

A variable speed water-to-water heat pump model used for ground-source applications

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The main objective of this study is to model variable speed water-to-water heat pumps (*VSHP*) and to examine the impact of the operation of such devices on ground heat exchanger sizing and energy consumption when they are used in ground-source applications.

In the first part of the paper, a complete physics-based steady-state model of a variable-speed water-to-water heat pump is briefly presented. A performance map approach is also used by modifying an existing TRNSYS variable speed heat pump model to provide a minimum speed of operation and a better representation at part load.

Simulation results over a heating season on a residential ground source *VSHP* indicate that the energy coverage (i.e. percentage of annual heat supplied by the heat pump) increases at a faster rate than the effect coverage (i.e. percentage of peak building heat supplied by the heat pump) for a small value of the effect coverage. For example, for an effect coverage of 60%, the energy coverage is $\sim 93\%$. It is also shown that the normalized length of the ground heat exchanger varies linearly up to an effect coverage of 60% where it is equal to 75% of the value encountered for an effect coverage of 100%.

Keywords: water-to-water heat pump; variable speed compressor; performance map, modelling

INTRODUCTION

Heat pump system models are valuable for estimating carbon emissions and energy costs, determining appropriate heat pump and ground heat exchanger (GHE) sizing, and assessing the effectiveness of advanced cycles (Underwood 2016). In this study, the emphasis is on variable speed operation which, in principle, should improve efficiency and performance (Chua, Chou, and Yang 2010).

Variable speed heat pumps (*VSHP*) offer the possibility of matching the building heating and cooling loads and improving thermal comfort by modulating the compressor speed. This is accomplished by tuning the inlet frequency (typically down to 15-30 Hz and up to 90-120 Hz) to the electric motor of the compressor. Contrary to fixed speed compressor heat pumps (*FSHP*) which operate in on/off mode, *VSHP* operate continuously except for low load (speed) conditions. At low speeds, the oil may not circulate properly in the compressor, leading to lubrication issues (Ossorio and Navarro-Peris 2021). In practical operating conditions, variable speed compressors usually have a minimum rotational speed, which corresponds typically to an inlet frequency between 15-30 Hz.

Relatively few studies have been conducted on the modeling of water-to-water heat pumps, and even fewer studies exist on those equipped with variable-speed compressors. Stefanuk, Aplevich, and Renksizbulut (1992) presented a steady-state simulation model of a water-to-water heat pump operating under superheat control. The model is derived from the basic conservation laws of mass, energy, momentum, and equations of state, as well as fundamental correlations of heat transfer. It is used to predict system performance over the full operating range of a heat pump. The results indicate that refrigerant superheats at the evaporator exit and that the refrigerant charge can be used as a control variable to maximize system performance.

Jin and Spitler (2002) presented a steady-state simulation model for a water-to-water reciprocating vapor compression heat pump, intended for use in building simulation programs. The model was developed from basic thermodynamic principles and heat transfer relations. It comprises several unspecified parameters that are estimated from catalog data using a multi-variable optimization procedure.

A residential-size ground-source variable-speed integrated heat pump was developed by Rice et al. (2013). The 2-ton (7 kW) unit has multiple functions such as space cooling, space heating, dedicated water heating, and simultaneous space cooling and water heating. A vapor compression simulation model in TRNSYS was calibrated for each of the four primary modes of operation using laboratory test data. The model was used to predict annual energy savings for five US locations. Their study indicates that *VSHP* offers average savings of 55% when compared to a typical air source heat pump (*ASHP*).

Bouheret and Bernier (2018) have proposed an approach to model a commercially available variable capacity water-to-air ground-source heat pump (ClimateMaster, 2022) that can provide heating, cooling, and domestic hot water (DHW). They showed that the annual seasonal performance factor (i.e. ratio of the annual energy requirement for heating, cooling, and DHW over the annual electricity consumption) can reach 3.64.

Ndiaye (2007) focused on the development of a dynamic model of a water-to-air heat pump to enable more accurate simulations. A literature review revealed that the existing models found in energy simulation programs are inadequate in reproducing the transient effects associated with cycling. The study uses an iterative algorithm that combines individual models of the various components to create a global model. The study also finds that the time constant used for modeling the start-up capacity is dependent on the operating conditions, which is a weakness of models found in simulation programs that use a fixed time constant.

The DOE/ORNL Heat Pump Design Model (HPDM) (Rice 2001) is a widely used software tool for designing heat pump systems. The HPDM uses simplified assumptions, such as steady-state operation, uniform indoor and outdoor temperatures, and in some cases ideal components. It is not easily adaptable to perform annual simulations.

VSHP used in ground-source applications will experience ground loads that are different than the ones of a *FSHP*. This will impact the required size of the ground heat exchanger (*GHE*) and heat pump energy consumption. The study by Rice et al. (2013) indicates that the required length of the *GHE* was 25% higher than a ground-source heat pump equipped with a desuperheater for heating domestic hot water.

Viviescas and Bernier (2022) compared the required *GHE* length and seasonal performance factors (*SPF*) associated with *FSHP* and *VSHP* with various effect coverage (percentage of peak load covered by the heat pump). Results indicate that a *VSHP* with a 100% effect coverage requires a *GHE* that is 40% longer than a typical *FSHP* with a 90% effect coverage. However, the *SPF* is much better for the *VSHP* (4.14 vs. 3.11 in favor of the *VSHP*).

Based on this review there is an apparent lack of detailed models of variable-speed water-to-water heat pumps. The objective of this paper is thus to develop such a model. The modeling of the evaporator and condenser (plate heat exchangers) has been presented elsewhere (Viviescas et al. 2021). Emphasis is put in this paper on the prediction of the compressor behavior and the solution of the complete system of non-linear equations. Improvements to TRNSYS's Type 1323_v2a, which uses a performance-based approach, are also proposed. Finally, preliminary results presented by Viviescas and Bernier (2022) are expanded to present the impact of *VSHP* operation on the energy coverage and *GHE* length for a range of effect coverage.

DETAILED MODEL

As stated earlier, this study aims to model a water-to-water heat pump featuring a variable speed compressor using two approaches: one based on detailed modeling of the various cycle components and the other based on performance maps generated from the complete detailed model.

The detailed *VSHP* model presented here is part of a broader study aimed at developing a detailed physics-based and experimentally validated model of a water-to-water heat pump. Initial steps towards this goal were documented recently (Viviescas et al. 2021).

The *VSHP* being modeled is shown schematically in Figure 1. The objective of the model is to predict the capacity (heating or cooling) and the required power for known inlet conditions (temperature and flow rate) on the load and source sides (i.e. $T_{wL\text{in}}$, $\dot{m}_{wL}$, $T_{wS\text{in}}$, $\dot{m}_{wS}$) as well as compressor speed.

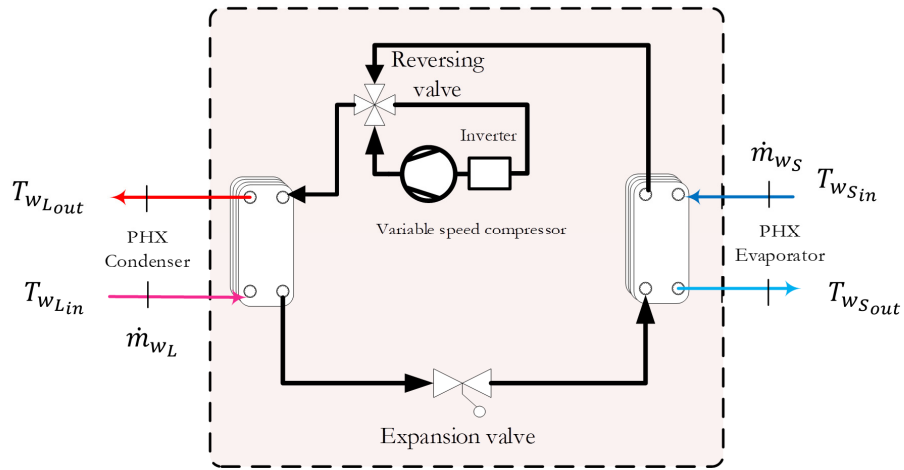


Figure 1 Schematic of the VSHP

Plate heat exchangers (*PHX*) are used in the evaporator and condenser. A variable speed motor (inverter in Figure 1) regulates the compressor speed and a thermostatic expansion valve controls the level of superheat at the evaporator exit. Individual models for these components are linked together via the refrigeration cycle.

The corresponding pressure-enthalpy ($P - h$) diagram is shown in Figure 2 where $PHXs$ are superimposed to show the heat exchanged with secondary fluids at the evaporator and condenser. The refrigerant (R-410A in the present case) leaves the evaporator as a superheated vapor at a temperature T_1 and exists the compression process at T_2 . In the condenser, the refrigerant is first de-superheated from T_2 to $T_{x,2}$ and then condensed down to a saturated liquid at T_3 . Then, there is an isenthalpic pressure drop in the expansion valve and the refrigerant enters the evaporator at T_4 . In the evaporator, the refrigerant undergoes evaporation from T_4 to $T_{x,1}$ and is then superheated up to T_1 .

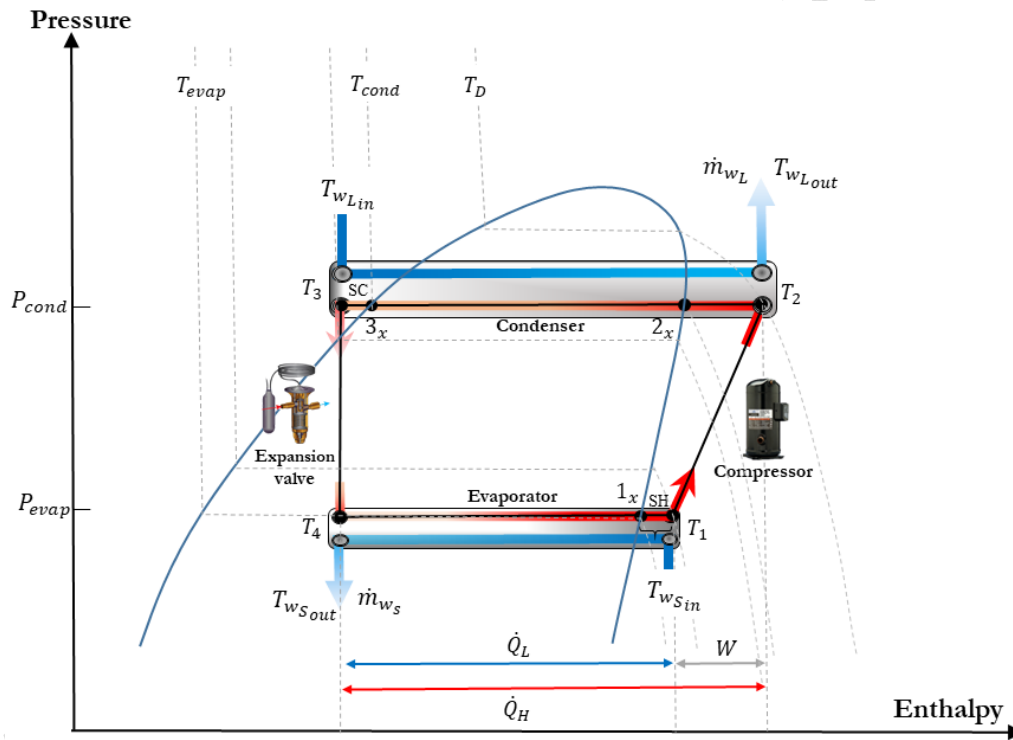


Figure 2 P-h diagram

Steady-state operation is assumed and pressure drops in both heat exchangers and in the reversing valve are assumed to be negligible. There are no heat losses between the heat exchangers and the surrounding environment. Finally, the influence of circulating oil is considered to be negligible.

