DYNAMIC MODELING OF HEAT PUMPS

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Abstract: On/off cycling is the most commonly used mode for capacity control in heat pumps. The effects of startup and shutdown on the coefficient of performance are described in this paper. Air to water and brine to water (geothermal) heat pumps were simulated using a physical model, which was validated using data from field and laboratory measurements. Subsequently, several parametric studies were carried out in order to evaluate the effects of cycle time on the performance of a heat pump system. It was shown, that on/off cycling of air-source heat pumps leads to a performance penalty of 2-5% for very short cycle times. The performance reduction is insignificant if the heat pump is run for 15 minutes or longer at a time. Geothermal heat pumps, on the other hand, profit from short cycle times. The benefit comes from the borehole regenerating during the off-time of the heat pump. Performance improvements in the area of 5% can be achieved using an intelligent control of the cycle time.

Key Words: heat pump, dynamic simulation, startup, shutoff, efficiency

1 INTRODUCTION

Currently the capacity of most small and medium scale heat pumps is adjusted to the required heat load using on/off-control (cycling). Several studies in the past have shown that cycling of heat pumps creates significant heat losses especially during the startup of the system. These losses have been indicated in field tests of approximately 300 heat pumps (Erb et al. 2004). The authors reported a reduction of 5-20% in heating capacity compared to standardized testing. Each shutoff and startup process leads to a reduction of the heat output until the asymptotic heating performance is reached. This has been demonstrated by (Ehrbar et al. 2003) in laboratory measurements and simulations.

It is important to understand the influence of the most important parameters on the heating performance and efficiency in order to improve heat pumps with respect to on/off cycling. Efficiency can then be increased using improved control algorithms for on/off controlled heat pumps. Additionally, a better understanding of the subject leads to a fair performance comparison of variable capacity heat pump units and on/off controlled systems. Therefore, a dynamic heat pump model for air-source heat pumps was developed in the first phase of this project. This dynamic model was validated with corresponding measurements (Gubser and Ehrbar 1997; Gubser at al. 1999; Hubacher and Ehrbar 2001; Shafai et al. 2000). While this model covers the fundamentals of the process, it was found that several parameters were difficult to extract from the measurements. Furthermore, numerical problems were encountered when calculating the results.

Based on the results of the first phase of the project, a new physical model was developed and partially presented by (Uhlmann and Bertsch 2010a). That paper also presents additional measurements on heat pumps that were taken in order to get a more solid database for the model verification. The aim was to achieve a stable model with only very few parameters, of which the majority can be easily deducted from component data sheets. Within the project two types of heat pumps were addressed: Air-water heat pumps, and brine-water (geothermal) heat pumps. Parametric studies were carried out using the validated heat pump models. It was shown that air-source heat pumps exhibit efficiency losses in part load behavior. In contrast, geothermal heat pumps benefit from the temperature recovery of the borehole and brine during shutoff time of the heat pump. A full report of the study including the complete description of the model and the measurement results were presented by (Uhlmann and Bertsch 2010b).

2 HEAT PUMP MODELING

All models are based on physical parameters such as mass and dimensions of the components. Most of these parameters can be extracted from component datasheets. There are only two parameters that have to be obtained from steady state measurements. These two parameters are a heat transfer enhancement factor on the condenser refrigerant side and an enhancement factor for the evaporator air side heat transfer.

The heat pump model consists of four modules. There is a module for each component; compressor, condenser, expansion valve and evaporator. Each of these dynamic component models is based on first principles of heat and mass conservation and solved by a central time step method. The overall model is solved by a forward time step method, since the time constants in the overall model are of larger magnitude than the time constants of the modules. A sensitivity analysis has been carried out, through which an optimum time step size of 0.1s was determined.

The model was implemented using the software EES (Klein, 2009). The EES software package includes all relevant information on the properties of the used refrigerants. In order to reduce calculation time, moving boundary models were used for all heat exchangers and an efficiency based model was used for the compressor. The main challenge in the modeling effort was presented by the changing refrigerant states in the heat exchanges. In the beginning this challenge led to instable systems of equations. In figure 1 the different states of the refrigerant in the condenser are shown during startup of the heat pump. After a long stoppage time usually only gaseous refrigerant can be found in the condenser. This effect can be explained by refrigerant bleeding through expansion valves and the compressors from the condenser to the colder evaporator. After a short runtime, this state switches to refrigerant being in a two-phase state and a superheated state. Finally, once the system is close to steady state we have to consider three different states of the refrigerant: Superheated, two-phase and subcooled. All these regimes that happen during the startup phase have been modeled in the present simulation model. Special care has to be taken if one of the zones in the model gets very small, the system of equations tends to get almost singular and difficult to solve. Similar issues have to be addressed for the evaporator. A small hysteresis to avoid oscillations and a central time step method for stability helped solve these issues. Due to the precise modeling of the different states of the heat pump, flooding of the evaporator and liquid surges in the compressor can be simulated.

