ do not take it into account.

Figure 6. Snapshot of the borehole design module showing the "Extra kW" panel.

Thermal loads definition

The first to do here is to open the *Average block loads* module. Alternatively, for bigger buildings with several independent spaces and their corresponding thermal load profiles and heat pump equipments, it is recommended to use the *Zone manager* module. GLD accepts data regarding thermal loads with monthly or even hourly resolution. To use one option or the other only affects the temperature evolution modelling throughout the project lifetime (according to the time domain chosen hourly, daily or monthly design). Besides, performance simulation of the GSHP only requires the information contained in the table shown in Figure 7, which is automatically generated by the software, regardless of the resolution used to define the thermal losses or gains. Actually, to fill-up these cells manually is enough to run the simulation.

From the thermal load data available (see §4.1.1.2 from the main document, Table 8), GLD constructs the *Design day loads* table in the following manner:

- GLD imposes the highest peak load $\dot{Q}^{peak}$ within the time period from 8 am to12 pm in heating mode (cool season), and from 12 pm to 4 pm in cooling mode (warm season).
- For the rest of the reference day, GLD attributes a heat loss or heat gain rate according to the following expression:

$$P_{load}^{rest} = \frac{E_{load}^{peak\ month} - 4h \cdot 31d \cdot P_{load}^{peak}}{20h \cdot 31d}$$

Example:

Peak month during the cool season is January, its total thermal month is $E_{load}^{peak\ cool\ month}$ = 15000 kWh and its peak heat loss is $P_{load}^{peak\ cool}$ = 30 kW. Then, GLD would attribute the following values of heat loss along a reference day like this:

- *8am-noon $\rightarrow P_{load} = P_{load}^{peak\ cool}$ = 30 kW*
- *Noon-4pm, 4pm-8pm, 8pm-8am $\rightarrow P = P_{load}^{rest\ cool} = \frac{15000kWh - 4 \cdot 31h \cdot 30kW}{20 \cdot 31h}$ = 18.2 kW*
- The annual equivalent full-load hours is evaluated as the ratio between total heat gains or losses by the corresponding peak load rate:

$$Annual\ equivalent\ full\ load\ hours = \frac{E_{load}^{total\ heat/cool\ season}}{P_{load}^{peak\ heat/cool}}$$
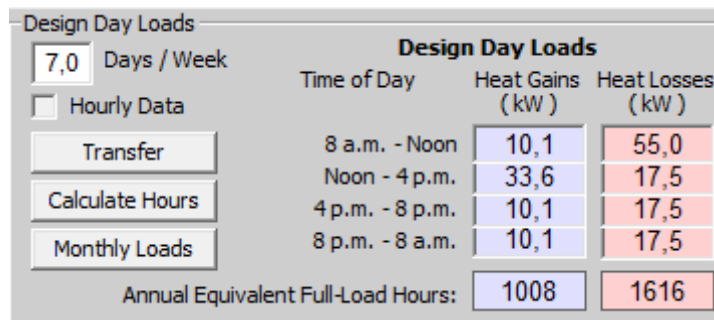
Figure 7. Average thermal load information built by GLD, based on heat gains and losses pattern within a single average day (either in warm and cool seasons), along with the yearly equivalent full-load hours.

After defining the thermal loads, we must select the desired heat pump from the GLD database (if we already added our own heat pump model data, we can select it), and define the average values for the heating fluid entering the load side of the heat pump (remember that this corresponds to $T_{evap_{in}}$ during the cool season, and $T_{cond_{in}}$ during the warm season). In our case, these values are approximated to the mean storage vessel temperatures, which are $T_{SV}^{warm\ season} = 11$ ºC and $T_{SV}^{cool\ season} = 42$ ºC (see Figure 8).
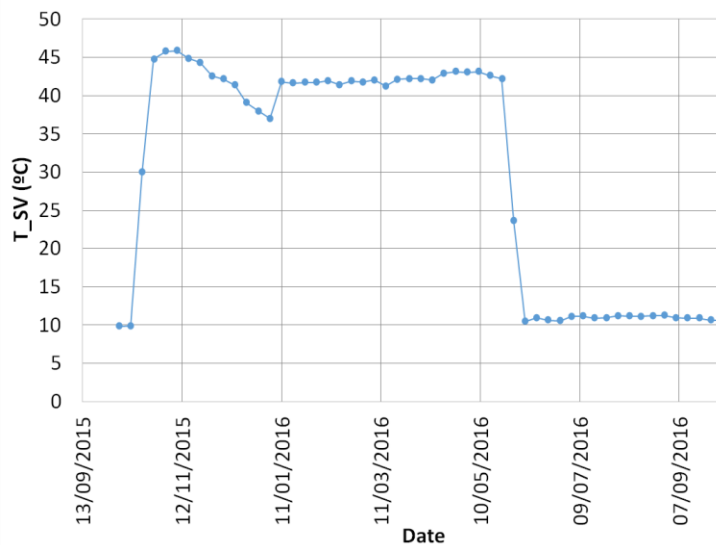