Detailed heat transfer models are used for both *PHX*s and are documented by Viviescas et al. (2021). A so-called moving boundary approach is used (Sarfraz, Bach, and Bradshaw 2018) where the *PHX*s are divided into separate zones (e.g., desuperheating, condensation, and subcooling zones in the condenser) that represent both single-phase and two-phase regions. Initially, the fractions of the total heat exchanger area used for each zone are unknown and are assigned suitable guess values. Then an iterative process, encompassing a system of nonlinear equations (energy balances and epsilon-NTU equations in each zone) is employed to determine the heat exchanger area used for each zone for a given set of inlet conditions (temperature and flow rate) of the secondary fluids and refrigerant flow rate determined by the expansion valve model.

The overall heat transfer coefficient in each zone is required to obtain the NTU value. It is determined based on the sum of three thermal resistances associated with heat transfer coefficients on the water and refrigerant sides and the plate's thermal conductivity and thickness. Heat Transfer coefficients for single-phase and two-phase flows are obtained using correlations found in the literature (Viviescas et al. 2021).

Two improvements have been implemented in the overall heat pump model since the earlier work of Viviescas et al. (2021). First, the refrigerant can exit the condenser as a subcooled liquid (determined by the *PHX* models) rather than in a fixed condition as a saturated liquid. Secondly, the heat exchangers can be modeled in both counterflow and parallel modes which implies that the cooling mode can now be modeled.

In summary, the heat exchanger models can predict the outlet refrigerant temperatures (T_1 and T_3) for given values of the secondary inlet fluid temperature fluids (T_{wsin} and T_{wLin}) and flow rate ($\dot{m}_{ws}$ and $\dot{m}_{wL}$) and inlet refrigerant temperatures (T_4 and T_2) provided by the compressor and expansion valve models. The variable speed compressor requires special treatment and the

next paragraphs describe the approach used here.

Researchers have proposed various approaches to model variable speed compressors. Winandy, Saavedra, and Lebrun (2002) proposed a semi-empirical gray-box model, known as the Winandy model, for forecasting scroll compressor performance. The model includes an isothermal wall that heats the suction gas, cools the discharge gas, absorbs electro-mechanical losses, and transfers heat to the ambient environment. The compression process is divided into two stages: first, isentropic compression up to an adjusted pressure, followed by adiabatic and isochoric compression up to the discharge pressure.

The Shao model (Shao et al. 2004) is a black-box model that incorporates modulation by using performance data at various operational frequencies. To fit the variable speed data, the model employs a second-order function to the mass flow rate and power ratio. These ratios establish a relationship between the mass flow rate and power at the base frequency and performance at different frequencies.

The Popovic and Shapiro model (Popovic and Shapiro 1995) is a semi-empirical compressor model derived to predict reciprocating compressor performance. The model utilizes an idealized polytropic compression and clearance volume re-expansion with constant pressure suction and discharge processes. This cycle is modified to include phenomena typical of a compressor. This includes utilizing volumetric efficiency, based on a clearance factor taken as an unknown. The authors then add suction and discharge pressure drop and, for simplicity, set the magnitudes equal. The model utilizes the thermodynamic work rate of a polytropic process as a basis for compressor power requirements.

A recent study by Gabel and Bradshaw (2022) concludes that the AHRI model with ten coefficients performs exceptionally well when trained with data sets in both standard and

variable superheat scenarios, with errors of only 0.43% and 0.57% for mass flow rate and power, respectively. Similar findings regarding training data performance were also reported by Aute, Martin, and Radermacher (2015) in their analysis of the AHRI model. However, the model is not recommended for data extrapolation and is superior to the Winandy, Shao, and Popovic models only within the range of data used for model training.

In the present work, the compressor is modeled using an extension of the approach suggested in standard CAN/ANSI/AHRI 540-2020 (AHRI 2020). This standard proposes to fit fixed-speed experimental data into a third-order polynomial equation with 10 coefficients relating either the mass flow rate or power input to T_D , the discharge dew point temperature (i.e. condensing temperature corresponding to saturation at $x=1$), and to T_S , the suction dew point temperature (i.e. evaporating temperature corresponding to the saturation line).

The AHRI model has been customized for variable speed compressors by Danfoss using a second-degree polynomial (Equation 1) with three variables and 30 coefficients: the two AHRI standard temperatures (T_D and T_S) and the operating frequency S (in Hz):

$$X = C_0 + C_1T_s + C_2T_D + C_3T_s^2 + C_4T_D^2 + C_5T_sT_DS^2 + C_6T_s^2T_DS^2 + C_7T_sT_D^2S^2 + C_8T_sT_DS + C_9T_s^2T_DS + C_{10}T_sT_D^2S + C_{11}T_sT_DS + C_{12}T_s^2T_D + C_{13}T_sT_D^2 + C_{14}T_s^3 + C_{15}T_D^3 + C_{16}S + C_{17}T_sS + C_{18}T_DS + C_{19}T_s^2S + C_{20}T_D^2S + C_{21}T_s^3S + C_{22}T_D^3S + C_{23}S^2 + C_{24}T_sS^2 + C_{25}T_DS^2 + C_{26}T_s^2S^2 + C_{27}T_D^2S^2 + C_{28}T_s^3S^2 + C_{29}T_D^3S^2 \quad (1)$$

Where X is either the mass flow rate or the power consumption. For this study, the Danfoss VZH028-CJ compressor was selected (Danfoss 2022). When operating at 60 Hz it has a heating capacity of 10.5 kW under rated conditions ($T_{w_{sin}} = 4^\circ\text{C}$ and $T_{w_{Lin}} = 35^\circ\text{C}$), and 15.33 kW when the frequency is increased to 100 Hz. The coefficients for this compressor are presented in Table 1 and are valid for a superheat of 5 °C.

Table 1 Input coefficients to equation 1 for Danfoss VZH028-CJ

Coefficient	Power [W]	Mass-flow [kg/h]	Coefficient	Power [W]	Mass-flow [kg/h]
C ₁	3.798E-01	-3.385E-01	C ₁₆	-8.656E-07	-6.286E-08
C ₂	1.219E-02	-8.774E-03	C ₁₇	-1.659E-02	6.610E-02
C ₃	-1.107E-02	7.639E-03	C ₁₈	-1.024E-03	1.556E-03
C ₄	3.360E-04	-6.869E-05	C ₁₉	1.013E-03	-2.383E-04
C ₅	1.791E-04	-5.442E-05	C ₂₀	-2.246E-05	1.224E-05
C ₆	-1.606E-07	2.317E-08	C ₂₁	-1.214E-05	8.866E-07
C ₇	-2.230E-09	6.884E-11	C ₂₂	-1.725E-07	5.900E-08
C ₈	1.376E-09	7.864E-11	C ₂₃	6.137E-08	3.348E-09
C ₉	3.163E-05	-3.597E-06	C ₂₄	1.610E-04	-8.453E-05
C ₁₀	3.516E-07	-8.793E-09	C ₂₅	4.633E-06	-2.416E-06
C ₁₁	-2.288E-07	-6.965E-09	C ₂₆	-3.753E-06	1.615E-06
C ₁₂	-4.840E-04	1.260E-04	C ₂₇	1.186E-07	-2.200E-08
C ₁₃	-6.353E-06	2.209E-07	C ₂₈	6.308E-08	-5.057E-09
C ₁₄	4.191E-06	-1.238E-07	C ₂₉	1.115E-09	-5.063E-11
C ₁₅	2.998E-06	-2.324E-07	C ₃₀	-3.294E-10	-3.751E-11

It should be noted that the degrees of superheat are not always equal to 5°C. The compressor power consumption is unaffected by variations in the degrees of superheat as stated in the AHRI 540 standard. However, the mass flow rate needs to be corrected and the equation contained in Appendix D of the AHRI standard (AHRI, 2020) is used. In these cases, the rated refrigerant mass flow rate at rated superheat, $\dot{m}_{rated}$, is corrected to obtain the corrected mass flow rate at the real suction conditions, $\dot{m}_{corrected}$, using equation (2):

$$\dot{m}_{corrected} = \left\{ 1 + F_v \left[\left(\frac{v_{rated}}{v_{corrected}} \right) - 1 \right] \right\} \cdot \dot{m}_{rated} \quad (2)$$

In this equation, F_v is the volumetric efficiency correction factor. It varies based on the volumetric efficiency of the compression technology used. The standard suggests a value of one (1) as an approximation. The other terms in Equation 2 are $v_{corrected}$, the specific volume at the suction condition, and v_{rated} , the specific volume at the rated condition.

The compressor portion of the model obtained using equations 1 and 2 with the coefficients in Table 1 was compared to the manufacturer's data measured under AHRI standard conditions with a superheat of 11.1°C. The results of this comparison are depicted in Figures 3 and 4. The

maximum error in power is 0.15% with an RMSE of 0.077 kW, while the maximum error in mass flow rate is 0.65% with an RMSE of 0.011 kg/h. It should be noted that errors of up to 5% in mass flow rate prediction are obtained if the correction indicated in Equation 2 is not applied.

The compressor power and refrigerant mass flow rate given by equation 1 are coupled to the rest of the components of the model resulting in a set of nonlinear equations. These equations are implemented in TRNSYS to form a stand-alone model (called a Type in TRNSYS). Thermophysical and thermodynamic properties are evaluated with CoolProp (Bell et al. 2014). The modified Powell method (Okawa. 2020) was implemented to solve the system of nonlinear equations within the TRNSYS Type.

