A NOVEL DESIGN TOOL FOR HEAT PUMP SYSTEMS

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ABSTRACT

Designing a competitive heat pump system for heating purposes involves thorough economic and technical considerations. In contrast to other conventional heating systems the performance of a heat pump system is strongly influenced by the temperature levels in the heat distribution system and the temperature and the characteristics of the heat source. The use of deep boreholes or horizontal ground coils as heat source that offer a relatively high and stable temperature level increases the performance in cold climates but also the investment cost. The high investment cost associated with a heat pump can only be justified if the running cost and emissions of green house gases are much less than that of a conventional heating system. This raises the need to evaluate the seasonal performance factor as well as the design of the heat source. There are a number of different tools and software available on the market today. Most of which are either looking into the design of the heat pump unit or the heat source. This paper describes a novel design tool developed in co-operation with the actors on the Swedish heat pump market. The tool has been developed in order to perform system performance analysis as well as the design of vertical boreholes.

Key Words: ***simulation, borehole design, domestic heat pump***

1 INTRODUCTION

Energy utilization in the built environment is one of the most important aspects that have to be addressed in the near future. In order to reach the targets of the Kyoto-protocol, the energy utilization in the built environment has to go through a transition. Up to now most of our space conditioning systems contribute to the global warming. Environmentally benign heating systems have to be introduced on a large scale in order to reduce the emissions of green house gases. The use of small-scale bio-fuel furnaces, heat pumps and an extensive use of district heating based on bio fuel have to be implemented. None of these techniques will on its own be able to fulfil the transition towards sustainable energy supply. All techniques will complement each other and find their markets where the conditions are most beneficial.

Domestic heat pumps are one of the most efficient ways to provide space heating and preparation of sanitary hot water. Even though technical know-how on the heat pumping technology is well proven, it has not yet reached public acceptance worldwide. In Europe, a sustainable market has only been established in small countries like Sweden, Switzerland and parts of Austria. The market for domestic heat pumps in Sweden has, during the last decade, gone through an enormous development (see Fig. 1). The total sales of domestic heat pumps reached over 66 thousand units 2004 (SVEP 2005). On top of that somewhere in between 40 000-50 000 reversible air/air heat pumps, of which only a minor part is included in the statistics compiled by the SVEP, were sold 2004. All together more than 100 thousand heat pumps were thus sold in Sweden 2004, a country consisting of approximately 1.6 million single-family houses. Due to the escalating price of oil and electricity in conjunction with the increase on energy related taxes the market for heat pumps continuous to grow at a high pace.

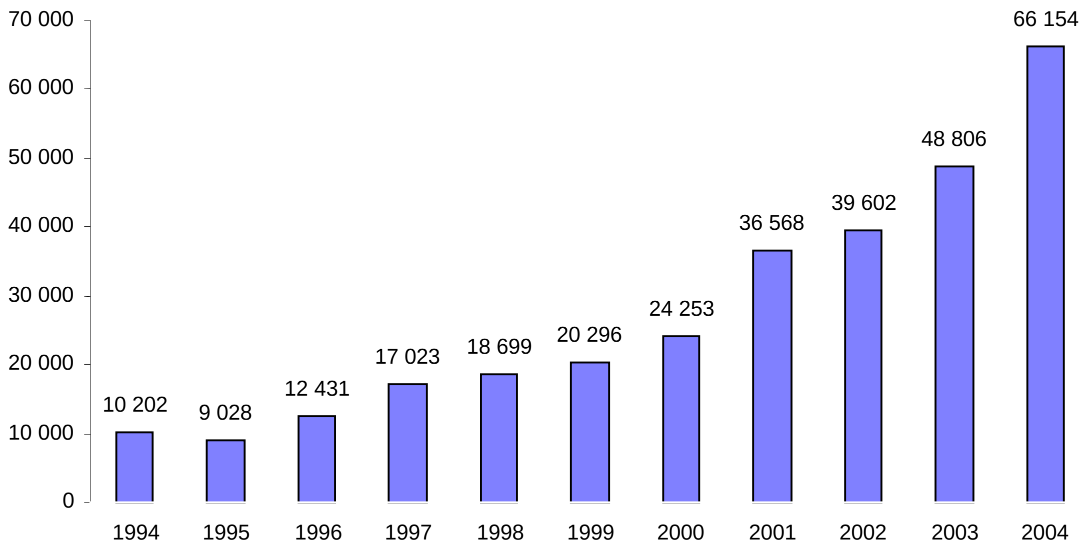

Fig. 1 Heat pump market development in Sweden 1994-2004 (Swedish Heat Pump Association 2005)

The high investment cost affiliated with ground source heat pumps (GSHP) can only be justified if the operating costs prove to be considerable lower than traditional heating systems. This raises the demand for an accurate tool for design and evaluation of different heat pump systems.