During shutoff of the heat pump, the heat transfer coefficients were calculated using pool boiling and free convection heat transfer coefficients instead of flow boiling and forced convection. This method ensured that the model predicted startup and shutoff well, as can be seen in the following chapter.

Aim of the study was to analyze the heat pump system itself. Therefore the boundarie of the model has been drawn at the condenser (no simulation of the hydronic heat distribution system). On the low pressure side of the air source heat pump the evaporator is the system boundary since only heat pumps with direct air source have been considered. Due to the storage effect of the ground probe and the soil the system boundary in case of the geothermal heat pump system is the ground probe including the soil affected by the temperature change.

Details on the modeling effort including all parameters can be found in (Uhlmann and Bertsch 2010a) and (Uhlmann and Bertsch 2010b), since a full description of the model is beyond the scope of this paper

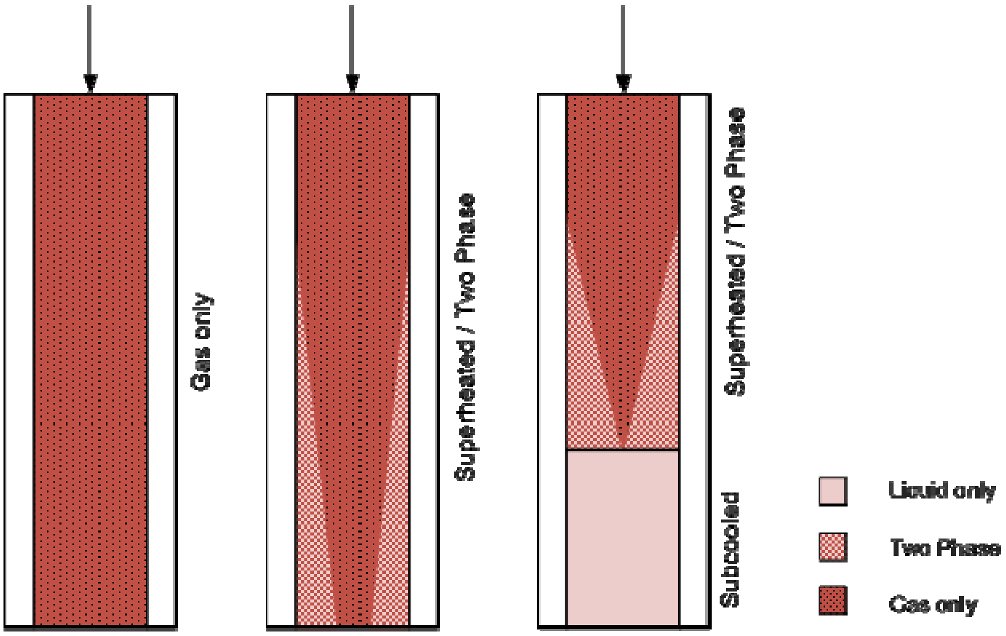

Figure 1: Possible states of the refrigerant in the condenser.

3 VALIDATION OF THE HEAT PUMP MODELS

In order to validate the air-source heat pump model, measurements on two heat pumps have been carried out at two operating points each. These measurements were carried out in a psychometric chamber where the air and water temperatures were kept within ±1 °C of the setpoint even during the startup period. Humidity was set at a very low level, such that no condensation or frosting of the heat exchangers occurred during testing. Figure 2 shows the setup of the investigated heat pump system and table 1 the according sensors and uncertainties.