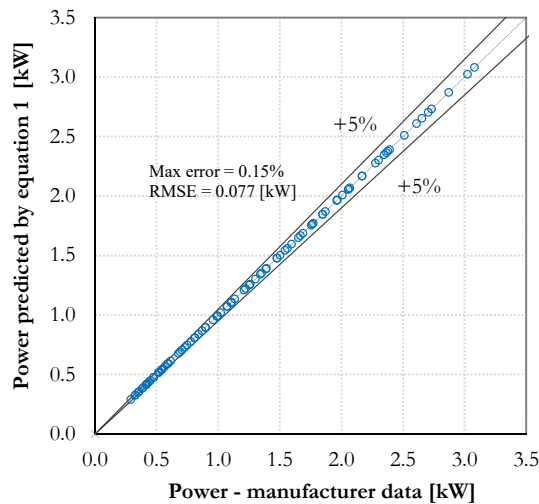


Figure 3 Power predicted by Equation 1

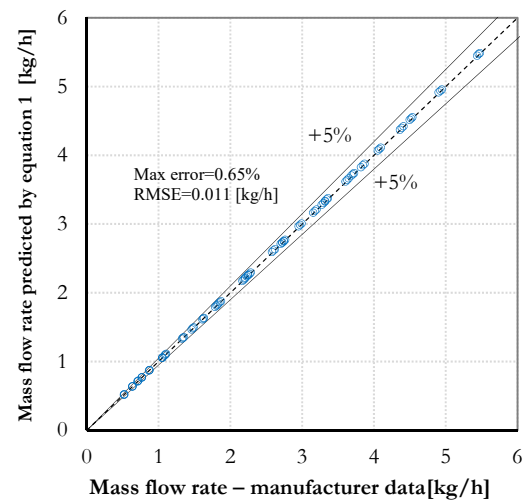


Figure 4 Mass flow rate predicted by Equation 1

Solving a system of nonlinear equations involving refrigerants that undergo phase changes can be challenging if guess values are not properly selected. In this study, the solution from the previous time step is used as guess values for the current time step. This approach usually does not lead to convergence problems during continuous heat pump operation as conditions do not change significantly from one time step to the next. However, when the model is used in long-

term simulations with on-off periods, conditions of the secondary fluids can change significantly and guess values from the previous time step are inadequate which leads to unrealistic results making the TRNSYS model inadequate for long-term simulations.

In parallel to the development of the new TRNSYS Type, the individual component models were also implemented and linked together in the Engineering Equation Solver (EES) environment which provides easily adaptable guess values and a powerful internal equation solver. Some results of the complete EES model are shown in Figure 5 which illustrates how the coefficient of performance (COP) in heating mode varies with compressor speed at four different source temperatures (5, 10, 15, and 20 °C) and three load-source temperature differences (20, 25, and 30 °C). The figure highlights several notable observations. First, for identical temperature differences, higher source temperatures result in higher COP. This is because refrigerant density is increasing at the compressor suction leading to a corresponding increase in mass flow rate. Second, the COP curves exhibit a maximum value that is not necessarily coincident with full capacity (i.e. at maximum speed). For example, for a $\Delta T = 20$ °C, the COP is maximum at 40 Hz and it decreases when the frequency is either increased or decreased. This is primarily because the compressor used in the present study reaches its maximum isentropic efficiency at speeds other than the maximum speed.

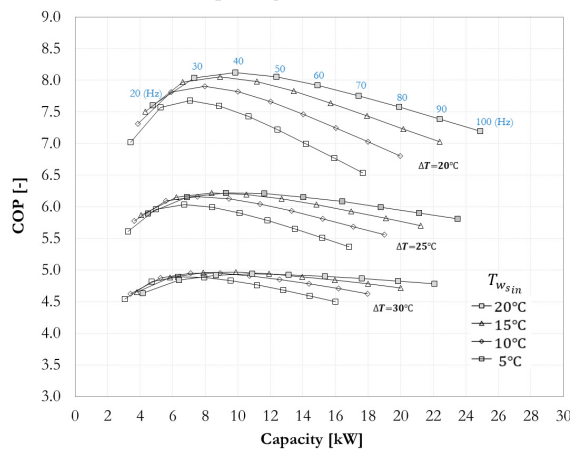


Figure 5 COP as a function of heating capacity for various operation conditions

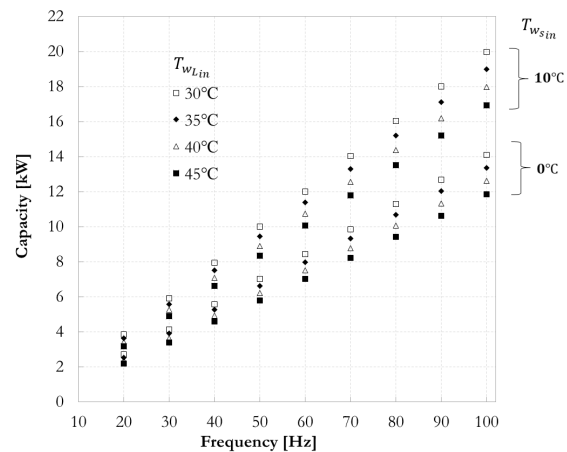


Figure 6 Heating capacity as a function of frequency for various operating conditions

Figure 6 shows the heating capacity for eight (8) combinations of $T_{w_{Lin}}$ and $T_{w_{Sin}}$, and nine (9) frequencies. As expected, the heating capacity increases as the temperature difference between the source and load decreases for a given frequency. For given values of $T_{w_{Sin}}$ and $T_{w_{Lin}}$ the heating capacity increases linearly with frequency. This may seem surprising considering that equation 1 has quadratic and cubic terms for frequency. However, statistical analysis revealed that these terms are significant for power prediction but have a negligible influence when mass flow rates are calculated.

Running annual system simulations is impractical in EES as it would require developing a specific model for the building, the GHE, etc. Faced with a possible non-converged solution for the TRNSYS type presented earlier and the impracticability of using EES in annual simulations, it was decided to use the complete EES model to develop performance maps that could be used in TRNSYS to perform annual simulations. In essence, performance maps present a solution to the complete model for a fixed set of inlet conditions.

PERFORMANCE MAP MODEL

In this section, a performance map model is developed for TRNSYS applications. TRNSYS is used to simulate the performance of energy systems including heat pump systems. It has several heat pump models for fixed and variable speed heat pumps with either air or water as the source/sink fluids. Most of these models use performance maps extracted from catalog data to predict capacity and power usage. The maps can be non-dimensionalized to represent a family of heat pumps with similar characteristics but different sizes (Hackel 2008).

The existing TRNSYS Type1323_v2a served as the basis for developing the new model based on performance maps. Type1323_v2a models a variable-speed water-to-water heat pump that can operate from zero to maximum capacity to match the desired load-side outlet temperature.

However, as indicated earlier, most variable-speed compressors have a minimum operational speed and thus a minimum capacity. In addition, Type 1323_v2a assumes that the normalized power consumption (also known as the Energy Input Ratio as a Function of Part Load Ratio – *EIRFPLR*) is only a function of the part load ratio (*PLR*), i.e. the ratio between the required capacity over the nominal capacity. However, as will be shown shortly, the *EIRFPLR* is also a function of the temperature difference between the inlet temperatures of both secondary fluids.

These two shortcomings are corrected in this work and a new Type is created (Type 1328). Table 2 displays the main features provided by the new Type 1328 compared to Type 1323_v2a. As shown in this table, Type 1328 requires four performance maps to obtain the capacity (heating and cooling) at the minimum and maximum frequencies. In addition, a series of coefficients (different in heating and cooling) are required to obtain the *EIRFPLR*.

Table 2 Characteristics of Type 1323_v2a and Type 1328

Main differences		
	Type1323_v2a (current TRNSYS type)	Type1328 (New Type)
Operation	Continuous operation at all partial load ranges	Continuous operation between minimum and maximum recommended frequencies
Parameters	Three performance data files : 1. Performance map at max. freq. in heating 2. Performance map at max. freq. in cooling 3. <i>EIRFPLR</i> ratio as a function of <i>PLR</i>	Four performance data files: 1. Performance Map (at max. speed) in heating 2. Performance Map (at max. speed) in cooling 3. Performance Map (at min. speed) in heating 4. Performance Map (at min. speed) in cooling
	The same <i>EIRFPLR</i> ratio in cooling and heating	$EIRFPLR = a_0 + a_1\Delta T + a_2\Delta T^2 + a_3PLR + a_4PLR^2 + a_5PLR\Delta T$ a_i coefficients are different in heating and cooling
Similarities		
Inputs	Inlet load and source temperatures of secondary fluids $[T_{wsin}, T_{wLin}]$ Load and source mass flow rate of secondary fluids $[\dot{m}_{wL}, \dot{m}_{wS}]$ Desired outlet load temperature in cooling and heating $[T_{wLout}]$ A control signal (0 or 1) for heating or cooling	
Parameters	Rated capacity and power in cooling and heating $[Q_{ref}, P_{ref}]$ Load and source specific heat and density of secondary fluids $[C_{p_{wL}}, C_{p_{wS}}, \rho_{wL}, \rho_{wS}]$	
Outputs	Water outlet Load and Source Temperatures $[T_{wsout}, T_{wLout}]$ Heat transfer to load, to source, and power consumption in heating and cooling mode Coefficient of performance (COP)	

Tables 3 to 6 show these performance maps at maximum and minimum frequencies (100 and 15 Hz in the present study) in heating and cooling obtained from the full set of data (Appendix A). These performance maps were obtained using the EES version of the complete model. For this study, the mass flow rates of the secondary fluids were kept constant (at 0.5 kg/s) to limit the number of independent variables. Data are normalized based on a set of reference conditions for the secondary fluid inlet temperatures: 4 °C on the source side and 35 °C on the load side for heating and 15 °C and 10 °C for cooling.

During a given time step, the model evaluates the nominal capacity at the minimum and maximum frequencies ($Q_{nom_{min,f}}$ and $Q_{nom_{max,f}}$) and the corresponding power consumption ($P_{nom_{min,f}}$ and $P_{nom_{max,f}}$):

$$Q_{nom_{max,f}} = Cap_{ratio_{max,f}} \times Q_{ref} \text{ and } Q_{nom_{min,f}} = Cap_{ratio_{min,f}} \times Q_{ref} \quad (3)$$

$$P_{nom_{max,f}} = Power_{ratio_{max,f}} \times P_{ref} \text{ and } P_{nom_{min,f}} = Power_{ratio_{min,f}} \times P_{ref} \quad (4)$$

where Q_{ref} and P_{ref} are, respectively, the reference capacity and power at the maximum frequency for the reference conditions. If required, linear interpolations are performed in the performance map tables. One advantage of using normalized performance data is that they can be used for any heat pump size by scaling the values of Q_{ref} and P_{ref} . In doing so it is assumed that the heat pumps follow the behavior of the heat pump (Figures 5 and 6) on which the performance map is based. This was verified for the present study and results (not presented here) indicate that the performance of the selected compressor could be scaled to represent the power usage and capacity of compressors of the same family.

