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# DYNAMIC MODELING AND SIMULATION OF GEOTHERMAL HEAT PUMP SYSTEMS BASED ON A COMBINED MOVING BOUNDARY AND DISCRETIZED APPROACH

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## ABSTRACT

The major goal of this study is to demonstrate the feasibility of the dynamic modeling and simulation of both conventional and direct exchange geothermal heat pump applications particularly with regard to performance evaluations. Therefore, in this research, dynamic models for the simulation of geothermal heat pump systems with the working fluid propane for the application in building-scale energy systems have been developed based on the Moving Boundary approach and the challenges of dynamically evaluating these kind of energy systems have been hereby met. Additionally, the performance of both horizontal (slinky-coil) and vertical (borehole) geothermal heat exchangers, which have been modeled based on the Conventional Discretized approach, for such systems has been evaluated. Since these two modeling approaches are not compatible with one another, the coupling of the moving boundary model with the discretized model is therefore another challenge and of great importance. On the basis of a case study, the complete conventional and direct exchange geothermal heat pump systems with vertical and horizontal heat exchangers have been simulated and energetically and exergetically analyzed.

## INTRODUCTION

A geothermal heat pump system consists of a conventional heat pump coupled with a ground heat exchanger where water or a water-antifreeze mixture exchanges heat with the ground. Although existing heat pump systems can already reach an annual coefficient of performance (COP) of over 4, there is still a substantial potential to tap.

A direct exchange geothermal heat pump system is a geothermal heat pump system in which the refrigerant circulates through tubing placed in the ground. Direct exchange geothermal heat pumps offer higher efficiencies at even lower installation costs compared to conventional geothermal heat pumps. In addition, direct exchange ground collectors require less space in the soil. However the correct dimensioning of this type of application is not yet very well investigated. Currently, only numerical simulations may offer a suitable tool for the tailor-made design of such applications and can make long term predictions of the performance. Therefore, with the aid of dynamic modeling and simulation, which is a convenient and low-cost engineering tool for the performance evaluation of HVAC systems, the feasibility of modeling such complex thermo-hydraulic systems in the modeling language Modelica has been proved and the potential of different designs for them has been demonstrated. Possible optimizations of the system can then be easily identified by means of a first and second law thermodynamic analysis, which provides detailed information on losses and the efficiency of each component.

## DYNAMIC MODELING

Two conventional ways of characterizing heat pump models are, firstly, by means of the physical approach of the model:

- Thermodynamic/physical approach: based on the geometry of the heat pump and general laws of physics (heat and mass transfer)
- Black box approach: completely empirical
- Grey box approach: intersection of physical and black box approach

and secondly, depending on the dynamic representation of the system [Blervaque et al., 2012]:

- Rated performances: part-load performance is calculated by a temperature-rated full-load and steady-state operation
- Quasi-static: Time is considered as a sequence of steady states
- Dynamic: Transient and steady state phases are both represented in a simulation.

Since no measurements for validation of the heat pump system is available, the physical approach must be relied on. Furthermore, we are interested in tracking the dynamics of the system. Dynamic simulations however are, depending on the system's complexity, much more computational intense than static or quasi-static simulations. Therefore, the main emphasis lies on finding transient formulations of the system's components, including the refrigerant, that can be solved within an acceptable time.

**Modeling and simulation environment**

In this study, the dynamic models have been developed in the modeling language MODELICA, and the MODELICA Standard Library version 3.2 have been applied to simulate the hydraulic and thermal behavior of the system. The translation of a Modelica model is performed in the simulation environment Dymola, version 2013 [Dassault Systems, 2011].

**Medium model**

The design and simulation of heat pumps or power cycles require an accurate representation of the working fluid. The selected refrigerant in this study is propane, which will be used for the application in both conventional and direct exchange geothermal heat pump systems. A medium model for the refrigerant propane entitled Propane_Fast developed by [Sangi et al., 2014] has been used in this research.

**Advanced transient modeling of two phase flow**

This sub-section will give an overview of two common dynamic modeling approaches for two phase flows and explain the challenges and difficulties that occur when a two phase fluid model is used for simulation. Additionally, a low order model based on a moving boundary formulation as well as the implementation of this model will be explained.