Table 1. Measurement system and uncertainty

Name	*Description*	*Measurement range*	*Uncertainty*
T_x	Temperature measurement, T-type thermocouple	-20C to +120C	± 0.5C
V_x	Volume flow rate, ultrasonic Sensor	0 to 5 m^3/h	± 2 %
$\dot{Q}_{el,x}$	Electrical power, - 1 Phase, true RMS - 3 Phase, true RMS	 0 to 1.2 kW 0 to 10 kW	 ± 0.2 % ± 0.2 %
p_H	Low pressure, piezoelectric sensor	0 to 30 bar	± 1%
p_L	High pressure, piezoelectric sensor	0 to 50 bar	± 1%
φ_x	Relative humidity, capacitive sensor	20-99%	± 2%

While the simulated pressure and electricity demand matched the measured values very well, the temperature measurements showed some delay compared to the simulation, as can be seen in figure 3. The model predicts an immediate increase of the supply temperature, whereas in reality a time lag of approximately 8 seconds was observed. It was found that the temperature sensor was installed with a nonzero distance to the heat pump, and therefore, it exhibited a time delay. In addition, the sensors show PT1 behavior, since their masses have to be heated up by the water. Considering these effects and applying them to the simulation results in a reduced time lag between measurement and simulation. This modification is also shown in figure 3. For the validation of the heat flow, the corrected simulation results were used.

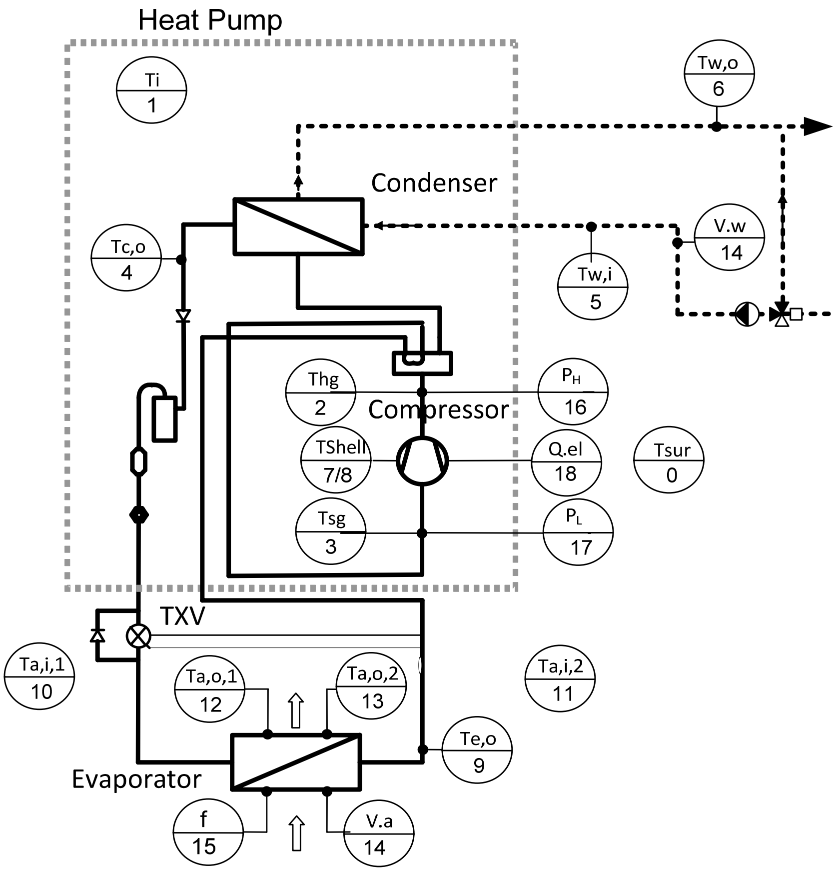

Figure 2. Test setup of the air-water heat pumps set up in the psychometric room.