Table 3 Performance map of the VSHP in heating mode at full load (100 Hz)

$Q_{ref}=15.33\text{ kW}; P_{ref}= 3.99\text{ kW}$ for $T_{wL_{in}} = 35^{\circ}\text{C}, T_{ws_{in}} = 4^{\circ}\text{C}, 100\text{Hz}$										
$T_{wL_{in}}\text{ [}^{\circ}\text{C]}$	30		35		40		45		50	
$T_{ws_{in}}\text{ [}^{\circ}\text{C]}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$
-2	0.874	0.893	0.870	0.968	0.865	1.049	0.859	1.137	0.852	1.233
0	0.917	0.903	0.912	0.979	0.906	1.061	0.900	1.150	0.890	1.247
2	0.960	0.912	0.956	0.990	0.949	1.073	0.941	1.163	0.931	1.260
4	1.005	0.920	1.000*	1.000*	0.993	1.085	0.984	1.176	0.972	1.273
6	1.052	0.927	1.046	1.010	1.038	1.096	1.027	1.188	1.014	1.287
8	1.099	0.934	1.093	1.018	1.083	1.107	1.071	1.200	1.055	1.299
10	1.147	0.940	1.139	1.026	1.128	1.117	1.114	1.211	1.096	1.311

Table 4 Performance map of the VSHP in cooling mode at full load (100 Hz)

$Q_{ref}=17.44$ kW; $P_{ref}= 2.398$ kW for $T_{wL_{in}} = 15^{\circ}\text{C}$, $T_{wS_{in}} = 10^{\circ}\text{C}$, 100Hz									
$T_{wL_{in}} [^{\circ}\text{C}]$	7		15		20		25		
$T_{wS_{in}} [^{\circ}\text{C}]$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$	
10	0.822	1.031	1.000*	1.000*	1.113	0.963	1.275	0.910	
15	0.811	1.159	0.992	1.146	1.100	1.116	1.262	1.076	
20	0.796	1.287	0.976	1.292	1.073	1.268	1.232	1.239	
25	0.778	1.417	0.955	1.439	1.071	1.432	1.219	1.415	
30	0.756	1.550	0.927	1.588	1.029	1.588	1.166	1.579	
35	0.730	1.688	0.892	1.739	1.014	1.758	1.140	1.759	
40	0.701	1.834	0.849	1.896	0.959	1.920	1.105	1.940	
45	0.667	1.988	0.799	2.057	0.934	2.099	1.046	2.118	

Table 5 Performance map of the VSHP in heating mode at (15 Hz)

$Q_{ref}=15.33\text{ kW}; P_{ref}= 3.99\text{ kW for }T_{wL_{in}} = 35^{\circ}\text{C}, T_{ws_{in}} = 4^{\circ}\text{C}, 100\text{Hz}$										
$T_{wL_{in}}\text{ [}^{\circ}\text{C]}$	30		35		40		45		50	
$T_{ws_{in}}\text{ [}^{\circ}\text{C]}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$
-2	0.192	0.197	0.191	0.213	0.190	0.231	0.189	0.250	0.187	0.271
0	0.202	0.199	0.201	0.215	0.199	0.234	0.198	0.253	0.196	0.274
2	0.211	0.201	0.210	0.218	0.209	0.236	0.207	0.256	0.205	0.277
4	0.221	0.202	0.220	0.220	0.218	0.239	0.216	0.259	0.214	0.280
6	0.231	0.204	0.230	0.222	0.228	0.241	0.226	0.261	0.223	0.283
8	0.242	0.206	0.240	0.224	0.238	0.243	0.236	0.264	0.232	0.286
10	0.252	0.207	0.251	0.226	0.248	0.246	0.245	0.266	0.241	0.289

Table 6 Performance map of the VSHP in cooling mode at (15 Hz)

$Q_{ref}=17.44 \text{ kW}; P_{ref}= 2.398 \text{ kW}$ for $T_{w_{Lin}} = 15^\circ\text{C}, T_{w_{Sin}} = 10^\circ\text{C}, 100\text{Hz}$								
$T_{w_{Lin}} [^\circ\text{C}]$	7		15		20		25	
$T_{w_{Sin}} [^\circ\text{C}]$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$	Cap_{ratio}	$Power_{ratio}$
10	0.181	0.227	0.220	0.220	0.245	0.212	0.280	0.200
15	0.178	0.255	0.218	0.252	0.242	0.246	0.278	0.237
20	0.175	0.283	0.215	0.284	0.236	0.279	0.271	0.273
25	0.171	0.312	0.210	0.317	0.236	0.315	0.268	0.311
30	0.166	0.341	0.204	0.349	0.226	0.349	0.256	0.347
35	0.161	0.371	0.196	0.383	0.223	0.387	0.251	0.387
40	0.154	0.403	0.187	0.417	0.211	0.422	0.243	0.427
45	0.147	0.437	0.176	0.453	0.205	0.462	0.230	0.466

* reference conditions

For example, using the data in Table 3, the nominal heating capacity and power for $T_{w_{Lin}} = 0^\circ\text{C}$ and $T_{w_{Sin}} = 30^\circ\text{C}$ at 100 Hz are 14.04 kW (0.917×15.33) and 3.60 kW (0.903×3.99), respectively.

At a given time step, the required capacity, Q_{load} , is given by:

$$Q_{load} = \dot{m}_{w_L} C_{p_{w_L}} (T_{w_{Lout,d}} - T_{w_{Lin}}) \quad (5)$$

Where $T_{w_{Lout,d}}$ is the user-specified desired outlet temperature on the load side. Then, depending on the value of Q_{load} in relation to $Q_{nom_{min,f}}$ and $Q_{nom_{max,f}}$, the model will perform the calculations shown in Table 7:

Table 7 Heat pump operational conditions

Condition	Heat Pump operation	Required Power (excluding auxiliary heat)	$T_{w_{Lout}}$
$Q_{load} < Q_{nom_{min,f}}$ $PLR < \sim 0.2$	OFF	$P = 0$	-
$Q_{nom_{min,f}} \leq Q_{load} < Q_{nom_{max,f}}$ $\sim 0.2 < PLR < 1$	ON	$P = EIRFPLR \times P_{nom_{max,f}}$	$= T_{w_{Lout,d}}$
$Q_{load} \geq Q_{nom_{max,f}}$ $PLR \geq 1$	ON + auxiliary heating	$P = P_{nom_{max,f}}$	$= \frac{Q_{nom_{max,f}}}{\dot{m}_{w_L} C_{p_{w_L}}} + T_{w_{Lin}}$
With $PLR = Q_{load}/Q_{nom_{max,f}}$			

Thus, when the required capacity is between the nominal capacities at the minimum and maximum frequencies, the required power (P) is obtained by multiplying the nominal required power at the maximum frequency by the $EIRFPLR$. If the required capacity is less than $Q_{nom_{min,f}}$ (which typically corresponds to a $PLR \sim 0.2$), then the heat pump stops. If it is greater than $Q_{nom_{max,f}}$, then the set temperature cannot be reached and $T_{w_{L_{out,d}}}$ is evaluated as shown in Table 7.

Values of the $EIRFPLR$ were evaluated from the complete model. These values are shown in Figure 7 for the cooling mode (heating results are similar). It is apparent from these results that the $EIRFPLR$ is a function of the part load ratio (PLR) and ΔT , the temperature difference between the inlet temperature of both secondary fluids ($\Delta T = T_{w_{L_{in}}} - T_{w_{S_{in}}}$). To account for this dependency, the methodology used by Knapp, Michalak, and Sing (2022) is adopted to relate the $EIRFPLR$ to the PLR and ΔT using the DOE2-2 model (Hirsch 2014):

$$EIRFPLR = a_0 + a_1 PLR + a_2 PLR^2 + a_3 \Delta T + a_4 \Delta T^2 + a_5 PLR \Delta T \quad (6)$$

The coefficients a_i in each operating mode were obtained from linear regressions to the full set of data and are presented in Tables A-1 and A-2 in the Appendix. Curves based on Equation 6 are shown in Figure 7 where it can be seen that they represent a good fit for the data. In cooling, the RMSE is 0.0173, with a maximum error of 9.8%. In heating (not shown here) the RMSE is 0.0033 and the maximum error is 5.1%. Finally, it should be noted that the ΔT can take a negative value in cooling as shown in Figure 7.

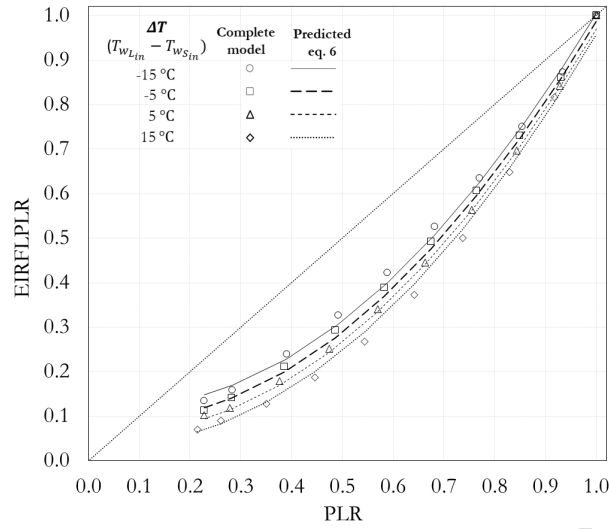


Figure 7 *EIRFPLR* as a function of the *PLR* and ΔT

Table 8 Coefficients for the *EIRFPLR* equation

Equation 6	Heating	Cooling
a_0	-0.0753787	0.076601
a_1	0.6873673	-0.098424
a_2	0.3638538	0.999283
a_3	0.0025503	-0.002881
a_4	-0.0000029	0.000012
a_5	-0.0017232	0.001280
RMSE	0.0033	0.0173
R^2 (%)	99.99	99.68
max error (%)	5.1	9.8

Results in Figure 7 indicate that *EIRFPLR* values are all below the line where *EIRFPLR* = *PLR*. This indicates that the power consumption decreases at a faster rate than the required capacity. For example, when the capacity is reduced to 20% (*PLR* = 0.2) and $\Delta T = 35$ °C, the value of the *EIRFPLR* is ~ 0.15 . Thus, this *VSHP* is proportionally more efficient at part load. This is primarily attributed to the fact that the plate heat exchangers are oversized at part load which enhances the efficiency of the heat exchange and reduces the temperature difference between the condenser and the evaporator, leading to a decrease in power consumption. Finally,

it should be noted that for a given PLR , the $EIRFPLR$ value decreases as the ΔT is reduced, owing to the improvement in cycle efficiency.