The motivation for introducing this relatively-new moving boundary method, also known as lumped parameter model, has its origin mainly in the observation of a greatly varying physical behavior in the sub-cooled, the two-phase and the superheated region in two-phase heat exchangers [Jensen, 2003]. The heat transfer coefficient for example, can differ significantly from the superheated region to the two-phase region. In each region however, it may be relatively constant. In case of utilizing the finite volume method, this circumstance has two consequences:

- The spacial discretization needs to be high so that the ratio of two phase and single phase can be represented accurately.
- The simulation will slow down each time one fluid volume switches from single phase to two phase or vise versa, as the solver reduces its time step time accordingly when the state variables have large transients.

***Moving Boundary Method***

To derive a low order model that can overcome the challenges that were pointed out in the previous sub-section, the devision into the three regions sub-cooled, two-phase and superheated was therefore a natural selection. The idea of this approach is to model each region as a Control Volume (CV) that has variable boundaries and average properties and track the length of the different regions dynamically [Jensen, 2003]. In this way, the occurrence of discontinuities can be avoided and the number of state variables is kept small, which allows large simulations to be run on any computer.

The description of the direct exchange geothermal heat exchanger using a moving-boundary formulation follows the approach presented by [Li and Alleyne, 2010] for refrigeration systems, [Zapata et al., 2013] for solar thermal once through cavity receivers and [Bonilla et al., 2012] for direct steam generating solar thermal power plants. The fluid and wall energy balances as well as the mass balance for the fluid result in three equations for each region. A moving boundary model assumes the momentum balance to be negligible since the frictional pressure drop and the gravitational pressure drop are very small (or non existing), compared to the pressure drop of other components like the compressor or the expansion valve, for conventional refrigeration systems with compact evaporators and condensers.

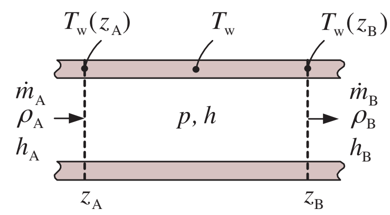


*Figure 1: Illustrastion of a fluid volume*

Under the assumption of a one-dimensional flow and with the nomenclature presented in Figure 1 the general fluid mass and energy balances with variable boundaries A and B yield equation (1) and (2). The wall mass balance is superfluous, however the wall energy balance can also be simplified and results in equation (3).

$$A\frac{d}{dt}\int_{z_A}^{z_B} \rho dz + A\rho_A \frac{dz_A}{dt} - A\rho_B \frac{dz_B}{dt} = \dot{m}_A - \dot{m}_B \quad (1)$$

$$A\frac{d}{dt}\int_{z_A}^{z_B} \rho h dz - A(z_B - z_A)\frac{dp}{dt} + A\rho_A h_A \frac{dz_A}{dt} - A\rho_B h_B \frac{dz_B}{dt} = \dot{m}_A h_A - \dot{m}_B h_B + \dot{q}(z_B - z_A) \quad (2)$$

$$
\begin{aligned}
&A_{\mathrm{w}}\rho_{\mathrm{w}}c_{\mathrm{w}}(z_{\mathrm{B}}-z_{\mathrm{A}})\frac{dT_{\mathrm{w}}}{dt}\\
&+A_{\mathrm{w}}\rho_{\mathrm{w}}c_{\mathrm{w}}(T_{\mathrm{w}}(z_{\mathrm{A}})-T_{\mathrm{w}})\frac{dz_{\mathrm{A}}}{dt}\\
&-A_{\mathrm{w}}\rho_{\mathrm{w}}c_{\mathrm{w}}(T_{\mathrm{w}}(z_{\mathrm{B}})-T_{\mathrm{w}})\frac{dz_{\mathrm{B}}}{dt}\\
&=\dot{q}_{\mathrm{w}}(z_{\mathrm{B}}-z_{\mathrm{A}})
\end{aligned}
\tag{3}
$$

Based on these simplified mass and energy balances, Differential Algebraic Equations (DAEs) can be found for all three regions. The derivation of these DAEs is not part of this study, since this has already been done by many other authors. This work follows the approach of [Bonilla et al., 2012], who describes an object-oriented library of switching moving boundary models for two-phase flow evaporators and condensers, as well as [Zapata et al., 2013], who describes a method for the dynamic simulation of the heat exchange in a solar thermal once through cavity receiver coupled with a switching approach that allows the application of the moving boundary method for a great variety of boundary conditions.