2 SIMULATION MODELS FOR HEAT PUMP SYSTEMS

There are a number of different tools available for energy performance calculation on the market. The fact that each tool is designed for its own objective and target group, lead to significant differences in terms of the structure for calculation, input, output, user interface and accuracy. A close examination of all available products three characteristic groups of tools can be distinguished. The main divergence between the groups are how the calculations are structured. Energy performance calculations are structured in static calculations, quasi-static calculations or dynamic calculations.

2.1 Static calculations

All available sales support software, provided by the Swedish manufacturers of heat pumps, stems from this category. The static calculation models are based on “bin-data” i.e. climate data is described in a table of the occurrence of outdoor temperatures. All relevant energy transfer processes are then calculated for each outdoor temperature. The result is then multiplied by the annual duration of the specific outdoor temperature. Adding all these results together forms the annual performance. Steady-state conditions are assumed for all conditions. The obvious advantage offered by this technique is the simplicity, which leads to fast computations and relatively low demands on climate data. “Bin” wheater data is available at a relatively low cost and there are a number of reasonable accurate synthetic climate generators available. The inherent disadvantages with static calculations are that no considerations for seasonal variations can be taken in to account in the calculation, thus the characteristics of a ground heat source is not adequately treated. Furthermore variations in electricity rates will not be reflected in the calculations.

2.2 Quasi-static calculations

In quasi-static simulations all calculations are performed in chronological order. The year is divided in an appropriate number of time steps. The choice of time step is based on the application, available climate data and calculation capacity. Typical time steps are 1 hour, 1 day, 1 week, 1 month. The shorter time step chosen the higher accuracy may be acquired. For all calculations steady-state conditions are assumed for each time step. The greatest benefits associated with a quasi-static calculation model are that seasonal effects will be reflected in the calculations as well as the opportunity to make comparisons between the calculations and monitored results in existing installations. The most commonly used time step is one hour. The advantages with a quasi-static calculation model appear at the price of more detailed climate data. The minimum time step is dependent on the time constant for the process being examined. There is a limit where the condition can no longer be considered to be at a steady state. In those cases a dynamic calculation method have to be applied.

2.3 Dynamic calculations

If the time constant for any of the described components is longer than the desired time step in a quasi-static calculation a dynamic method has to be applied. A simulation of the dynamic interaction between a set of components raises the demand for a meticulous description of all components. The amount of required input increase the burden on the user and limits the availability to a smaller group. The obvious advantage is that dynamic processes may be revealed in the simulation. The choice of calculation method will be decided on the overall aim of the simulation and competence level of the anticipated users. The general accuracy and required calculation capacity is depicted in Fig. 2.

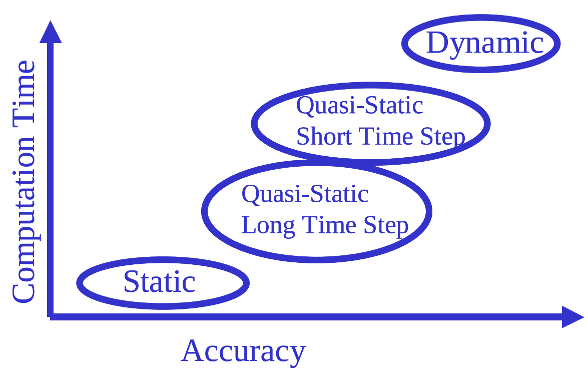

Fig. 2 Characteristics of different calculation methods

2.4 Choice of calculation method

In the initial phase of all software development the overall aim has to be defined. The aim and the identified users of the final product sets the boundaries for the user interface and serve as a guide in choosing the most appropriate calculation structure. Our objective, in this work, is to develop a design tool to be used in the contact between the contractor and the consumer. The results of the calculations shall, in an adequate way reflect realistic operating conditions and functionality of the simulated installation. The result of the simulation shall provide the consumer with information on the reduction of operating costs as well as required design of a borehole, in the case of a GSHP application. Based on the requirements for the simulation and taking into account for the identified group of users a quasi-static calculation structure has been chosen. The time step of 1 hour has been found to be most appropriate for this application.