APPLICATION

One of the reasons for developing a water-to-water heat pump model is to study the impact of such devices on the required length of GHE and heat pump energy consumption. The system chosen to undertake this study is shown in Figure 8 and the main characteristics of the system are presented in Table 9. It is a residential system located in a heating-dominated climate (Montréal, Canada). In such a system, the GHE length is determined based on the heating conditions and, therefore, only heating conditions are considered here. The $VSHP$ is linked to a vertical GHE on the source side and to a fan coil on the load side. An auxiliary electric heating system (10 kW) provides backup heat. The compressor speed is regulated to achieve the desired outlet temperature, $T_{w_{Lout,d}}$, which is adjusted using a simple external air temperature (or ambient temperature, T_{amb}) reset control commonly used by manufacturers (GeoConfort 2023).

$$T_{w_{Lout,d}} = -0.556 \times T_{amb} + 33.3 \quad (\text{where } T_{amb} \text{ is in } ^\circ\text{C}) \quad (7)$$

Thus, $T_{w_{Lout,d}}$ is lowered as the ambient temperature increases and the heating load decreases. This reduces the temperature difference $T_{w_{Lin}} - T_{w_{Sin}}$ and increases the heat pump COP as was shown earlier. The system is controlled with a two-stage thermostat. First, the heat pump is activated to maintain the building set point temperature (21 °C) by providing the fan coil with a fluid at $T_{w_{Lout}}$. The building is represented using Type 88, a single-zone lumped capacitance model.

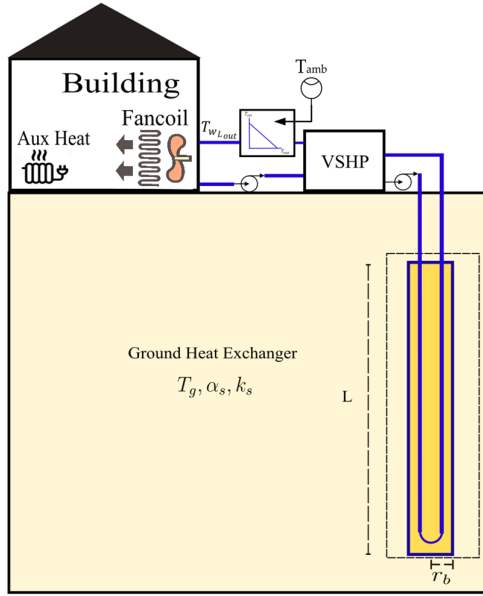


Figure 8 System under investigation

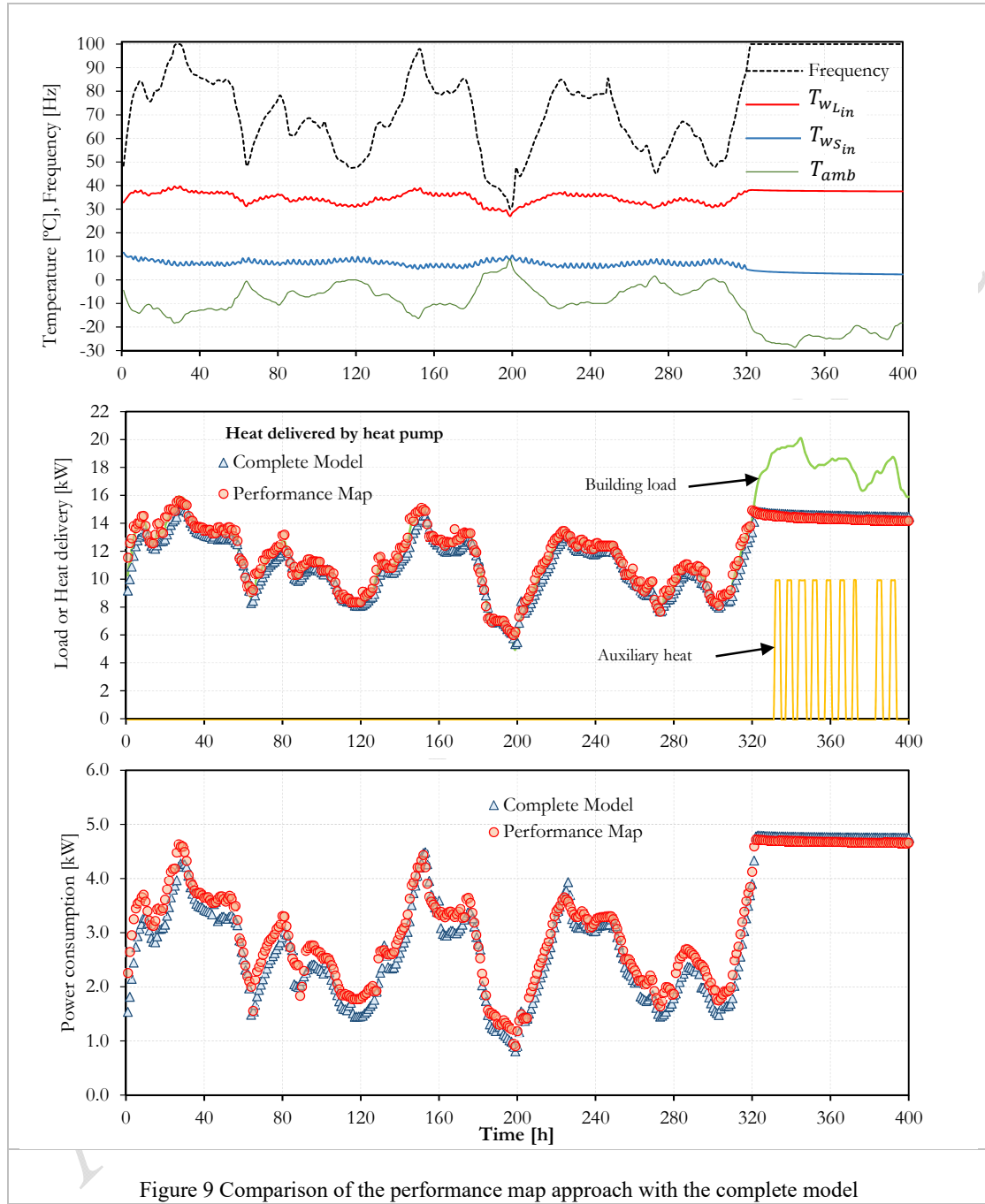
Component	TRNSYS TYPE	Principal characteristic	Value
Building	88	Overall building loss coefficient, UA [kW/K]	0.225
		Building mass, MCp [kJ/K]	20000
		Peak building load [kW]	11.1
		Internal gains [kW]	0.5
Fancoil	996	Rated Total Heating Capacity [kW]	10
		Rated Volumetric Air Flowrate [m ³ /h]	1700
		Rated Liquid Flowrate [kg/h]	2000
Auxiliary heat	930	Heating Capacity [kW]	10
VSGSHP	Type 1328	Continuous operation between minimum and maximum recommended frequencies. Four (4) performance maps data files.	
Borehole	557b-2 Vertical U-Tube Ground Heat Exchanger	Borehole radius [m]	0.075
		Borehole thermal resistance [m-K/W]	0.13
		Ground thermal conductivity [W/m-K]	1.3
		Ground thermal diffusivity α_s [m ² /s]	6.5×10^{-7}
		Undisturbed ground temperature [°C]	10

Table 9 Main characteristics of the building and GHE

The required borehole length is determined on a trial and error basis, where the length is varied until the inlet temperature to the heat pump reaches a value of 0 °C over 10 years. The system is controlled with a two-stage thermostat. First, the heat pump is activated to maintain the building set point temperature (21 °C) by providing the fan coil with a fluid at T_{wLout} . If the heat pump is unable to meet the building set point temperature, then the auxiliary heater is activated. Two constant flow circulating pumps (source and load side) are activated whenever the heat pump is operating.

Before examining the impact of the VSHP on the required length of the GHE and the heat pump energy consumption, the performance map approach (using Type 1328) is compared to the complete model for the system under investigation. The results of this comparison are presented in Figure 9 for the first 400 hours of a typical year based on simulations of the system presented in Figure 8 with a 6-minute time step. Results in Figure 9 have been integrated to be presented on an hourly basis. In this case, the VSHP is sized to cover 80% of the peak building load when

it operates at maximum speed. The top portion of the graph shows the evolution of the various temperatures and of the VSHP frequency obtained from the complete heat pump model. Over this period, the compressor is essentially always ON. It can be seen that during periods of low ambient temperatures ($t > 330$ h), the compressor operates at maximum frequency (100 Hz) and thus at maximum capacity. During that period, the return temperature from the borehole (i.e. heat pump inlet temperature) tends towards 0 °C. In contrast, when the ambient temperature is higher ($t \approx 200$ h) the heat pump reaches a frequency of 30 Hz.



As can be seen in the middle graph, the heat delivered predicted by both approaches is very similar except near peaks and valleys and also when the heat pump operates at full capacity ($t > 330$ h). Over the period $0 < t < 400$ h, the RMSE is 0.35 kW with a maximum error of 2.5%. At $t \sim 330$ h, the *VSHP* capacity is less than the building load, and auxiliary heating is activated

periodically to meet the set point temperature. The performance map approach and the complete model predict essentially the same auxiliary heating values and both curves are superimposed in Figure 9. The bottom graph in Figure 9 shows the heat pump power predicted by both approaches. Here again, the agreement between the two approaches is good with an RMSE of 0.24 kW and a maximum error of 5.1%.

The relationship between the size of the *VSHP* and the *GHE* length and the heat pump energy consumption will now be examined. A full heating season, which extends from October 1st and May 1st (5088 hours) in Montréal, was simulated with a 6-minute time step using the most recent Typical Meteorological Year for Montréal for a building with a peak building load of 11.1 kW (Table 9). Results are also compared with simulations where the *VSHP* (Figure 9) is replaced by a fixed-speed compressor heat pump (*FSHP*) equipped with the same type of compressor but running at one speed corresponding to a frequency of 60 Hz.