In the most general case, an evaporator has three prevalent regions. As the evaporator's operating condition is not always subcooled liquid at the inlet and superheated vapor at the outlet, it is necessary to provide further models that describe their behavior. Also it is necessary to have a consistent description during the transition from one model to another.

**Switching algorithm for the moving boundary approach**

As indicated previously, when one zone appears or disappears, it must be switched between different model equations for the different modes. In total, there are twelve switching possibilities with the corresponding switching criteria for the evaporator or condenser. All switching criteria involve an expression for the specific enthalpy at the crossing boundary. For disappearing zones the length is also taken into account. Thresholds can be chosen for each condition and should be non zero, otherwise the solver might get trouble solving the DAEs.

After switching from one mode to another, the whole system must be reinitialized, since it is not certain, especially if a zone disappears, that both region length and specific enthalpy converge simultaneously towards their respective switching references. Just switching between the model equations would then result in wrong calculations and the pseudo-equations would introduce a non-conservative behavior, which is not desired and unacceptable for performance evaluations. To avoide that, all state variables shoul be reinitialized in a conservative way.

**Thermodynamic model of the heat pump**

A vapor compression heat pump cycle must at least consist of the the four following basic components: evaporator, compressor, condenser and expansion valve. Figure 2 shows the general model of this cycle. Each component is coupled internally through fluid connectors that equalize the outputs of one model with the inputs of another model.

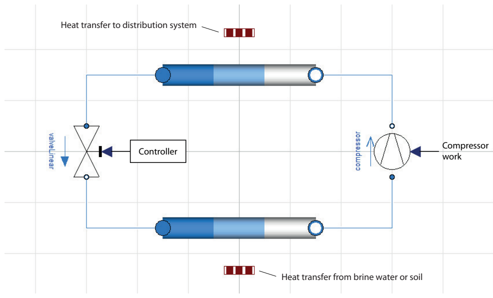


*Figure 2: Components of the heat pump cycle*

***Control strategies for the evaporator***

The valve is controlled with a simple proportional-integral (PI) controller. Therefore, two methods have been investigated: the first control strategy takes the evaporator outlet pressure as a reference, while the second strategy acts on changes in the temperature difference between the inlet and the outlet of the evaporator. After evaluating the suitability for the purposed application, the second strategy was chosen for controlling the heat exchanger. The reason for this decision is the fact that evaluating the temperature difference over the evaporator ensures the presence of a superheated region at all times. This circumstance is essential for a safe operation of the compressor. Also the two phase length can be maximized even under dynamic conditions. A pressure controller however can not overcome these challenges and the set point must be chosen according to the prevalent temperatures in evaporator. In this case, a change in temperature of 1 K on the source-side of the evaporator may actually change the flow situation on the sink-side from almost completely flooded to almost completely superheated. This behavior is not desired and can reduce the efficiency of the heat exchanger.

**Geothermal field model**

In this research, a model for a field of closed loop borehole heat exchangers and a model for horizontal slinky-coil heat exchanger including the model for the surrounding soil has been developed. Only three heat ports are necessary for these models. The side connector is connected to the undisturbed soil temperature, the bottom heat port is connected to the constant geothermal heat flow from the earth core and the top connector is connected to the surface temperature, which basically varies with the ambient temperature and the radiation from the sun. For a brine water heat pump the fluid connectors are simply connected to the heat pump model. In case of a direct expansion heat pump, the whole system becomes a little bit more complicated and the models have to be modified, as the moving boundary heat exchangers are used for this purpose. Here the entire internal interface of the developed geothermal model has been replaced with the heat exchanger model presented in Figure 2.

**Coupling of the moving boundary model with the discretized model**

Since the moving boundary model has variable cell sizes and adjacent cells of other models have, in case of a discretized approach, fixed sizes, the coupling of the moving boundary model with the discretized model therefore is of great importance. Mathematically, these methods were described by utilizing "`min()`" and "`max()`" functions that return the smallest or largest value of an array. In this way, it is possible to obtain an explicit set of equations, which is also continuous when the moving boundary crosses a fixed boundary.