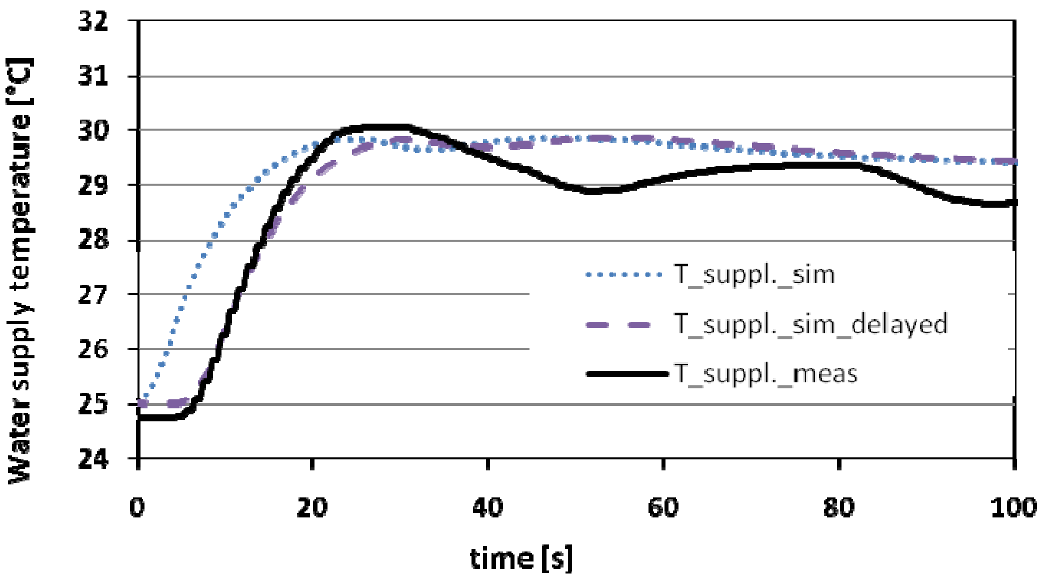

Figure 3: Measured and simulated water supply temperature.

Table 2 shows a quantitative comparison between measured and simulated coefficients of performance. The two adjustment parameters of the heat pump model were set at the operating point of Q3/S25, where Q stands for heat source and S stands for heat sink. Therefore, Q3/S25 stands for an air inlet temperature of 3°C and a water supply temperature of 25°C. The adjustment parameters were determined once, and they were held constant for all following simulations. The standard deviation between simulation and measurement was then calculated for two different time intervals. Firstly, it was calculated for the first 5 minutes of run time in order to compare the differences during startup time. Secondly, it was calculated for the first 40 minutes of run time in order to compare the steady state behavior. As detailed in table 2, the maximum standard deviation for all operating points lies under 6% with an average error of 3.5%

Table 2: Validation of the air source heat pump model.

Heat pump model	Operating point	Time interval	Standard deviation (COP)
HP1	Q3/S25	0-5 min	1.8 [%]
HP1	Q3/S25	0-40 min	3.7 [%]
HP1	Q-10/S45	0-5 min	5.6 [%]
HP1	Q-10/S45	0-40 min	2.2 [%]
HP2	Q3/S25	0-5 min	3.1 [%]
HP2	Q3/S25	0-40 min	2.9[%]
HP2	Q-10/S44	0-5 min	2.7 [%]
HP2	Q-10/S44	0-40 min	4.0 [%]

More detailed results for one operating point can be found in table 3. Here the heating capacity ($\dot{Q}_{heat}$), the electrical input power ($\dot{Q}_{el}$), and the coefficient of performance are shown for the operating point Q3/S25 using heat pump 1. The heating capacity is slightly under-predicted by the simulation, and the electrical power demand is slightly higher. The result is a total error of 1.8% in heating capacity and of 3.7% in electrical power demand.

Table 3: Simulation and measurement results for heat pump 1 at the operating point Q3/S25.

Parameter	Measured value	Simulated value	Simulation (delayed)	Error (delayed)
$T_{sink,\ out,5min}$	29.0 [C]	29.3 [C]	29.2 [C]	0.2 [C]
$T_{sink,\ out,40min}$	29.2 [C]	29.3 [C]	29.3 [C]	0.1 [C]
$T_{sink,\ in,5min}$	24.9 [C]	25.0 [C]	25.0 [C]	0.1 [C]
$T_{sink,\ in,40min}$	24.9 [C]	25.0 [C]	25.0 [C]	0.1 [C]
$\dot{Q}_{heat,5\min}$	9.6 [kW]	10.8 [kW]	9.9 [kW]	-3.1 [%]
$\dot{Q}_{heat,40\min}$	9.7 [kW]	10.0 [kW]	9.6 [kW]	-1.0 [%]
$\dot{Q}_{el,5\min}$	1.83 [kW]	1.89 [kW]	1.89 [kW]	3.3 [%]
$\dot{Q}_{el,40\min}$	1.83 [kW]	1.89 [kW]	1.89 [kW]	3.3 [%]
$COP_{5\min}$	5.3[-]	5.7[-]	5.2[-]	-1.8[%]
$COP_{40\min}$	5.3[-]	5.3[-]	5.1[-]	-3.7[%]

A sensitivity analysis showed that a small difference between steady state values shows very little impact on the results of the parametric study. Therefore the air-source heat pump model was considered valid for further use in the parametric study.