First, the heating capacities of the *VSHP* and *FSHP* are plotted as a function of ambient temperature (Figure 10). The heating capacity of a typical air-source heat pump (*ASHP*) is also shown for comparison. The rated capacity (Q_{ref}) is 8.88 kW at 100 Hz for the *VSHP* and 60 Hz for the *FSHP*. The corresponding capacity of the *VSHP* operating at 60 Hz at the same rated condition is 5.62 kW, a reduction of 36%. The curve for the *FSHP* shows a slow decline of the heating capacity with a decrease in the ambient temperature. This is because as the building load increases, more heat is extracted from the ground and the return fluid temperature (not shown in Figure 10) from the borehole ($T_{ws_{in}}$) decreases (from ~ 10 down to ~ 0 °C). In turn, this decreases the heating capacity from ~ 9 to ~ 8 kW when T_{amb} decreases from ~ 10 °C down to ~ -30 °C. Below the balance point (i.e. at $T_{amb} \sim -16$ °C), the capacity of the *FSHP* is insufficient, and auxiliary heat is required. The behavior of the *VSHP* is different in the region

where $-16\text{ }^{\circ}\text{C} < T_{amb} < 10\text{ }^{\circ}\text{C}$ where, as expected, the heating capacity follows the building load curve. The balance point where auxiliary heating is required corresponds also to $T_{amb} \sim -16\text{ }^{\circ}\text{C}$, which is to be expected as the *VSHP* and *FSHP* are both sized to cover 80% of the peak building load. For *ASHP*, the capacity is directly dependent on the ambient temperature and the slope of the heating capacity is steeper with a warmer balance point temperature. Thus, *FSHP* and *VSHP* require less auxiliary heat than a typical *ASHP*.

One way to quantify the amount of auxiliary heat required is to determine the so-called “energy coverage” defined as:

$$\text{Energy Coverage (\%)} = \frac{\text{Total heat supplied by the heat pump (excluding auxiliary heat)}}{\text{Total heat required by building over the heating season}} \quad (8)$$

The energy coverage will depend on the size of the heat pump in comparison to the peak building load. This is defined as the “effect coverage”:

$$\text{Effect Coverage} = \frac{\text{Rated heating capacity of the heat pump (kW)}}{\text{Peak building load (kW)}} \quad (9)$$

For example, in Figure 10, the effect coverage of the *FSHP* and *VSHP* is 80% (8.88/11.1).

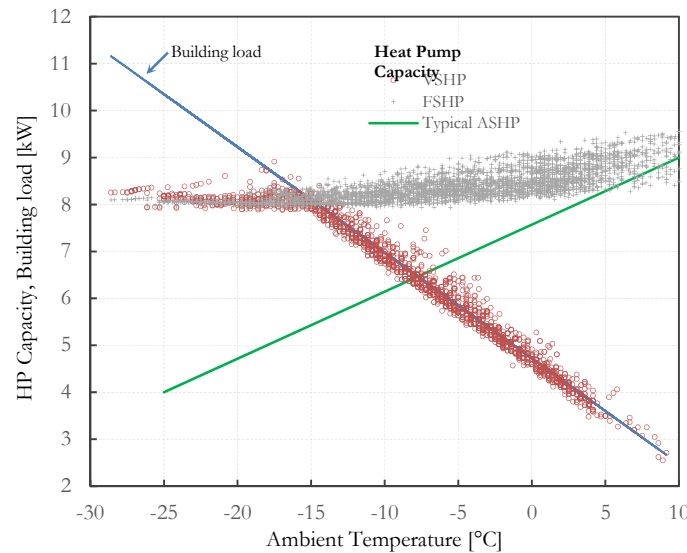


Figure 10 Variation of heating capacity as a function of ambient temperature for three types of heat pump

Figure 11 shows a plot of the energy coverage as a function of the effect coverage. Three curves are shown: the curve reported by Rosén et al. (2001) for a typical Swedish home equipped with a *FSHP* and curves obtained for the *FSHP* and *VSHP* described in this study and simulated using the performance map approach. The effect coverage is varied by keeping the building characteristics constant (Table 9) and varying Q_{ref} and P_{ref} for the heat pumps. The *GHE* length is different in each case and is determined, as indicated earlier, based on a minimum return borehole temperature of 0 °C.

The curves in Figure 11 indicate that the energy coverage increases at a faster rate than the effect coverage up to an effect coverage of 60%. At that point, the energy coverage is ~ 93%. Thus, *VSHP* designed to meet 60% of the peak building demand will provide 93% of the heat required while the remaining 7% will be supplied by auxiliary heat. Beyond an effect coverage of 60%, there is little gain in energy coverage.

The curves for the *FSHP* and *VSHP* are almost identical. Thus, there appears to be no significant difference between the energy coverage curves for the *FSHP* and *VSHP*. The curve presented by Rosén et al. (2001) is slightly different than the one in the present study. Rosén et al. (2001) do not give details related to his curve. Thus, it is difficult to identify the sources of the discrepancies. It might be due to different heat pump performance, climate, or ground properties.

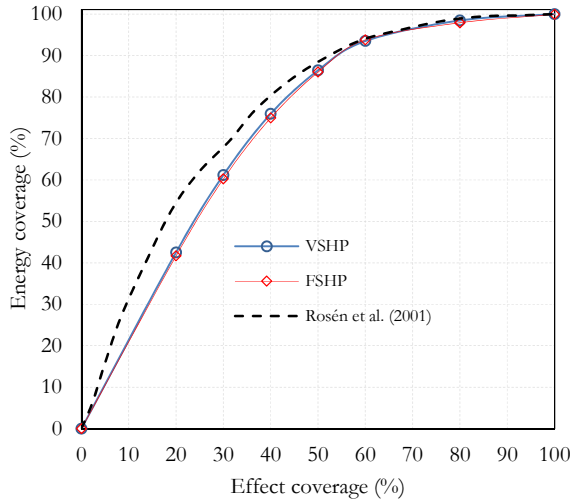


Figure 11 Energy coverage as a function of the effect coverage

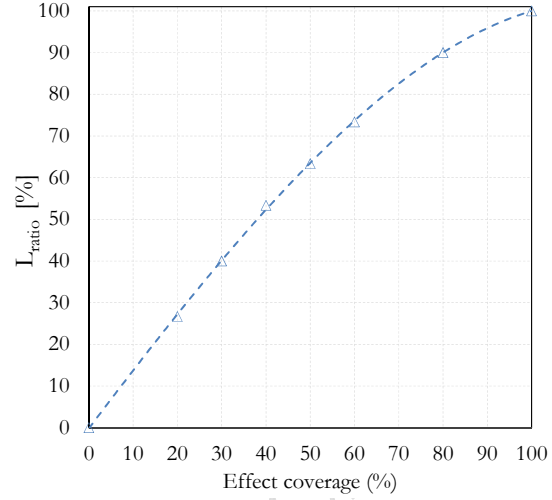


Figure 12 Normalised *GHE* length as a function of the effect coverage.

The impact of effect coverage on the *GHE* length is shown in Figure 12. The length has been normalized with the length at a 100% effect coverage, which is 215 m in the present case. The L_{ratio} is proportional to the effect coverage up to an effect coverage of 60%. At that point, the normalized length is equal to 75%. After 60%, the L_{ratio} increases at a slower rate. The results presented in Figures 11 and 12 should be used with caution as the curves were obtained with a particular heat pump and specific ground properties. More work is needed to see how these curves vary for other conditions.

CONCLUSION

A complete water-to-water variable speed heat pump (*VSHP*) model is developed in this study. The physics-based approach incorporates individual models for the plate heat exchangers (evaporator and condenser), the expansion valve, and the variable speed compressor. The model requires five inputs: The inlet temperature and mass flow rate of both secondary fluids as well as compressor speed. The outputs are the capacity and power consumption. Using this model,

it is shown (Figure 5) that the COP curves exhibit a maximum value that is not necessarily coincident with full capacity (at maximum speed). For example, for a temperature difference of 20 °C between the source and load temperatures, the COP is maximum at 40 Hz and it decreases when the frequency is either increased or decreased. It is also shown that the capacity varies linearly with frequency. The implementation of the complete model in a new TRNSYS type was not possible for long simulations due to convergence problems. Indeed, the convergence of the solution to the set of non-linear equations is highly dependent on the choice of guess values which becomes problematic for rapidly changing conditions in actual system simulation.

To solve this problem, a performance map approach, based on the complete model, is also developed based on an existing TRNSYS Type (Type1323_v2a). Two important modifications are made to this Type. First, a minimum speed of operation is implemented to reflect the fact that variable-speed compressors stop below a certain speed. Secondly, a more accurate representation of power consumption at part load is implemented. In a 400-hour simulation, the performance map approach produced good results when compared to the complete model, with maximum errors of 2.5% for the heating capacity and 5.1% for power.

The performance map-based model was used in an annual simulation of a *VSHP* operating with a ground heat exchanger for a residential building in a heating-dominated climate. Results indicate that the energy coverage (i.e. percentage of annual heat supplied by the heat pump) increases at a faster rate than the effect coverage (i.e. percentage of peak building heat supplied by the heat pump) for values of the effect coverage below 60%. The energy coverage is ~ 93% for an effect coverage of 60%. The non-dimensional *GHE* length is proportional to the effect coverage up to an effect coverage of 60%. At that point, the normalized length is equal to 75%. After 60%, the non-dimensional *GHE* length increases at a slower rate.

The energy-coverage versus effect-coverage curve was formulated following a set of particular parameters. Further investigations should be made for various heat pumps, ground thermal properties, and climates.