## RESULTS AND DISCUSSION

Based on a case study described in the next section, a dynamic simulation for a conventional brine water heat pump has been performed and the results have been presented and analyzed. Furthermore, the three technologies of brine water heat pump, direct exchange slinky-coil heat pump and direct exchange borehole heat pump have been analyzed from a thermodynamic point of view.

**Description of the case study**

To demonstrate the feasibility of the developed approaches in this research, a case study has been set up and a complete heating system of a one-family house equipped with a brine water heat pump has been investigated. Since the emphasize on this research is placed on the thermo-hydraulic part of the system, the heat distribution system has been idealized and a theoretical heat demand profile has been generated. Figure 3 shows the overall system. As it can be seen, the case study contains four main sub-systems, which are explained in the following paragraphs.

***Heat demand profile***

The theoretical heat demand profile has been generated with a single-family house model and weather data for a location near Hamburg. Already-developed house and weather models have been used for this purpose. The Test Reference Year (TRY) is 2010, and the following assumptions have been made:

- The room temperature is held constant at 21°C
- Negative heat flows in summer are neglected.

***Buffer storage***

The buffer storage has been idealized by means of a lumped heat capacity. The stratification, observable in a real system, has been thereby completely neglected. This simplification is feasible as the major dynamic behavior that is introduced by a buffer storage is its inertia. The storage capacity has been chosen to be 1000 liters. The circulating pump on the hot side of the heating system is controlled continuously according to the heat demand.

***Heat pump model***

A heat pump model has been developed within this study. The compressor is controlled by an On-Off controller, which evaluates the set point temperature and shows a hysteresis behavior. In this case, the compressor power is 1000 W. Also the heat exchanger dimensions have been chosen in a way that realistic values for their terminal temperature difference occurred under steady state conditions. The controller set point temperature difference for the evaporator was set to 2.5 K.

***Soil model***

The soil model used in this part of this research is the the soil with slinky coil heat exchanger described in the geothermal field model section and the dimensions have been chosen according to [VDI 4640 2, 2010].

**Simulation results of the case study**

Simulation results of the case study for a conventional brine water heat pump system have been presented in Figure 4. The heating demand and the heat pump power, the flow and return temperatures of the heating water cycle and the maximum temperature in the heat pump (after the compressor), the flow and return temperatures for the geothermal field, the condenser pressure, the evaporator pressure, the two phase zone length of the condenser and the two phase zone length of the evaporator have been depicted in the corresponding graphs.

The compressor is either switched on or off, as can be seen by the block structure of the power signal. This is caused by the hysteresis behavior of the On-Off controller.

The temperature after the compressor (black line in the second graph) fluctuates as a result of the dynamic operation of the heat pump. The flow temperature differs from the refrigerant temperature at the inlet of the condenser by approximately 15 K at almost all times during operation. The refrigerant temperature after the compressor decreases by around 5 K after the shutdown of the heat pump. As the recirculation pump is also connected to the controller signals of the heat pump, a constant temperature difference of approximately 3 K between flow and return is maintained. To sum up, the flow temperature (red line in graph B) fluctuates around the set point of the heat pump.

In the next graph C, the flow and return temperature of the collector field has been portrayed. The water temperature of the evaporator outlet is also influenced by the dynamic behavior of the heat pump. During operation, the controller is capable to maintain a constant temperature difference of around 5 degrees between flow and return. After the shut down of the heat pump, the flow temperature converges towards the average temperature of the evaporator. Notable are the peaks