The validation of the geothermal heat pump proved more complicated, since a geothermal probe is difficult to measure in a laboratory experiment. Therefore, a heat pump system in the field was equipped with a data acquisition unit, and it was measured for several months. Measurements were taken at intervals of 5 seconds after startup and then at a rate of 1 minute after steady state was reached. This precaution helped manage the volume of data and to still provide sufficient information about the startup period.

After creating a model for the geothermal heat pump and the corresponding geothermal probe, the simulation was compared to the measurement data again. Table 4 shows the results of the comparison for heat capacity, electrical demand and coefficient of performance. The model of the geothermal heat pump matches the measurement values closely. Therefore the geothermal heat pump model can also be used for a parametric study.

Table 4: Simulation and measurement results for the geothermal heat pump.

Parameter	Measured value	Simulated value	Simulation (delayed)	Error (delayed)
$\dot{Q}_{heat,5\,\mathrm{min}}$	9.6 [kW]	10.8 [kW]	9.9 [kW]	-3.1 [%]
$\dot{Q}_{heat,40\,\mathrm{min}}$	9.7 [kW]	10.0 [kW]	9.6 [kW]	-1.0 [%]
$\dot{Q}_{el,5\,\mathrm{min}}$	1.83 [kW]	1.89 [kW]	1.89 [kW]	3.3 [%]
$\dot{Q}_{el,40\,\mathrm{min}}$	1.83 [kW]	1.89 [kW]	1.89 [kW]	3.3 [%]
$COP_{5\,\mathrm{min}}$	5.3[-]	5.7[-]	5.2[-]	-1.8[%]
$COP_{40\,\mathrm{min}}$	5.3[-]	5.3[-]	5.1[-]	-3.7[%]

4 PARAMETRIC STUDY OF AN AIR-SOURCE HEAT PUMP

Two parametric studies were carried out, both for the air-source and the geothermal heat pump. In the first study for air-source heat pumps, the cycle time was fixed at 120 minutes. During this period, the heat pump was cycled as shown in figure 4. This cycle was then repeated three times. The repetition was necessary since the heat pump system was completely cooled off at the first startup.

The ratio of run time to total cycle time, the so called duty cycle was then varied from 10% to 100%, where 100% corresponds to steady state operation. The results of this study are presented in figure 5. It can be seen that the coefficient of performance in the first run is lower than that for the second and third runs. The difference between runs 2 and 3 is small, supporting the conclusion that the influence of the complete cool-off before the initial startup is remedied after the first run. We can see a decrease of the coefficient of performance at low duty cycles. This is expected due to the heat loses to the surrounding during the shutoff time. Also, there is a considerable movement of refrigerant charge from the condenser to the evaporator during standstill of the heat pump. This charge then has to be recompressed after startup. We can also see that the reduction of performance at low run time ratios is approximately 5%. This means that the reduction of efficiency compared to steady state operation is small. Residential sized air-source heat pumps generally do not exhibit significant performance losses as long as the run time is longer than 10 – 15 minutes.

At first view it might be astonishing, that the efficiency at 100% run time is smaller than at 90% runtime. This phenomenon is explained by the reduced condensing pressure and the increased evaporating pressure due to thermal storage of the heat exchangers. The compressor needs less work to overcome the smaller pressure ratio after a brief shutoff period. This effect is only seen at very high run time ratios; at low run time ratios the heat losses outweigh the heat storage effect.