NOMENCLATURE

$k_s =$	Ground thermal conductivity (W/m-K)
$\dot{m}_{corrected} =$	Corrected refrigerant mass flow rate (kg/h)
$\dot{m}_{rated} =$	Refrigerant mass flow rate at rated superheat (kW)
$\dot{m}_{wL} =$	Secondary fluid flow rate on the load side (kg/h)
$\dot{m}_{wS} =$	Secondary fluid flow rate on the source side (kg/h)
$P_{nom_{min,f}} =$	Nominal power at the minimum frequency (kW)
$P_{nom_{max,f}} =$	Nominal power at the maximum frequency (kW)
$P_{ref} =$	Rated power at reference conditions (kW)
$Q_{ref} =$	Rated capacity at reference conditions (kW)
$Q_{load} =$	Required capacity (kW)
$Q_{nom_{min,f}} =$	Nominal capacity at the minimum frequencies (kW)
$Q_{nom_{max,f}} =$	Nominal capacity at the minimum frequencies (kW)
$Q_L =$	Heat extracted from the low-temperature source (kW)
$Q_H =$	Heat rejected to the high-temperature source (kW)
$r_b =$	Borehole radius (m)
$T_{amb} =$	Ambient temperature (°C)
$T_D =$	Discharge dew point temperature (°C)
$T_g =$	Undisturbed ground temperature (°C)
$T_s =$	Suction dew point temperature (°C)
$T_{set} =$	Temperature load set point (°C)
$T_{sL_{in}} =$	Inlet source temperature to the heat pump (°C)
$T_{sL_{out}} =$	Outlet source temperature from the heat pump (°C)
$T_{wL_{in}} =$	Inlet load temperature to the heat pump (°C)

$T_{w_{Lout}}$	=	Outlet load temperature from the heat pump (°C)
$T_{w_{Lout,d}}$	=	User-specified desired outlet temperature on the load side (°C)

Greek symbols

α_s	=	Ground thermal diffusivity (m ² /s)
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Abbreviations

$EIRFPLR$	=	Energy Input Ratio as a function of Part Load Ratio
$FSHP$	=	Fixed speed heat pump
P	=	Power (kW)
PHX	=	Plate heat exchanger
PLR	=	Part Load Ratio
$VSHP$	=	Variable speed heat pump
W	=	Electrical power supplied by the compressor (kW)

Appendix

Table A-1 Simulation results for the water-to-water variable speed heat pump model in heating mode at part load and full capacity

$T_{wL_{in}}$ [°C]	$T_{wS_{in}}$ [°C]	Frequency [Hz]	Capacity [kW]	Power [kW]	PLR	EIRFPLR	$T_{wL_{in}}$ [°C]	$T_{wS_{in}}$ [°C]	Frequency [Hz]	Capacity [kW]	Power [kW]	PLR	EIRFPLR
30	-2	15	2.91	0.55	0.217	0.154 ¹	35	-2	15	2.84	0.61	0.213	0.159
30	-2	20	3.54	0.68	0.264	0.191	35	-2	20	3.47	0.76	0.260	0.197
30	-2	30	4.78	0.96	0.357	0.270	35	-2	30	4.71	1.07	0.353	0.276
30	-2	40	6.02	1.27	0.449	0.355	35	-2	40	5.94	1.40	0.445	0.362
30	-2	50	7.24	1.59	0.540	0.447	35	-2	50	7.17	1.75	0.537	0.453
30	-2	60	8.47	1.94	0.632	0.545	35	-2	60	8.4	2.13	0.630	0.551
30	-2	70	9.7	2.32	0.724	0.650	35	-2	70	9.63	2.53	0.722	0.655
30	-2	80	10.93	2.71	0.816	0.760	35	-2	80	10.86	2.95	0.814	0.764
30	-2	90	12.16	3.13	0.907	0.877	35	-2	90	12.1	3.40	0.907	0.879
30	-2	100	13.4	3.57	1.000	1.000	35	-2	100	13.34	3.86	1.000	1.000
30	4	15	3.26	0.53	0.212	0.143	35	4	15	3.2	0.60	0.209	0.150
30	4	20	4.01	0.67	0.260	0.181	35	4	20	3.94	0.75	0.257	0.188
30	4	30	5.48	0.96	0.356	0.262	35	4	30	5.41	1.08	0.353	0.270
30	4	40	6.93	1.28	0.450	0.348	35	4	40	6.85	1.42	0.447	0.357
30	4	50	8.36	1.62	0.543	0.441	35	4	50	8.28	1.79	0.540	0.450
30	4	60	9.78	1.98	0.635	0.540	35	4	60	9.7	2.19	0.633	0.548
30	4	70	11.19	2.37	0.726	0.646	35	4	70	11.11	2.61	0.725	0.653
30	4	80	12.6	2.78	0.818	0.758	35	4	80	12.52	3.04	0.817	0.763
30	4	90	14.01	3.21	0.909	0.875	35	4	90	13.93	3.51	0.909	0.879
30	4	100	15.41	3.67	1.000	1.000	35	4	100	15.33	3.99	1.000	1.000
30	10	15	3.64	0.50	0.207	0.132	35	10	15	3.58	0.57	0.205	0.139
30	10	20	4.52	0.64	0.257	0.170	35	10	20	4.46	0.73	0.255	0.178
30	10	30	6.24	0.94	0.355	0.251	35	10	30	6.18	1.07	0.354	0.261
30	10	40	7.93	1.27	0.451	0.339	35	10	40	7.86	1.43	0.450	0.350
30	10	50	9.59	1.63	0.546	0.433	35	10	50	9.51	1.82	0.545	0.444
30	10	60	11.23	2.01	0.639	0.535	35	10	60	11.14	2.23	0.638	0.544
30	10	70	12.84	2.41	0.730	0.641	35	10	70	12.76	2.66	0.731	0.649
30	10	80	14.44	2.83	0.821	0.754	35	10	80	14.35	3.12	0.822	0.760
30	10	90	16.02	3.28	0.911	0.874	35	10	90	15.94	3.60	0.913	0.878
30	10	100	17.58	3.75	1.000	1.000	35	10	100	17.46	4.10	1.000	1.000
40	-2	15	2.75	0.69	0.207	0.164	45	-2	15	2.64	0.77	0.200	0.169
40	-2	20	3.37	0.85	0.254	0.202	45	-2	20	3.26	0.94	0.248	0.207
40	-2	30	4.62	1.18	0.348	0.282	45	-2	30	4.51	1.31	0.342	0.288
40	-2	40	5.86	1.54	0.442	0.368	45	-2	40	5.76	1.70	0.437	0.374
40	-2	50	7.08	1.92	0.534	0.459	45	-2	50	6.99	2.12	0.531	0.466
40	-2	60	8.32	2.33	0.627	0.557	45	-2	60	8.23	2.56	0.625	0.564
40	-2	70	9.55	2.76	0.720	0.660	45	-2	70	9.46	3.03	0.718	0.667
40	-2	80	10.78	3.22	0.813	0.768	45	-2	80	10.7	3.51	0.812	0.773
40	-2	90	12.02	3.69	0.906	0.881	45	-2	90	11.93	4.02	0.906	0.885
40	-2	100	13.26	4.19	1.000	1.000	45	-2	100	13.17	4.54	1.000	1.000
40	4	15	3.11	0.68	0.204	0.156	45	4	15	3.01	0.76	0.200	0.163
40	4	20	3.86	0.84	0.254	0.195	45	4	20	3.74	0.95	0.248	0.201
40	4	30	5.31	1.20	0.349	0.276	45	4	30	5.2	1.33	0.345	0.283
40	4	40	6.75	1.57	0.443	0.363	45	4	40	6.63	1.74	0.440	0.370
40	4	50	8.17	1.97	0.537	0.456	45	4	50	8.05	2.17	0.534	0.463
40	4	60	9.59	2.40	0.630	0.553	45	4	60	9.46	2.63	0.627	0.560
40	4	70	11	2.85	0.723	0.657	45	4	70	10.87	3.11	0.721	0.663
40	4	80	12.41	3.32	0.815	0.766	45	4	80	12.28	3.62	0.814	0.770
40	4	90	13.82	3.81	0.908	0.880	45	4	90	13.68	4.14	0.907	0.883
40	4	100	15.22	4.33	1.000	1.000	45	4	100	15.08	4.69	1.000	1.000
40	10	15	3.5	0.65	0.202	0.146	45	10	15	3.4	0.74	0.199	0.154
40	10	20	4.37	0.83	0.253	0.186	45	10	20	4.264	0.94	0.250	0.193
40	10	30	6.08	1.20	0.352	0.269	45	10	30	5.954	1.34	0.349	0.278
40	10	40	7.74	1.60	0.448	0.358	45	10	40	7.609	1.77	0.446	0.367
40	10	50	9.39	2.01	0.543	0.452	45	10	50	9.241	2.23	0.541	0.460
40	10	60	11.01	2.46	0.637	0.551	45	10	60	10.86	2.70	0.636	0.559
40	10	70	12.62	2.92	0.730	0.655	45	10	70	12.45	3.20	0.729	0.662
40	10	80	14.2	3.41	0.821	0.765	45	10	80	14.03	3.72	0.822	0.770
40	10	90	15.76	3.92	0.912	0.880	45	10	90	15.59	4.27	0.913	0.883
40	10	100	17.29	4.46	1.000	1.000	45	10	100	17.07	4.83	1.000	1.000

¹ To calculate the EIRFPLR at part load, the power required at that load (0.55 kW at 15Hz) is divided by the nominal power at full capacity (3.57 kW at 100Hz) leading to EIRFPLR = 0.154

Table A-2
Simulation results for the water-to-water variable speed heat pump model in cooling mode at part load
and full capacity