3 THE SIMULATION MODELS

As the tool is designed to evaluate the potential for cost reduction by the use of heat pump technology in an existing building, the calculations are based on knowledge of present energy utilisation. A further limitation is that ducted air systems for heat distribution are not included in the models due to negligible existence on the Swedish market. The general simulation flow is depicted in Fig. 3 and in short described as follows. Based on the annual heat demand and climate data for the chosen location, an energy profile is established. The energy profile reveals the heating demand for each occurring outdoor temperature. The information in the climate database and energy profile is then used, in order to establish the heat demand, on an hourly basis. Given the design temperatures for the hydronic heat distribution system, the temperature levels (supply- and return temperature) may be acquired for all prevailing outdoor temperatures. This step is omitted in the case of an air-air heat pump. In the case of a GSHP a desired mean temperature for the incoming brine is set. Together with the information of the hourly heat demand, all governing operating conditions for the heat pump are set and all energy transfers may be acquired.

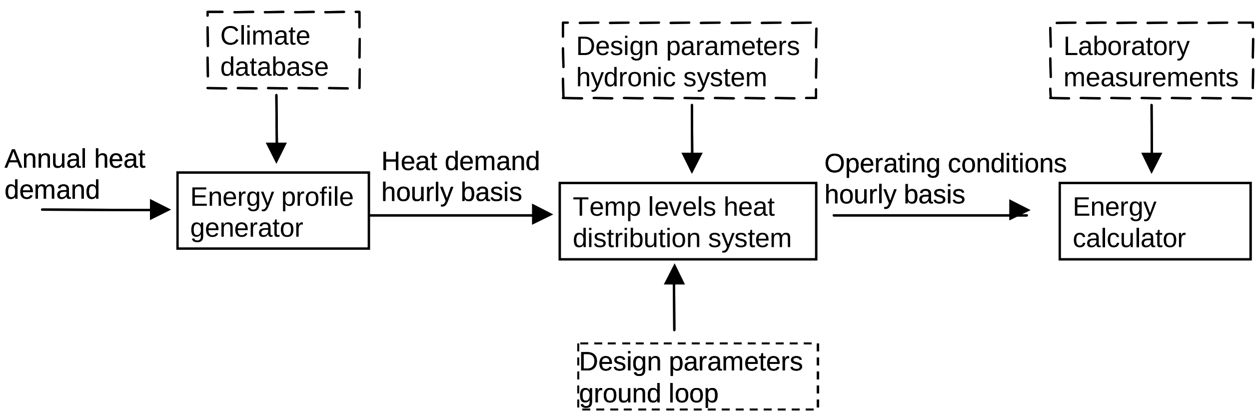

Fig. 3 Simulation flow

The following sections describe the procedures to obtain the energy profile, temperature levels for the simulated heat distribution system, design of vertical ground loop and modelling of the heat pump performance.

3.1 Establishing the energy profile

The procedure to establish the energy profile is based on a simplification that all the heat losses from a building can be lumped together in one general heat loss factor (HLF). The HLF is determined from the annual heat demand and the number of degree hours at the location.

$$\dot{Q} = HLF(t_{indoor} - t_{outdoor}) - \dot{Q}_{intgain}$$
$$Q_A = HLF \cdot degh \qquad (1a,b)$$

The concept of degree hours is widely used in many countries. The definition of how to calculate the number of degree hours may vary (Forsén 2000, SMHI), but the general idea remains the same. In this context the degree hours are calculated for each day when the mean temperature is below 11 °C. The internal heat gain ($t_{intgain}$) is set to 3 K.

$$\mathrm{degh} = \sum_{i=1}^{8760} (\mathrm{t}_{\mathrm{indoor}} - \mathrm{t}_{\mathrm{intgain}} - \mathrm{t}_{\mathrm{outdoor}}(i)) \tag{1c}$$