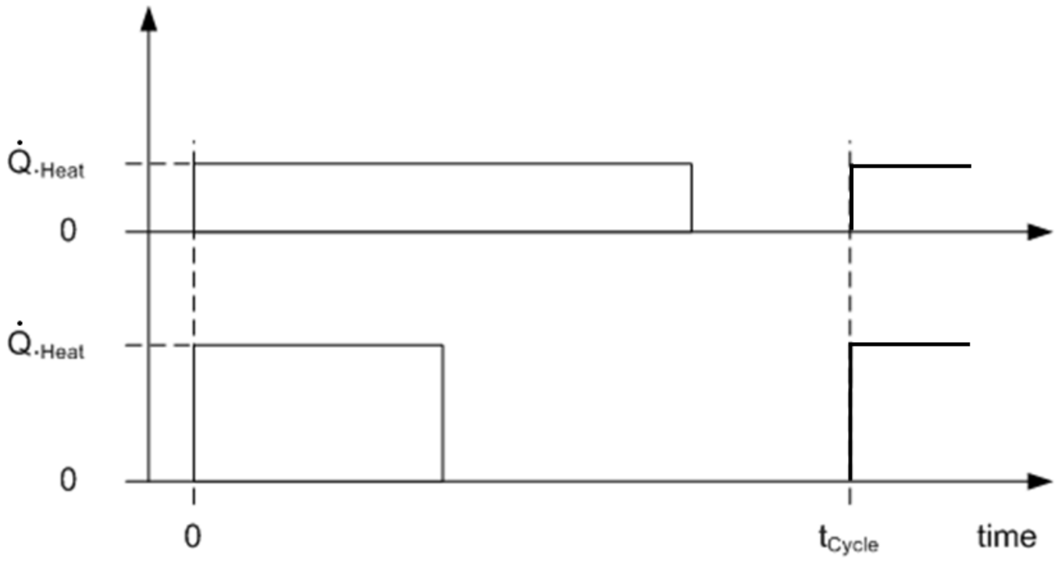

Figure 4: Two ways of achieving the same heating energy. Top: longer run time with lower heating capacity. Bottom: Short run time with higher capacity.

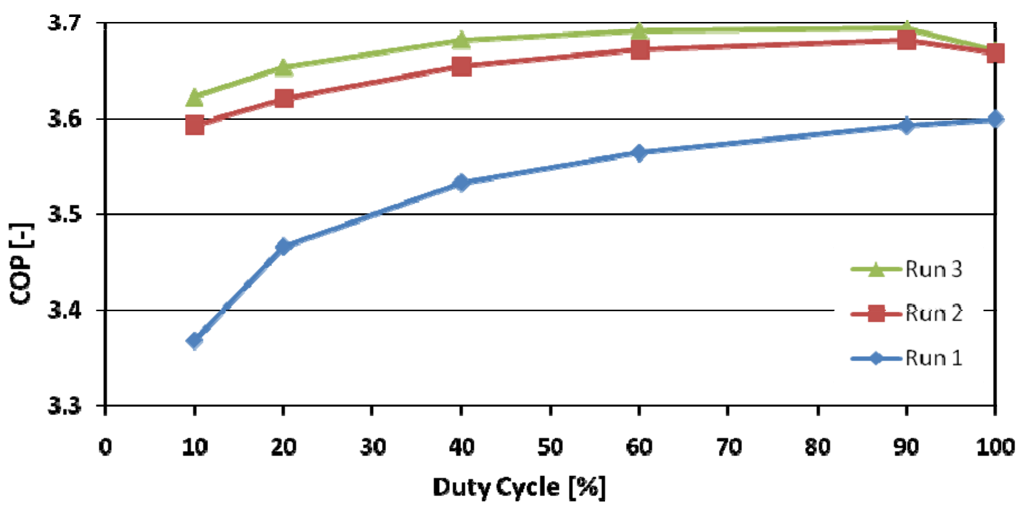

Figure 5: Coefficient of performance of the air source heat pump with respect to run time ratio.

The second parametric study on air-source heat pumps was carried out as follows: The duty cycle was kept constant at 40%, and the cycle time was varied from 30 to 270 minutes. Therefore, the run time ratio is constant but distributed in a different way in each simulation. The distribution of the run time is shown in figure 6. On one end there is a short run time combined with a short shut off time, on the other end the heat pump is running for a long time followed by an equally long shutoff. Again three runs were simulated in order to achieve a fair comparison.

The results are shown in figure 7. Again the first run shows a lower performance due to the initial startup condition. Runs 2 and 3 show a very similar result. The performance of the heat pump system is reduced by a few percentage points at short cycle times. This effect affirms the conclusion of the first parametric study. Air-source heat pumps suffer performance losses at short run times, but the losses are small as long as a minimum run time of 10 – 15 minutes is maintained.