$T_{w_{lin}}$ [°C]	$T_{w_{sin}}$ [°C]	Frequency [Hz]	Capacity [kW]	Power [kW]	PLR	EIRFPLR	$T_{w_{lin}}$ [°C]	$T_{w_{sin}}$ [°C]	Frequency [Hz]	Capacity [kW]	Power [kW]	PLR	EIRFPLR
7	10	15	3.39	0.30	0.237	0.122 ²	15	10	15	3.98	0.27	0.228	0.112
7	10	20	4.14	0.36	0.289	0.148	15	10	20	4.85	0.30	0.278	0.126
7	10	30	5.57	0.52	0.388	0.211	15	10	30	6.57	0.43	0.377	0.179
7	10	40	6.95	0.72	0.485	0.290	15	10	40	8.27	0.60	0.474	0.251
7	10	50	8.30	0.94	0.579	0.380	15	10	50	9.94	0.82	0.570	0.340
7	10	60	9.60	1.19	0.670	0.483	15	10	60	11.57	1.07	0.663	0.445
7	10	70	10.87	1.47	0.759	0.596	15	10	70	13.16	1.35	0.755	0.564
7	10	80	12.11	1.78	0.845	0.720	15	10	80	14.71	1.67	0.843	0.696
7	10	90	13.30	2.11	0.928	0.854	15	10	90	16.20	2.02	0.929	0.842
7	10	100	14.33	2.47	1.000	1.000	15	10	100	17.44	2.40	1.000	1.000
7	20	15	3.27	0.42	0.235	0.137	15	20	15	3.88	0.35	0.228	0.113
7	20	20	4.01	0.50	0.289	0.161	15	20	20	4.80	0.44	0.282	0.142
7	20	30	5.43	0.73	0.391	0.236	15	20	30	6.58	0.65	0.386	0.211
7	20	40	6.79	0.99	0.489	0.321	15	20	40	8.28	0.91	0.486	0.294
7	20	50	8.09	1.28	0.583	0.414	15	20	50	9.92	1.20	0.582	0.388
7	20	60	9.35	1.59	0.673	0.515	15	20	60	11.50	1.53	0.675	0.493
7	20	70	10.57	1.93	0.761	0.625	15	20	70	13.01	1.88	0.764	0.607
7	20	80	11.76	2.29	0.847	0.742	15	20	80	14.47	2.26	0.850	0.730
7	20	90	12.90	2.68	0.929	0.867	15	20	90	15.86	2.67	0.931	0.860
7	20	100	13.89	3.09	1.000	1.000	15	20	100	17.03	3.10	1.000	1.000
7	30	15	3.06	0.59	0.232	0.158	15	30	15	3.68	0.47	0.228	0.123
7	30	20	3.77	0.76	0.286	0.204	15	30	20	4.58	0.61	0.284	0.159
7	30	30	5.12	0.95	0.388	0.256	15	30	30	6.31	0.91	0.390	0.239
7	30	40	6.41	1.28	0.487	0.343	15	30	40	7.95	1.25	0.492	0.327
7	30	50	7.66	1.62	0.581	0.437	15	30	50	9.52	1.61	0.589	0.423
7	30	60	8.86	1.99	0.672	0.536	15	30	60	11.02	2.00	0.682	0.525
7	30	70	10.02	2.39	0.760	0.642	15	30	70	12.45	2.42	0.770	0.635
7	30	80	11.14	2.81	0.845	0.755	15	30	80	13.81	2.86	0.855	0.750
7	30	90	12.22	3.25	0.927	0.874	15	30	90	15.09	3.32	0.934	0.872
7	30	100	13.18	3.72	1.000	1.000	15	30	100	16.16	3.81	1.000	1.000
20	10	15	4.39	0.25	0.226	0.106	25	10	15	4.77	0.17	0.215	0.080
20	10	20	5.30	0.26	0.273	0.111	25	10	20	5.79	0.20	0.260	0.094
20	10	30	7.15	0.36	0.369	0.155	25	10	30	7.78	0.28	0.350	0.127
20	10	40	9.05	0.51	0.466	0.222	25	10	40	9.91	0.41	0.446	0.187
20	10	50	10.94	0.71	0.564	0.309	25	10	50	12.09	0.59	0.544	0.268
20	10	60	12.82	0.96	0.660	0.415	25	10	60	14.27	0.81	0.642	0.373
20	10	70	14.64	1.24	0.754	0.538	25	10	70	16.40	1.09	0.738	0.500
20	10	80	16.40	1.56	0.845	0.677	25	10	80	18.45	1.42	0.830	0.649
20	10	90	18.09	1.92	0.932	0.832	25	10	90	20.40	1.78	0.918	0.816
20	10	100	19.41	2.31	1.000	1.000	25	10	100	22.23	2.18	1.000	1.000
20	20	15	4.30	0.32	0.230	0.106	25	20	15	4.83	0.29	0.225	0.098
20	20	20	5.33	0.39	0.285	0.129	25	20	20	5.98	0.33	0.278	0.111
20	20	30	7.33	0.59	0.392	0.193	25	20	30	8.25	0.49	0.384	0.165
20	20	40	9.27	0.83	0.495	0.273	25	20	40	10.49	0.72	0.488	0.241
20	20	50	11.14	1.12	0.595	0.369	25	20	50	12.66	1.00	0.589	0.335
20	20	60	12.93	1.45	0.691	0.476	25	20	60	14.73	1.32	0.686	0.445
20	20	70	14.62	1.81	0.781	0.595	25	20	70	16.69	1.69	0.777	0.569
20	20	80	16.21	2.20	0.866	0.723	25	20	80	18.49	2.09	0.861	0.704
20	20	90	17.66	2.61	0.944	0.858	25	20	90	20.10	2.52	0.936	0.847
20	20	100	18.71	3.04	1.000	1.000	25	20	100	21.48	2.97	1.000	1.000
20	30	15	4.15	0.43	0.231	0.113	25	30	15	4.66	0.39	0.229	0.104
20	30	20	5.20	0.56	0.290	0.147	25	30	20	5.87	0.50	0.289	0.132
20	30	30	7.21	0.85	0.402	0.224	25	30	30	8.18	0.77	0.402	0.204
20	30	40	9.12	1.19	0.508	0.312	25	30	40	10.38	1.10	0.511	0.291
20	30	50	10.93	1.56	0.609	0.410	25	30	50	12.47	1.48	0.613	0.391
20	30	60	12.66	1.96	0.706	0.516	25	30	60	14.43	1.89	0.710	0.500
20	30	70	14.29	2.40	0.797	0.629	25	30	70	16.25	2.34	0.799	0.617
20	30	80	15.82	2.85	0.882	0.749	25	30	80	17.89	2.80	0.880	0.741
20	30	90	17.23	3.33	0.960	0.874	25	30	90	19.31	3.29	0.950	0.869
20	30	100	17.94	3.81	1.000	1.000	25	30	100	20.33	3.79	1.000	1.000

² To calculate the EIRFPLR values at part load, the power required at that load (0.30 kW at 15Hz) is divided by the nominal power at full capacity (2.5 kW at 100Hz) leading to EIRFPLR = 0.122

REFERENCES

- AHRI. 2020 National standard CAN / ANSI / AHRI 540-2020 Performance rating of positive displacement refrigerant compressors.
- Aute, V., C. Martin, and R. Radermacher. 2015. AHRI project 8013: A study of methods to represent compressor performance data over an operating envelope based on a finite set of test data. Air-Conditioning, Heating, and Refrigeration Institute (AHRI)
- Bell, H., J. Wronski, S. Quoilin, and V. Lemort. 2014. Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp. *Industrial & Engineering Chemistry Research* 53 (6):2498-508. doi: 10.1021/ie4033999.
- Bouheret, S., and M. Bernier. 2018. Modelling of a water-to-air variable capacity ground-source heat pump. *Journal of Building Performance Simulation* 11 (3):283-93. doi: 10.1080/19401493.2017.1332686.
- Chua, K. J., S. K. Chou, and W. M. Yang. 2010. Advances in heat pump systems: A review. *Applied Energy* 87 (12):3611-24. doi: <https://doi.org/10.1016/j.apenergy.2010.06.014>.
- Climatemaster. 2022, Trilogy-45. Accessed May 19, 2023.
<https://www.climatemaster.com/geothermal-dealer/products/trilogy-45--qe-and-ve-series>
- Danfoss. 2022. Scroll compressor, VZH028CJ. Accessed May 19, 2023.
<https://store.danfoss.com/en/Climate-Solutions-for-cooling/Compressors/Compressors-for-air-conditioning/Inverter-compressors/Scroll-compressor%2C-VZH028CJANB/p/120G0186>.
- Gabel, K. S., and C. R. Bradshaw. 2022. Evaluation and quantification of compressor model predictive capabilities under modulation and extrapolation scenarios. *International Journal of Refrigeration*. 149:1-10, doi: <https://doi.org/10.1016/j.ijrefrig.2022.11.032>.
- GeoConfort. 2023. Guide WV: Variable-Speed Water-to-Water Installation & operations manual. Accessed May 19, 2023.
<https://geocomfort.com/residential-products/item/guide-wv>
- Hackel, S.P. 2008. "Development of Design Guidelines for Hybrid Ground-Coupled Heat Pump Systems." ASHRAE TRP-1384.
- Hirsch, J. J. 2014. DOE-2.2 Building Energy Use and Cost Analysis Program. Volume 2: Dictionary. James J. Hirsch & Associates. Camarillio (US).
- Jin, H., and J. Spitler. 2002. A Parameter Estimation Based Model of Water-to-Water Heat Pumps for Use in Energy Calculation Programs. *ASHRAE Transactions* vol. 108(1).

- Knapp, D., M. Michalak, and M. Sing. 2022. Modelling waste heat recovery for renewable thermal energy. Paper presented at the 12th eSim Building Simulation Conference of IBPSA-Canada, Ottawa, June 21-24.
- Ndiaye, D. 2007. Étude numérique et expérimentale de la performance en régime transitoire de pompes à chaleur eau-air en cyclage. Ph.D. thesis, École Polytechnique de Montréal.
- Okawa, Y. 2020. A manual for the Modified Powell's Hybrid Method. <https://github.com/yoki/Optimization>.
- Ossorio, R. and E. Navarro-Peris. 2021. Study of oil circulation rate in variable speed scroll compressor working with propane. *International Journal of Refrigeration* 123:63-71. doi: <https://doi.org/10.1016/j.ijrefrig.2020.12.002>.
- Popovic, P., and H N. Shapiro. 1995. A semi-empirical method for modeling a reciprocating compressor in refrigeration systems. United States.
- Rice, C K. 2001. The ORNL Modulating Heat Pump Design Tool User's Guide, United States. doi: <https://doi.org/10.2172/788665>.
- Rice, K., V. Baxter, S. Hern, T. McDonwell, J. Munk, and B. Shen. 2013. Development of a residential ground-Source Integrated heat pump. Paper presented at 11th IEA Heat Pump Conference 2014, May 12-16, Montréal (Québec) Canada.
- Rosén, B., A. Gabrielsson, J. Fallsvik, G. Hellström, and G. Nilson. 2001. System för värme och kyla ur mark: en nulägesbeskrivning [Systems for Ground Source Heating and Cooling- A Status Report - in Swedish]. *Statens geotekniska institut (SGI), Linköping, Sweden*.
- Shao, S., W. Shi, X. Li, and H. Chen. 2004. Performance representation of variable-speed compressor for inverter air conditioners based on experimental data. *International Journal of Refrigeration* 27 (8):805-15. doi: <https://doi.org/10.1016/j.ijrefrig.2004.02.008>.
- Stefanuk, N., J. Aplevich, and M. Renksizbulut. 1992. Modeling and Simulation of a Superheat Controlled Water-to-Water Heat Pump. *ASHRAE Transactions* 98:172-84.
- Underwood, C. P. 2016. Chapter 14 - Heat pump modelling. In *Advances in Ground-Source Heat Pump Systems*, edited by Simon J. Rees, 387-421. Woodhead Publishing.
- Viviescas, G., S. Houaida, M. Bernier, and M. Kummert. 2021. Energy performance advantages of using multiple compressors in a heat pump operating in heating mode. Paper presented at the 11th eSim Building Simulation Conference of IBPSA-Canada, online, June 14-16. Available at: http://www.ibpsa.org/proceedings/eSimPapers/2021/Contribution_1211_final_a.pdf

Viviescas, G., and M. Bernier. 2022. Ground heat exchanger performance with variable speed ground-source heat pumps. Paper presented at IGSHPA (2022), International Ground Source Heat Pump Association.

doi: <http://dx.doi.org/10.22488/okstate.22.000029>.

Winandy, E., C. Saavedra, and J. Lebrun. 2002. Experimental analysis and simplified modelling of a hermetic scroll refrigeration compressor. *Applied Thermal Engineering* 22 (2):107-20. doi: [https://doi.org/10.1016/S1359-4311\(01\)00083-7](https://doi.org/10.1016/S1359-4311(01)00083-7).

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