Figure 8. Temperature evolution of the heating fluid in the storage vessel (weekly averaged values).
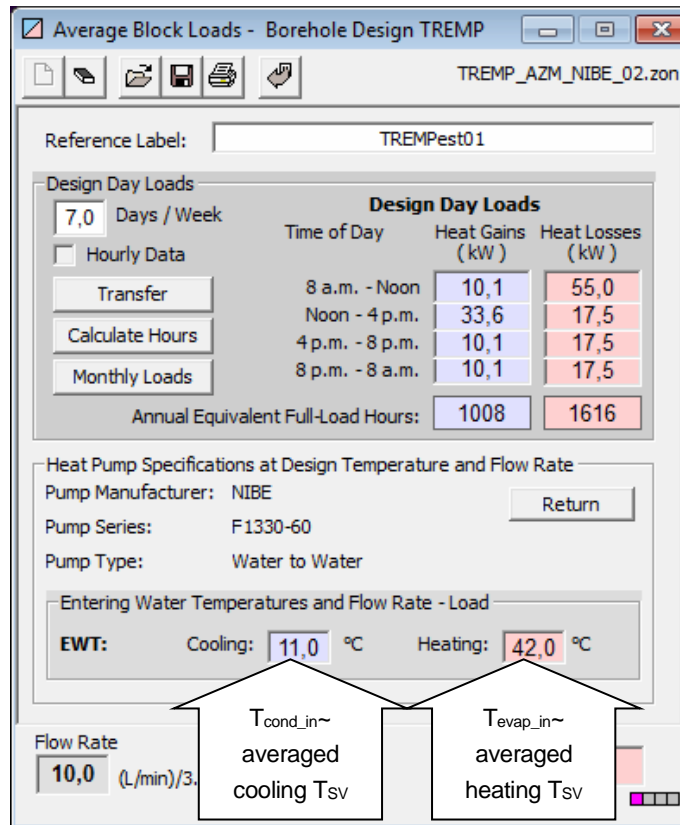
Figure 9. Snapshot of the *Average Block Loads* module, showing the data correspondint to the EWT average values (EWT= T_{evap_in} when heating and EWT=T_{cond_in} when cooling).

Heat pump addition to the data base

Since the heat pump used in the system is not within the software data base, a new entry has been input by us. For this the data available from the heat pump manufacturer has been used. In Table 2 we show the selected values from the original data that were used to built-up a new entry to the software heat pump own database. These parameters correspond to different things, depending on the season (Table 3).

T_{evap_in} (ºC)	T_{cond_in} (ºC)	Nominal capacity: 60kW			
		Heating capacity (kW)	Cooling capacity (kW)	Electric Power (kW)	COP$_{EN255}$
-5	35	52,58	39,4	13,18	3,99
-2	35	57,44	43,86	13,58	4,23
0	35	60,68	46,84	13,84	4,38
5	35	70,1	55,46	14,64	4,79
10	35	80,62	64,94	15,67	5,14
10	45	76,73	59,07	17,65	4,35
10	55	73,16	52,99	20,17	3,63
10	65	69,92	46,68	23,24	3,01

Table 2. Operating condition data from the manufacturer. Values chosen to be introduced in the *Edit/Ad Heat Pump* module are highlighted.

	Cool season (heating mode)	Warm season (cooling mode)
$T_{evap_{in}}$	$\cong T_{brine_{out}}$	$\cong T_{SV_{out}}$
$T_{cond_{in}}$	$\cong T_{SV_{out}}$	$\cong T_{brine_{out}}$

Table 3. Equivalence between inlet temperature at the evaporator and condenser with the temperature of the flow connected to it.

Step 1: Specify pump model and type of heat pump

In the general panel the most important is to check the correct type of heat pump. In our case, we have a *water-to-water* heat pump.