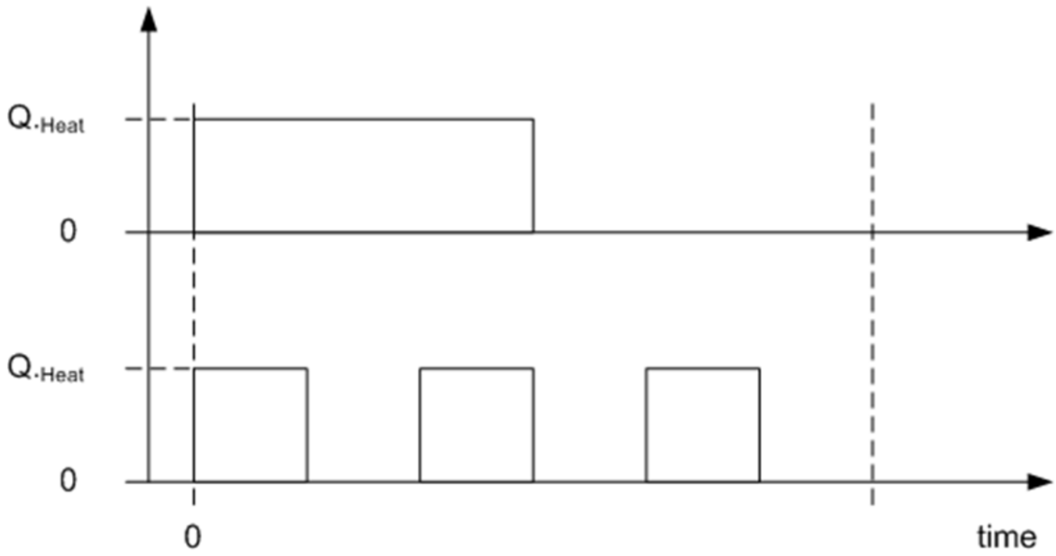

Figure 6: Pulse width modulation in order to achieve necessary heat demand. Top: Long cycle time, Bottom: short cycle time with same duty cycle.

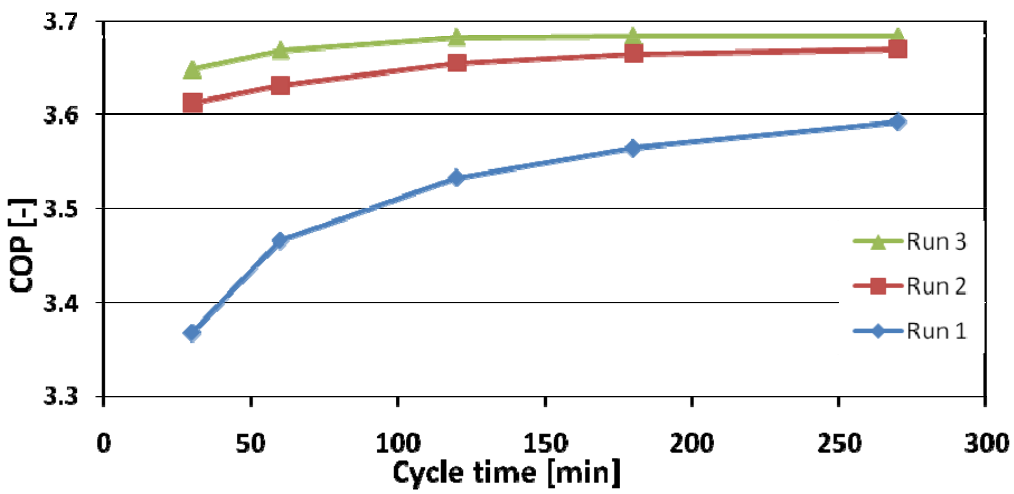

Figure 7: Coefficient of performance of the air-source heat pump with respect to cycle time.

In all simulaitons presented above, the water supply pump on the condensor is turned off together with the heat pump. An easy method to reduce heat losses during standstill of the heat pump is to run the supply pump for approximately one more minute after shutoff. The performance degradation at short run time ratios can be reduced in this way.

5 PARAMETRIC STUDY OF A GEOTHERMAL HEAT PUMP

Similar to the parametric study of the air-source heat pump, the effect of the duty cycle of the heat pump on the coefficient of performance has been evaluated. In this case, the geothermal heat pump model was used to simulate two runs with varying run time ratios at a cycle time of 90 minutes. Figure 8 presents the improvement of the coefficient of performance at short run times compared to steady state operation. It was shown that the performance increases at decreasing duty cycles. This effect can be explained through consideration of the heat transfer mechanisms in a geothermal borehole. The brine leaving the evaporator and entering the borehole is cooled off due to the extraction of heat. Due to the length of time for the fluid to circulate the borehole heat exchanger, the brine temperature at the evaporator inlet experiences the temperature drop only after several minutes. Therefore, the evaporating temperature after startup stays high for several minutes. The heat pump works more efficiently during this time. The performance drops only after several minutes of operation.