3.2 Heat distribution system

The general methodology to determine the supply- and return water- temperature for a radiator system has been described in the literature (Peterson, Nilsson 1988). The temperature levels of the hydronic heat distribution system are derived from an overall heat balance. At steady state condition the heat supplied to the radiator must equal the heat supplied to the room and the heat loss through the climate shield of the house. The heat supplied to the hydronic distribution system may be decided by equation 2a.

$$\begin{aligned} \dot{Q} &= \dot{m}\, cp (\mathrm{t}_{\mathrm{supply}} - \mathrm{t}_{\mathrm{return}}) \\ \dot{Q} &= c_1 (\mathrm{t}_{\mathrm{supply}} - \mathrm{t}_{\mathrm{return}}) \end{aligned} \tag{2a}$$

Heat loss from the climate shield may be determined by the following expression.

$$\dot{Q} = c_2 (\mathrm{t}_{\mathrm{indoor}} - \mathrm{t}_{\mathrm{outdoor}}) \tag{2b}$$

The overall heat transfer coefficient for the radiator can not be treated as a constant since it depends on the logaritmic mean temperature difference of the radiator. The relation for the overall heat transfer coefficient can be written as follows.

$$\begin{aligned} U &= U_{DOT} \left(\frac{\Delta t}{\Delta t_{DOT}} \right)^n \\ \Delta t &= \frac{t_{\mathrm{supply}} - t_{\mathrm{return}}}{\ln\left(\frac{t_{\mathrm{supply}} - t_{\mathrm{indoor}}}{t_{\mathrm{return}} - t_{\mathrm{indoor}}} \right)} \end{aligned} \tag{2b,c}$$

Index DOT denotes values at design outdoor temperatur and index n denotes the radiator exponent. The overall heat transfer coefficient at DOT (U_{DOT}) and the exponent n are given in catalogs from radiator manufacturers. The following relation can be written for the heat transfer from the radiator.

$$\dot{Q} = U_{DOT} A \left(\frac{1}{\Delta t_{DOT}} \right)^n \cdot (\Delta t)^{1+n} \tag{3a}$$

Which may be rewritten in

$$\dot{Q} = c_3 (\Delta t)^{1+n} \tag{3b}$$

By knowledge of the heat power demand ($\dot{Q}$), t_{supply}, t_{return}, $t_{outdoor}$, t_{indoor}, and the radiator exponent all constants c_1, c_2, and c_3 may be determined, thus the supply and return temperatures may be found for all occurring outdoor temperatures.

3.3 Design of vertical ground loop

There are numerous models and for the design of vertical boreholes described in the literature. An overview of existing software was performed by Hellstöm and Sanner (2001). The models are generally based on the line source theory or the cylindrical heat source method derived by Carslaw and Jaeger (1947). The cylindrical heat source method was later on expressed in a more convenient manner for this application by Ingerzoll et al (1954). In common for most of the models described in the literature as well as commercially available design tools are that they are giving a detailed description of all heat transfer processes that take place in the ground loop but only a rudimentary description of the heat pump. This is a drawback since the performance of the heat pump is interlaced to the temperature levels of the heating/cooling distribution system. The general simulation tool TRNSYS however, offer the opportunity to incorporate detailed ground source simulation into a complete building system. The user interface within TRNSYS is developed for use of a vast range of applications and consequently not all that easy to deal with. Bernier and Randriamiarinjatovo (2004) have implemented a complete set of models for system simulation in the more user-friendly environment offered by EES (2000). The system simulation presented by Bernier and Randriamiarinjatovo incorporates simulation of the building heat/cooling load as well as the heat pump and the ground loop. In this work, much inspired by the work of Bernier (2000), the cylindrical heat source method is applied. Variations in the thermal load of the borehole are treated by superposition.

3.3.1 General assumptions

In order to calculate the temperature at the wall of the borehole (T_V), the undisturbed ground temperature (T_B) for the location has to be established. The undisturbed ground temperature is the natural temperature of the ground unaffected by any ground heat exchanger, at a depth where seasonal variations are not observed. The estimation of the undisturbed ground temperature is based on the annual mean temperature of the location. This temperature is however an underestimation for locations that for longer periods are covered by snow. The snow cover serves as insulation for the ground, which results in that the undisturbed ground temperature will be somewhat higher than the annual mean temperature for the location. A correction of 1.5 K/100 days of coherent snow cover is recommended in Swedish national guidelines (VVS-Handboken 1963). Furthermore a correction of 1,6 K/100 m for the natural vertical temperature gradient is assumed as an average of Swedish bedrock.