Step 2: Specify *source* data for cooling and heating

The data that we introduce into the *Heating* and *Cooling* panels define the interpolation curves for the heating/ cooling capacity and power consumption vs. *EWT*, that GLD builds with the quadratic structure (valid for capacity and electric power):

$$P_{cool}\left(T_{evap_{in}}^{ref}, T_{cond_{in}}\right) = P_{cool}(10ºC, T_{cond_{in}}) = a + b \cdot T_{cond_{in}} + c \cdot T_{cond_{in}}^{2}$$

$$P_{heat}\left(T_{evap_{in}}, T_{cond_{in}}^{ref}\right) = P_{heat}\left(T_{evap_{in}}, 35ºC\right) = a + b \cdot T_{evap_{in}} + c \cdot T_{evap_{in}}^{2}$$

Where $T_{evap_{in}}$ and $T_{cond_{in}}$ corresponds to *EWT* in heating and cooling, respectively. When introducing data into the *heat pump edit/add* module from GLD (Figure 10), it must be taken

into account that the values corresponding to *EWT* (Entering water temperature) are those of the brine coming from the ground (source). So we would use values of the $T_{evap_{in}}$ column for the *Heating* panel (considering a constant value of $T_{cond_{in}}$ = 10 ºC), and values from the "T$_{cond\_in}$" column for the *Cooling* panel (considering a constant value of $T_{evap_{in}}$ = 35 ºC). This can be done for two different values of flow rate, but unfortunately we do not have such information.

Figure 10. Snapshot of the *Edit/Add Heat pump* module, showing the *Cooling* and *Heating* panels. In the case of cooling mode, EWT takes values from the $T_{cond_{in}}$ column and in the case of heating; from the $T_{evap_{in}}$ column in Table 2.

Step 3: Specify *load* data for cooling and heating

This panel gathers data about the ratio between capacity and power at a reference pair $(T_{evap_{in}}, T_{cond_{in}})$ and other values when changing $T_{evap_{in}}$ (in the case of cooling) and when changing $T_{cond_{in}}$ (in the case of heating). This creates additional interpolation curves that permit a complete evaluation of the capacity and power at whichever pair of temperature conditions $(T_{evap_{in}}, T_{cond_{in}})$ for heating and cooling modes. These curves are built according to a 3rd degree polynomial expression (valid for capacity and electric power):

$$\frac{P_{cool}(T_{evap_{in}}, T_{cond\_in}^{ref})}{P_{cool}(T_{evap_{in}}^{ref}, T_{cond\_in}^{ref})} = \frac{P_{cool}(T_{evap_{in}}, 35ºC)}{P_{cool}(10ºC, 35ºC)} = a + b \cdot T_{evap_{in}} + c \cdot T_{evap_{in}}^2 + d \cdot T_{evap_{in}}^3$$

$$\frac{P_{heat}(T_{evap_{in}}^{ref}, T_{cond_{in}})}{P_{heat}(T_{evap_{in}}^{ref}, T_{cond\_in}^{ref})} = \frac{P_{heat}(10ºC, T_{cond_{in}})}{P_{heat}(10ºC, 35ºC)} = a + b \cdot T_{cond_{in}} + c \cdot T_{cond_{in}}^2 + d \cdot T_{cond_{in}}^3$$
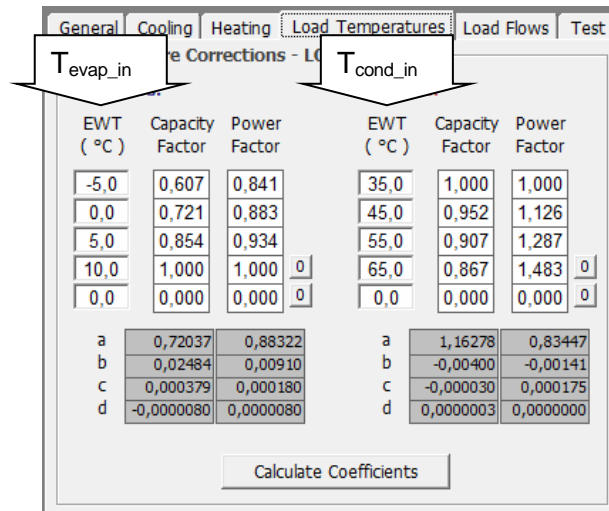
Figure 11. Snapshot of the *Edit/Add Heat pump* module, showing the *Load Temperatures* panel.