Once the heat pump is turned off, the borehole can regenerate and provide a high heat source temperature again for the next run time. Depending on the properties of the ground, this effect can be severe. A dry borehole with a heat load of approximately 45 W/m borehole-length was used in the model presented here.

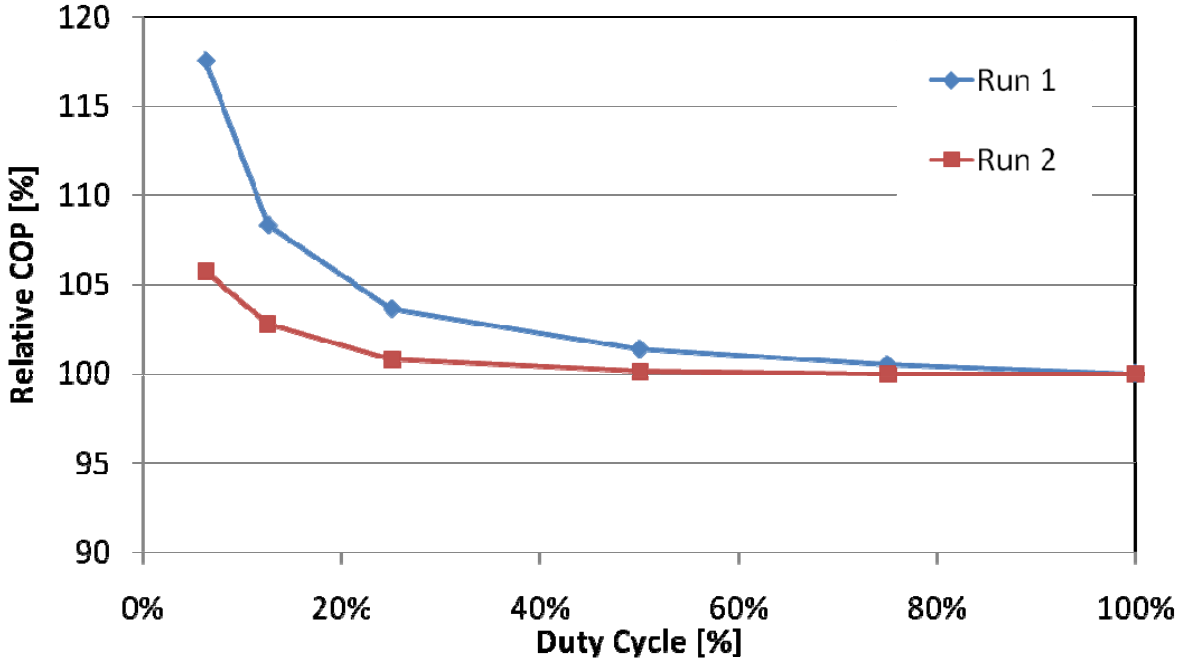

Figure 8: Coefficient of performance of the geothermal heat pump with respect to run time ratio.

Considerable energy savings can be achieved combining the effects of the regenerating borehole and an appropriate control for the water supply pump.

CONCLUSIONS

A dynamic model for air source and geothermal heat pumps was developed and verified against laboratory and field measurements. The model is based on first principles and all but two model parameters can be extracted from component datasheets. The models show good agreement (max 5.6% error) to laboratory and field measurements on heat pumps.

In several parametric studies it was shown that air-source heat pumps experience performance losses in the range of 2-5% at short run times. These heat losses to the ambient can be reduced by extending the run time of the water supply pump for approximately one minute. In general, the performance degradation is insignificant for run times longer than 10 – 15 minutes.

In the case of geothermal heat pumps, the performance of the system increases with a decreasing duty cycle. This effect can be attributed to the regeneration of the borehole during shutoff time. It is possible to gain up to 5% efficiency in the coefficient of performance for dry boreholes. For wet boreholes it is expected that the performance increase is higher, due to shorter regeneration times.

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