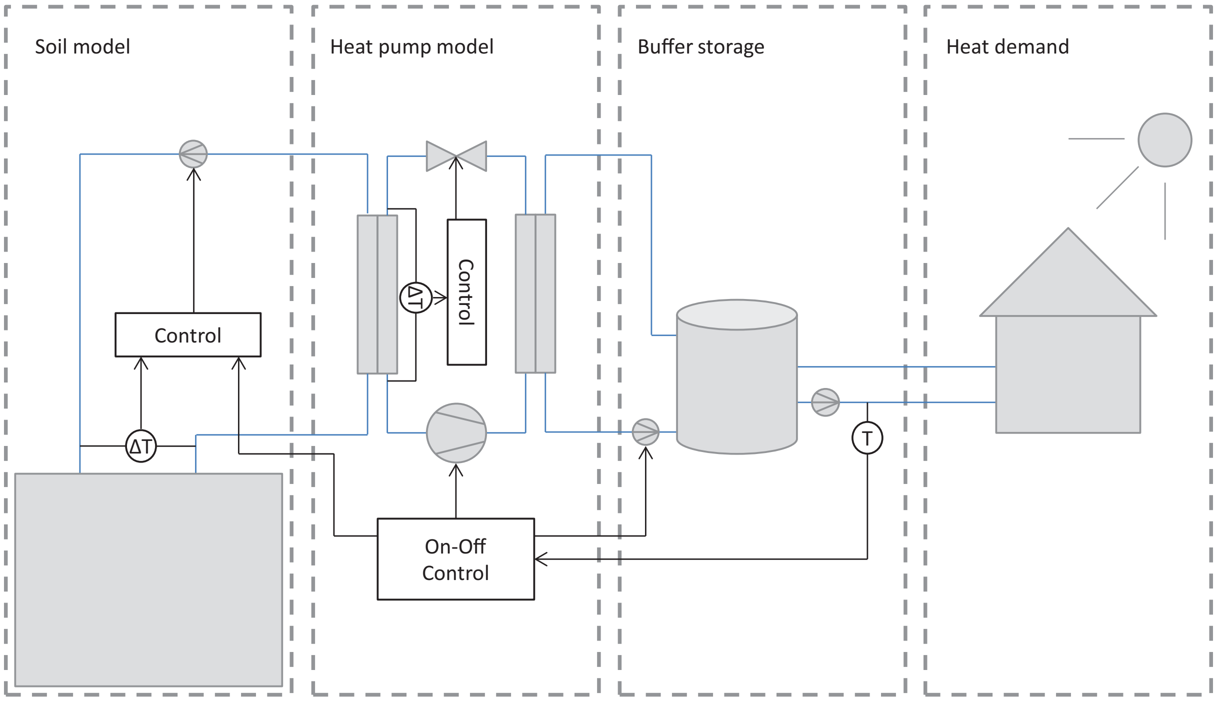


*Figure 3: Illustration of the overall system of the case study*

during the shut down, which can be explained by the decrease of the evaporator pressure (graph E) originating from the chosen control strategy of the evaporator. At the beginning, the water at the inlet of the evaporator has the same temperature as the initialization temperature of the geothermal field, but its temperature slowly decreases due to the heat extraction.

In the condenser, the pressure is predominantly determined by the flow and return temperatures of the heating system. This value varies between 12 and 20 bars. The pressure in the evaporator however is influenced actively through the controller of the expansion valve. This maintains a certain temperature difference between in and outlet of the heat exchanger at the refrigerant side.

During operation, the two-phase zone in the condenser is relatively large and accounts for approximately 90 % of the total length. Directly after the shout down, the super heated region disappears almost completely. A totally different behavior is observable for the evaporator. Here, strong fluctuations of the two-phase zone length occur. During operation, this lengths is around 90 % of the evaporator length. Despite the fact that the pressure in the evaporator deceases by around 0.5 bar, the two-phase zone length remains constant for the whole simulation. This can be explained by the control strategy which aims to achieve a good wetting of the evaporator wall. As soon as the heat pump shouts down, the temperature in the whole evaporator rises abruptly. The refrigerant evaporates almost completely and consequently the pressure increases. Choosing a pressure control introduces less pressure dynamics to the heat exchanger but is not as flexibel as the chosen control strategy in terms of varying operation temperatures for the evaporator.
These simulation results appear to be plausible with regard to the dynamic operation of heat pump heating systems.