Step 4: Specify flow corrections

GLD software allows to input data regarding the variation of capacity and power with the flow rate $\dot{V}$ at the load side (Figure 12), but in our case, again there is no information from the manufacturer concerning this issue. The correction is done analogously to that of the temperature correction in the previous step (expressions valid for capacity and power):

$$\frac{P_{heat}(\dot{V})}{P_{heat}(\dot{V}_{ref})} = a + b \cdot \left(\frac{\dot{V}}{\dot{V}_{ref}}\right) + c \cdot \left(\frac{\dot{V}}{\dot{V}_{ref}}\right)^2 + d \cdot \left(\frac{\dot{V}}{\dot{V}_{ref}}\right)^3$$

$$\frac{P_{cool}(\dot{V})}{P_{cool}(\dot{V}_{ref})} = a + b \cdot \left(\frac{\dot{V}}{\dot{V}_{ref}}\right) + c \cdot \left(\frac{\dot{V}}{\dot{V}_{ref}}\right)^2 + d \cdot \left(\frac{\dot{V}}{\dot{V}_{ref}}\right)^3$$
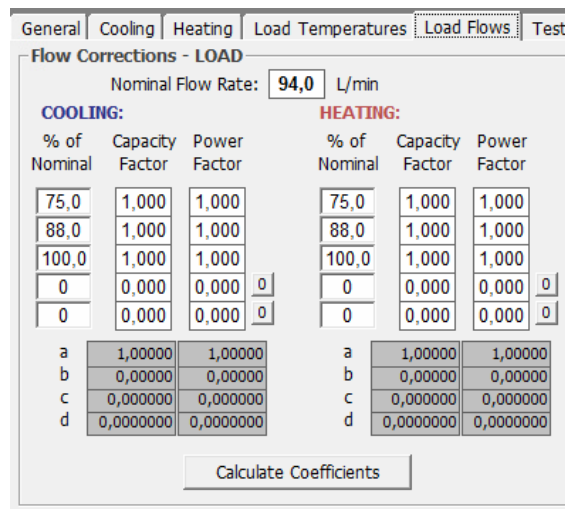

Figure 12. Snapshoot of the *Edit/Add Heat pump* module, showing the *Load Flows* panel.

The final section of *heat pump edit/add* module is a calculation panel that let us evaluate the cooling and heating capacity of the pump whichever the pair $(T_{evap_{in}}, T_{cond_{in}})$ is (Figure 13).
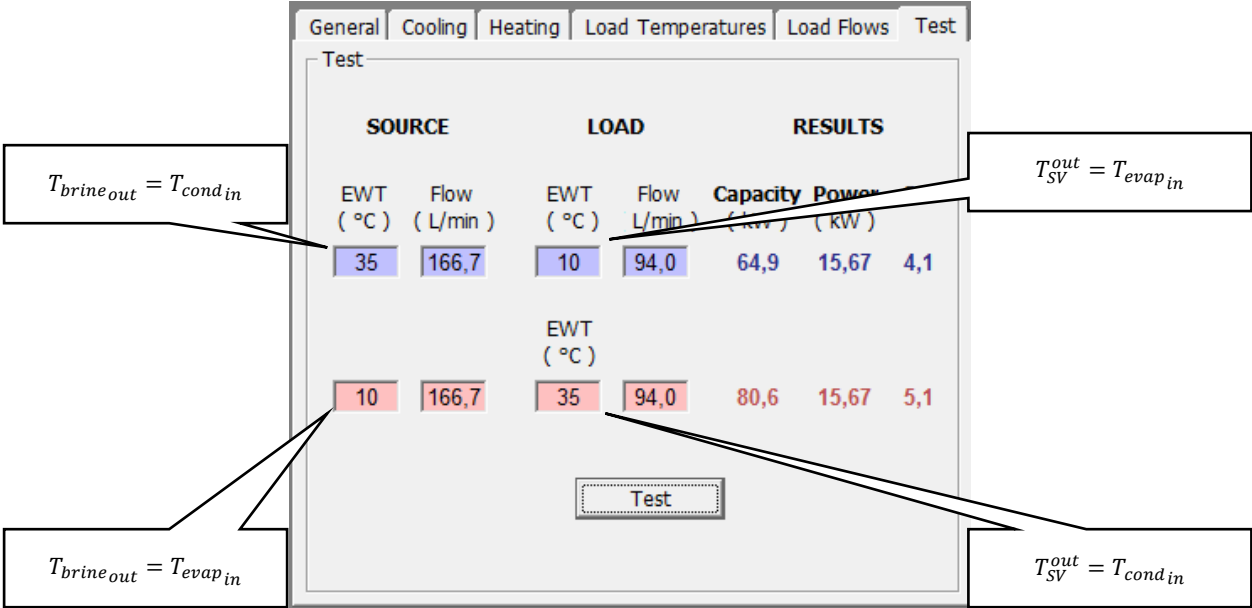