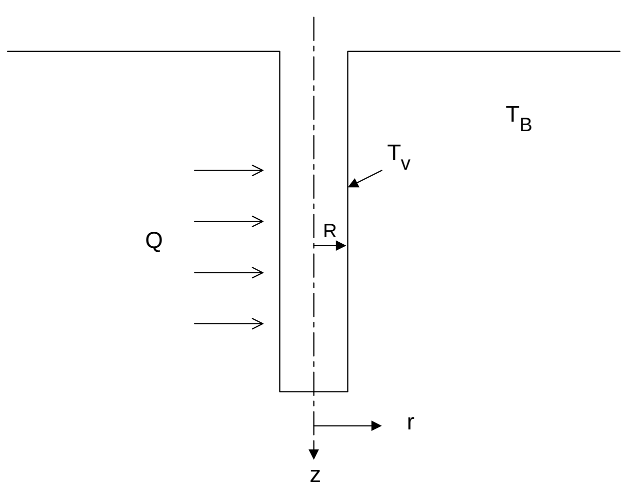

Fig. 4 Geometry of vertical borehole

3.3.2 Heat transfer model

The borehole is considered as an infinitively long cylinder embedded in infinite bedrock with known heat transfer characteristics. All heat transfer is assumed to be perpendicular to the borehole. An analytical solution for the temperature decrement around the cylinder has been derived by Ingerzoll et al (1954).

$$T_B - T_{V(t)} = \frac{\dot{Q}}{L\lambda\pi^2}\int_0^\infty \frac{e^{-\beta^2 Fo(t)} - 1}{j_1^2(\beta) + Y_1^2(\beta)}\left[j_0(\frac{r}{R}\beta)Y_1(\beta) - j_1(\beta)Y_0(\frac{r}{R}\beta)\right]\frac{\partial\beta}{\beta^2} \quad (4)$$

For simplicity reasons part of the expression above is expressed in terms of a function of the Fourier number

$$\frac{1}{\pi^2}\int_0^\infty \frac{e^{-\beta^2 Fo(t)} - 1}{j_1^2(\beta) + Y_1^2(\beta)}\left[j_0(\frac{r}{R}\beta)Y_1(\beta) - j_1(\beta)Y_0(\frac{r}{R}\beta)\right]\frac{\partial\beta}{\beta^2} = f(Fo) \quad (5)$$

The expression, which is valid for constant heat transfer rate, may be used for variable heat transfer rate by the use of superposition. The method of superposition or load aggregation is in detail described in other publications (Kavanaugh and Rafferty 1997) and (Bernier 2001). Using the simplified expression above and applying the superposition method of three different heat transfer rates result in the following expression.

$$T_B - T_{V(t_3)} = \frac{\dot{Q}_1}{L\lambda}\left[f\left(Fo_{t_3-0}\right) - f\left(Fo_{t_3-t_1}\right)\right] + \frac{\dot{Q}_2}{L\lambda}\left[f\left(Fo_{t_3-t_1}\right) - f\left(Fo_{t_3-t_2}\right)\right] + \frac{\dot{Q}_3}{L\lambda} f\left(Fo_{t_3-t_2}\right) \quad (6)$$

As the ground loop is designed for many years of continuous operation the simulation is performed for 11 years of operation. The mean cooling load over 1 year is used for the first 10 years of operation and thereafter the mean cooling load for each day of the year is used. In this way the number of terms in equation 6 is reduced to 366.

3.3.3 Determination of borehole resistance and brine temperature

The method previously described will establish the temperature at the borehole wall. The following text will describe the method used to obtain the thermal resistance of the borehole and the temperature of the brine entering the evaporator. Configuration of the borehole heat exchanger (BHE) is in this study limited to single U-pipe surrounded by ground water. This is the prevalent configuration on the Swedish market. Back-filling of boreholes is restricted to a very small number of installations in Sweden. Figure 4 depicts a cross-section of the idealised borehole.