## Thermodynamic analysis

The first and second laws of thermodynamics have been applied to the overall heat pump system of the case study and each component has been investigated both energetically and exergetically. Figure 5 shows the general system setup for the conventional ground source heat pump investigated in the case study as well as a horizontal and a vertical direct exchange heat pump. The boundaries of the two-cycle heat pump have been chosen according to the figure. Here, for each of the three heat pump systems, the heat demand is fixed to 3 kW and consequently the compressor power is adjusted.
For the two-cycle heat pump, the same heat pump model from the case study has been used. The only difference is a fixed soil temperature of 10 °C. Also the buffer storage is not considered in this analysis since for all three systems it has been assumed that the heat distribution is not changed. One further assumption, to obtain a more realistic situation, is that the return temperature of the ground source collector has already reached 5 °C.
It would not be fair to compare two technologies with the assumption that both only have the same amount of heat available from the soil. This would automatically affect the more efficient technologies adversely. In order to be able to yet compare the three systems, it has been assumed that the outlet temperature from the

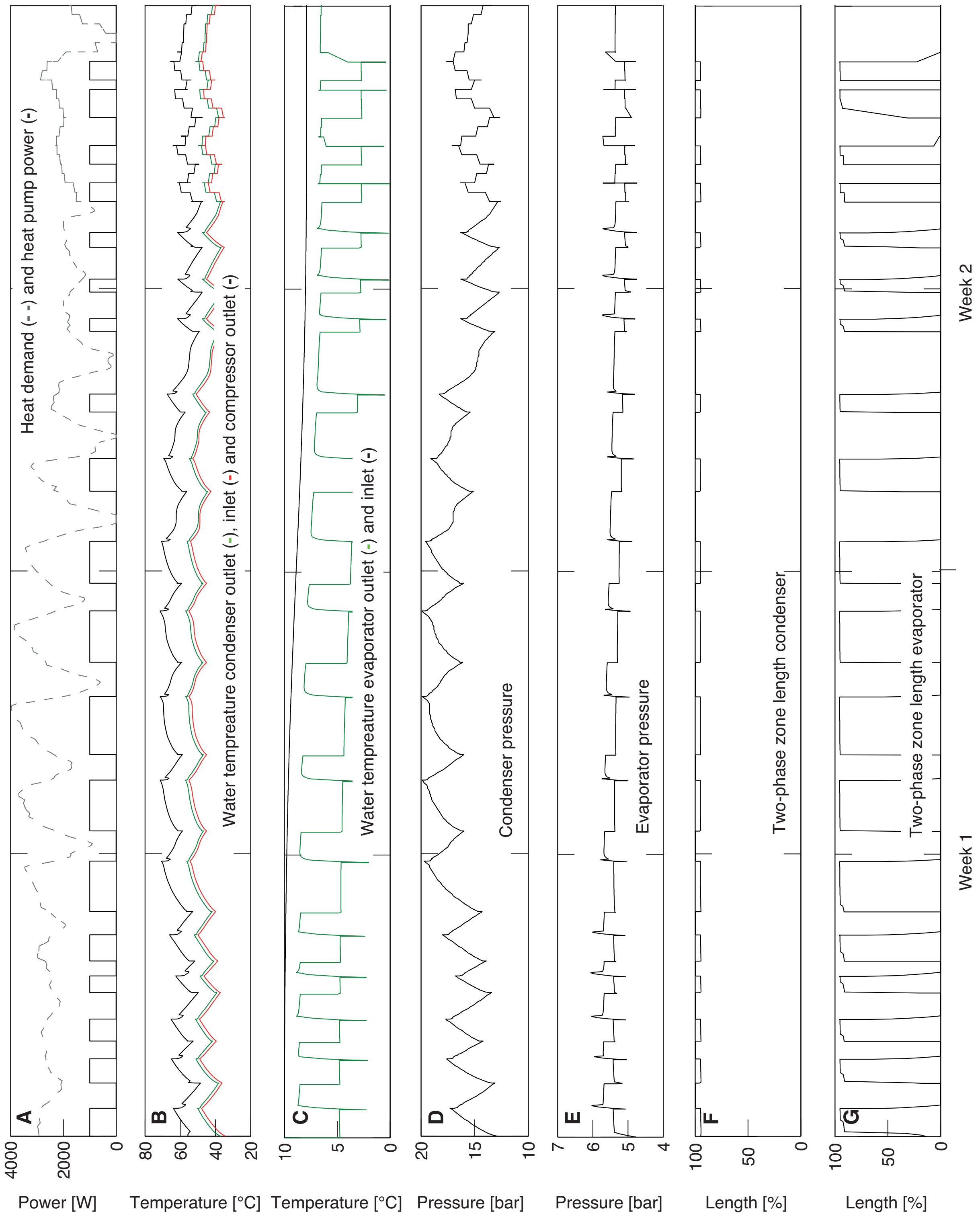


*Figure 4: Simulation results*

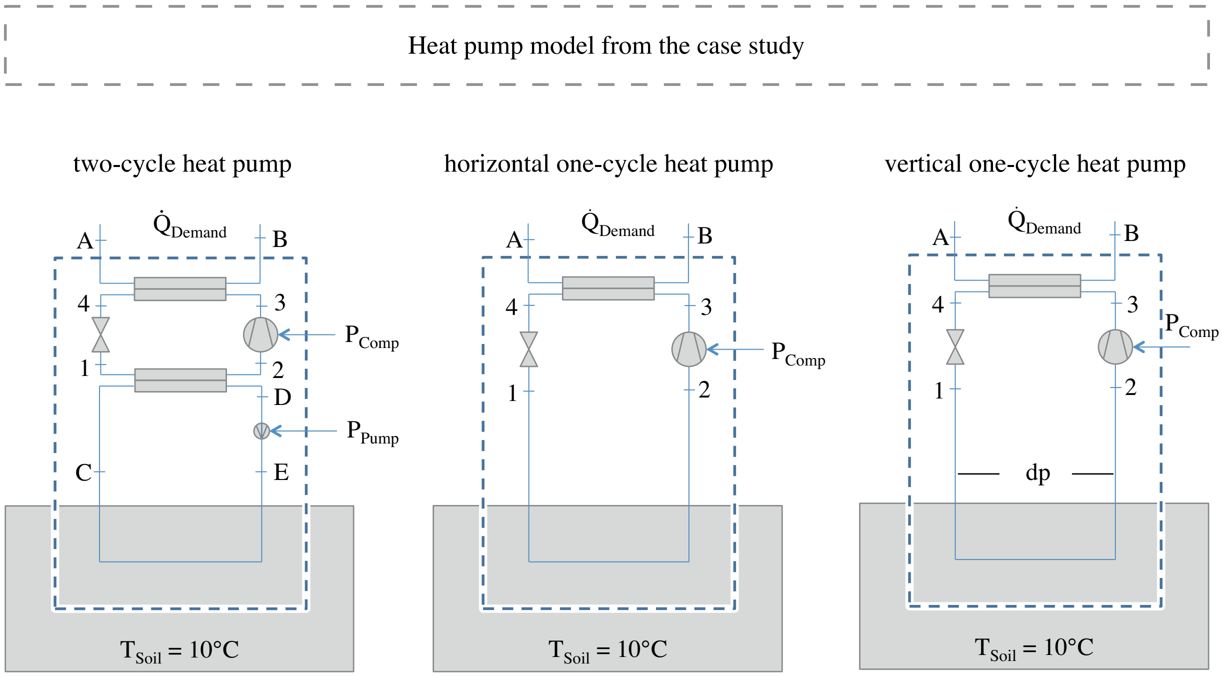


*Figure 5: Boundaries for the exergy analysis*

soil is approximately the same for all three systems. This of course requires a different size of the ground source collector or the borehole heat exchanger.

For a brine water heat pump it is totally sufficient to know the inlet and outlet conditions for the ground source heat exchanger when it comes to exergy analysis. For the direct exchange systems, an assumption for the wall temperature has to be made so that a component wise analysis is still possible. Therefore, the wall temperature has been assumed to be constant (in this case 5 °C).

The vertical one-cycle heat pump has the same setup as the horizontal one. The only difference is that a constant pressure increase between the inlet and the outlet of the borehole heat exchanger has been assumed. This pressure drop of 0.2 bar corresponds to a borehole depths of 100 m.

In order to compare the two-cycle application with the one cycle application, all boundary conditions are the same for both systems. The ambient temperature has been chosen to be at −5°C and the ambient pressure is 1 bar. The undisturbed soil has the same temperature of 10°C for all applications.