Figure 13. Snapshot of the *Edit/Add Heat pump* module, showing the *Test* panel. The shown values illustrates the correspondence between the Source/Load EWT cells with $T_{evap_{in}}$ and $T_{cond_{in}}$.

Simulation

Once all the "feeding" data has been input, it is time to run the simulation, but first, it is important to distinguish between the different choices we have. When we select *Design Day* or GLD applies a thermo-physical theoretical model where the boreholes are approximated as cylindrical heat sources (or drains). This model is more accurate when we deal with single boreholes or a small number in a simple arrangement, but it fails when it is applied to complex borehole patterns. When we select *Monthly* or *Hourly*, the thermo-physical model applied is 1-dimensional, approximating the boreholes as line sources. This model is preferred when we have complex geometries and we also have monthly data concerning heat loads, which allows a better determination of the temperature time-evolution. However, the design day calculation method also admits monthly load data if used under the *Average Block Manager* (see [2] for more information and details).
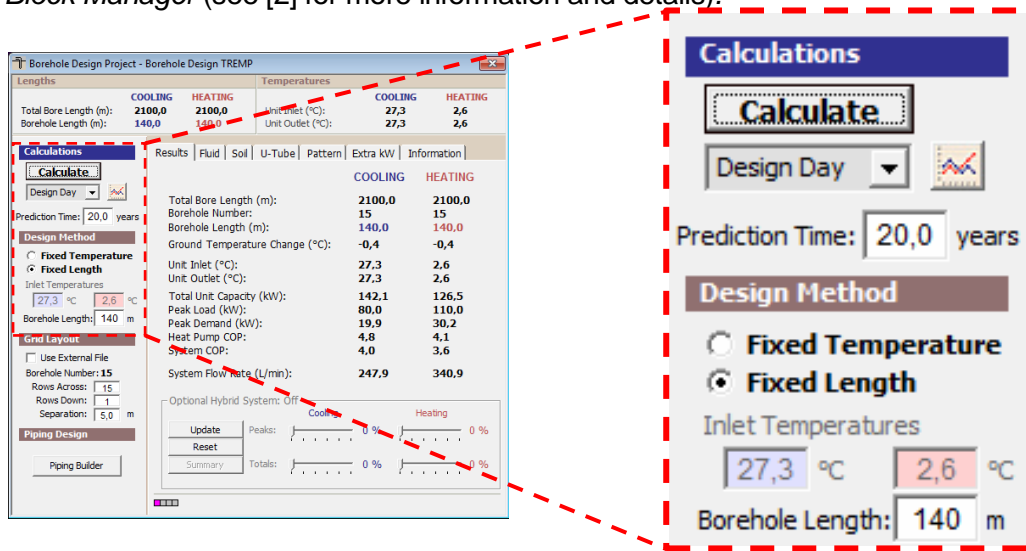

Figure 14. Snapshot of the *Results* panel. The magnified part corresponds to the section where different physical models can be chosen in order to characterize the boreholes, and also to choose the prediction time and whether we are looking for a fixed length model, or fixed temperature ($T_{brine_{in}}$ and $T_{brine_{out}}$).

		Average Block Manager (admits monthly loads)	*Zones manager* (only admits daily load design)	Borehole geometry approximation
Calculation method	Design day	OK	OK	Cylindrical
	Monthly	OK	Not possible	Line
	Hourly	OK	Not possible	Line

Table 4. Calculation methodsin GLD 2016 and its compatibilities with other options.

Bibliography

[1]. Kavanaugh S. and Rafferty K. , Geothermal heating and cooling, design of ground source heat pump systems, ASHRAE (Atlanta, 2014).

[2]. Ground Loop Design (v2016) user's guide. Available online at:
 <http://www.groundloopdesign.com/downloads/GLD_2.x/manual.pdf>