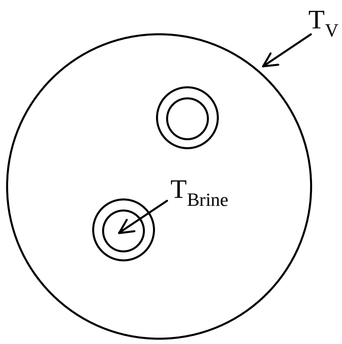

Fig. 5 Cross-section of borehole

The temperature difference in between the borehole wall (T_V) and the temperature of the brine (T_{Brine}) is obtained by the use of electric resistance equivalence.

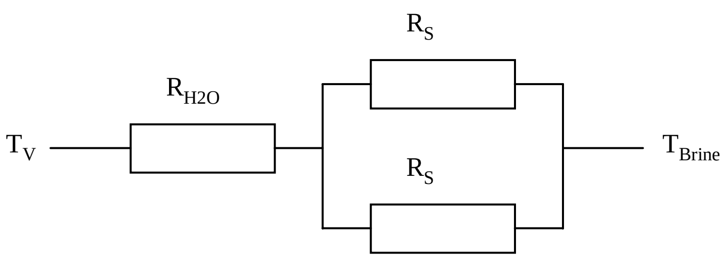

Fig. 6 Borehole resistance

$$T_V - T_{Brine} = \frac{\dot{q}}{R_b}$$

$$R_b = \frac{R_s}{2} + R_{H_2O} \qquad (7a,b)$$

The thermal resistance of the pipe (R_s) is determined by:

$$R_s = \frac{\ln(D_y / D_i)}{2\pi\lambda_{pipe}} + \frac{1}{\pi D_i \alpha_i} \qquad (8)$$

Heat transfer coefficient on the inside of the pipe (α_i) is determined by:

$$\mathrm{Re} \geq 2300$$
$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}$$
$$\mathrm{Re} < 2300$$
$$Nu = 1.86(\mathrm{Re}\,\mathrm{Pr}\,d_i / L)^{1/3} \qquad (9a,b)$$

The thermal resistance induced by the material surrounding the pipes (R_{H2O}) in the borehole is calculated by use of a shape factor (S_b). The shape factor suggested by Remund (1999) is used in this project.

$$R_{H_2O} = \frac{1}{S_b \lambda_{H_2O}} \qquad (10)$$

An energy balance of the borehole loop then gives the incoming brine temperature ($T_{in\ brine}$) to the evaporator.

$$T_{in\ brine} = T_B + \frac{\dot{q}}{2\dot{m}C_p} \qquad (11)$$

3.4 Modelling of the heat pump

As the overall aim of the project is to develop a tool for evaluation of existing heat pumps, the modelling of the heat pump is done by the use of performance data files. The performance data files are produced by use of monitored results from performance tests run according to the European norm EN 14511. In order to reach good accuracy the number of test points has been increased. The performance data files have to go through an evaluation by a third party before it is being approved for use in the software. Heat output as well as electric input has to be monitored at the following test points.

Brine-water heat pump: In addition to the test points in the table below, the maximum temperature of the water leaving the condenser has to be specified. The EN 14511 gives the flow rate for the brine- and water-circuit.

Table 1 Test points for brine-water heat pumps

	Water temp leaving condenser	35°C	45°C	55°C
Brine temp entering evaporator	-5°C	x	x	x
	0°C	x	x	x
	+5°C	x	x	x

Air-water heat pump: In addition to the test points in the table below, the lowest operation temperature has to be specified. The outdoor temperatures are given for dry bulb temperature and wet bulb temperature (with in the brackets).

Table 2 Test points for air-water heat pumps

	Water temp leaving condenser	35°C	45°C	55°C
Outdoor temperature	-7°C(-8°C)	x	x	x
	+2°C(+1°C)	x	x	x
	+7°C(+6°C)	x	x	x

Capacity controlled air-air heat pump: Heat output and electric input for the given test points in the table below are to be taken as the average value over a whole test cycle, including defrosting period, if required. If the heat pump is supplied with an auxiliary heating cable the electric input for that has to be included. In addition the lowest operation temperature has to be specified.

Table 3 Test points for capacity controlled air-air heat pumps

	Relative capacity	100%	75%	50%
Outdoor temperature	-15°C	x		
	-7°C(-8°C)	x		
	+2°C(+1.0°C)	x		x
	+7°C(+6°C)	x	x	x

Linear interpolation is used to obtain the performance in between test points.