In this thermodynamic analysis three approaches where followed: The first one only evaluates the relative irreversibilities and the system efficiency on the heat pump basis, the second one considers all components of the system and the third one takes into account that the soil has an exergy content too, which is regarded as fuel.

For all three cases and different boundaries, the exergy efficiency and the coefficient of performance have been shown in Figure 6. Most noticeable here is the fact that the direct applications are slightly better than the conventional type. This is because of the absence of a recirculation pump. The vertical type is even more efficient than the horizontal direct exchange heat pump. In conclusion, the application of direct exchange geothermal heat pumps is to be considered positively in terms of exergy efficiency and overall system performance.

## CONCLUSIONS

In the course of this work a dynamic model of a complete heat pump system was developed in the object-oriented programing language Modelica. At first, a medium model for propane was implemented, which proved to be stable, accurate and much faster than the other availabe media libraries. However, the simulation speed was still not good enough to simulate a closed thermo-fluid cycle with standard components of the Modelica libraries. Hence, most attention was paid to finding a transient and yet computational simple model for the heat exchangers of the heat pump. Therefore, a two-phase flow model using the moving boundary method in combination with a switching approach was implemented and tested. With the aid of a coupling approach developed in this research, it was then possible to build a closed cycle model of a heat pump system and connect it to other adjacent discretized components.

The developed Modelica component of the moving boundary heat exchanger is universally usable for dynamic simulations with any two phase flows and can thus be used not only for the modeling of refrigerant applications, but also for any kind of boiler or condenser. Especially for the design of accurate controllers, the moving boundary approach offers a valu-

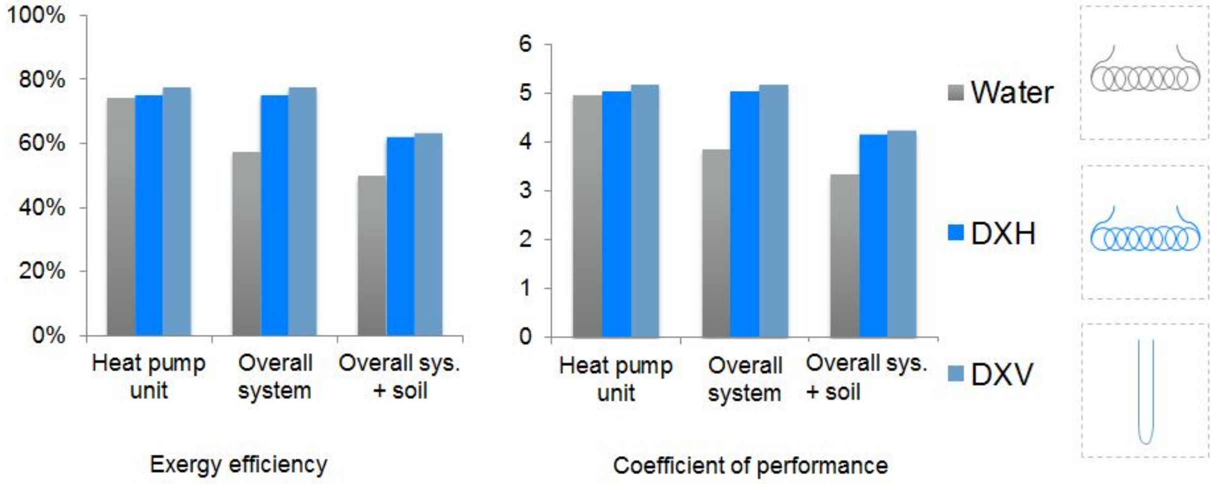


*Figure 6: Exergy efficiency and Coefficient of performance*

able tool.
On the basis of a case study, a complete heating system was simulated. Particularly, the dynamics introduced by the start-up and shut-down of the compressor could be reproduced. The simulation speed for this fully dynamic heat pump model showed to be in an acceptable range.
Besides the dynamic simulations, the model was also used for a thermodynamic analysis of three heat pump systems. This thermodynamic analysis revealed the potential of direct exchange geothermal heat pumps over conventional ground source heat pump systems. Furthermore, it could be shown, for one particular case, that vertical direct exchange heat pumps offer the potential to be more efficient than horizontal direct exchange systems.

## ACKNOWLEDGEMENT

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