4 IMPLEMENTATION OF PROPOSED MODELS IN COMPUTER SOFTWARE

All the models that are described in the previous sections have been implemented in the commercially available software, Prestige, distributed by The Swedish Heat Pump Association. Much attention has been paid into the development of a user-friendly interface; enabling sound evaluations of heat pump systems for users with unpretentious experience in system simulation and advanced simulation tools. The software is distributed with climate data files for more than 100 Swedish locations and a number of standardised heat pump performance files. There are at present time more than 600 licensed users of the software.

5 FURTHER WORK

In order to accomplish the user-friendly interface a lot of the parameters used in the underlying models are hidden for the user and thus hindering more experienced users for advanced parameter studies. This automatically raises the demand for a more advanced version of the software, which is currently under development. An English user interface and climate files for locations outside Sweden will be implemented to facilitate international use. In order to supply the user with information of thermal conductivity of the ground rock collaboration with Geological Survey of Sweden has been initiated.

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NOMENCLATURE

A	Area	m²
C_p	Specifik heat capacity	J/(kg·K)
degh	Degree hour	K h
D_y,D_i	D_y, Diameter external, D_i, Diameter internal	m
Fo	Fourier number	-
HLF	Heat loss factor	W/K
L	Borehole length	M
$\dot{m}$	Mass flow rate	kg/s
N	Radiator exponent	-
Nu	Nusselt number	-
Pr	Prandtl number	-
$\dot{Q}$	Heat load	W
R, r	Radius, Thermal resistance	M, (m K)/W
Re	Reynolds number	-
S_b	Shape factor	-
T, t,	Temperature, time in equation 6	K, s eqn 6
α	Heat transfer coefficient	W/(m²·K)
λ	Thermal conductivity	W/(m·K)

REFERENCES

Bernier, B., Randriamiarinjatovo, D. 2004. Annual Simulations of Heat Pump Systems With Vertical Ground Heat Exchangers. Proceedings from The bi-annual conference of IBPA-Canada, Se-Sim 2004, June 9-11, 2004, Vancouver, BC, Canada.

Bernier, B. 2000. A Review of the Cylindrical Heat Source Method for the Design and Analysis of Vertical Ground-Coupled Heat Pump Systems, 4th International Conference, Heat Pumps In Cold Climates, Caneta Research, August 17-18, Ottawa, Canada.

Bernier, B. 2001. Ground-Coupled Heat Pump System Simulation. ASHRAE Transactions 107 part 1.

Carslaw, H. S., Jaeger, J. C. 1947. Conduction of Heat In Solids. Oxford, U.K.

Forsén M. 2000. Degree Days At 13 Nordic Locations, Dept. of Energy Technology, Royal Institute of Technology. Stockholm.

Hellström G., Sanner B. 2001. PC-programs and modelling for borehole heat exchanger design, International Geothermal Days Germany, Bad Urach, ISS, Skopje, Macedonia.

Ingersoll L. R., Zobel O. J., Ingersoll A. C.1954. Heat Conduction With Engineering, Geological and Other Applications, Revised Edition, McGraw-Hill.

Kavanaugh S.P., Rafferty K. 1997. Ground-Source Heat Pumps: Design of Geothermal Systems for Commercial and Institutional Building. ASHRAE, Atlanta, GA, USA.

Klein S. A. 2000. EES- Engineering Equation Solver, F-chart software, Madison, Wisconsin, USA.

Nilsson P-E. 1988. Anslutning av värmepumpar till befintliga värmesystem. Värmetekniska och ekonomiska begränsningar, Document D6, Chalmers University of Technology.

Peterson F. Reglerkurvan, Debatt #1, A4-Serien #100, Dept. of Energy Technology, Royal Institute of Technology. Stockholm.

Remund C. P. 1999. Borehole Thermal Resistance: Laboratory and Field Studies, ASHRAE Transactions 105 part 1.

Swedish Metrological and Hydrological Institute, SMHI. Produktblad Graddagar, SMHI, Norrköping.

Swedish Heat Pump Association, SVEP 2005. Unofficial sales statistics. SVEP Information & Service AB, Stockholm.

VVS-Handboken 1963. Förlags AB VVS, Stockholm, Sweden.