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A Parameter Estimation Based Model of Water-to-Water Heat Pumps for Use in Energy Calculation Programs

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ABSTRACT

A steady-state simulation model for a water-to-water reciprocating vapor compression heat pump is presented. The model is intended for use in energy calculation and/or building simulation programs. The model was developed from basic thermodynamic principles and heat transfer relations. The model includes several unspecified parameters that are estimated from catalog data using a multi-variable optimization procedure. It was developed with the objective of only requiring input data that are readily available from manufacturers' catalogs. Compared to equation-fit models, by retaining the physically based representation of the heat pump, a better match to the catalog data is obtained, and some extrapolation is feasible. Compared to a detailed deterministic model recently published, approximately the same RMS error is obtained without requiring additional experimental data or data for internal components.

INTRODUCTION

Reciprocating vapor compression heat pumps and chillers have been the target of a number of simulation models. Hamilton and Miller (1990) presented a classification scheme for air-conditioning equipment with two extremes. At one end of the spectrum are equation-fit models—called "functional fit" models by Hamilton and Miller—which treat the system as a black box and fit the system performance to one or a few large equations. At the other end are deterministic models—called "first principle" models by Hamilton and Miller—which are detailed models based on applying thermodynamic laws and fundamental heat and mass transfer relations to individual components.

Many of the models found in the literature might actually fall between the two extremes, although the detailed deterministic models often apply equation-fitting for some of the components. For example, in the reciprocating chiller model proposed by Bourdouxhe et al. (1994), the chiller was modeled as an assembly of several simplified components. Each component (e.g., compressor, evaporator, condenser, expansion device) is modeled with a detailed deterministic approach. The parameters describing the detailed physical geometry and operation of each component are then adjusted (i.e., in an equation-fitting procedure) to reproduce the behavior of the actual unit as accurately as possible. The model of Bourdouxhe et al. requires more details for each component than are usually available from manufacturers' catalogs. This type of model is most suitable for users that have access to internally measured data (e.g., in Bourdouhxe's model, condensing and evaporating temperatures and subcooling and superheating temperature differences) from the chiller or heat pump.

The alternative approach, equation-fitting, alleviates the need for internally measured data and usually maintains better fidelity to the catalog data. It also usually requires less computational time. These models are most suitable for users that only have access to catalog data. These models would not be useful for someone attempting to design a heat pump or chiller by modifying or replacing internal components. Especially troublesome for some applications, extrapolation of the model may lead to unrealistic results.

For use in energy calculation and/or building simulation programs, it is desirable to have a model that not only requires catalog data but allows extrapolation beyond the catalog data. In the authors' experience, this model has also been useful for

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modeling of ground-source heat pumps in novel applications where the fluid temperatures occasionally go beyond the catalog data. It has also proved useful in simulations that are part of a ground-loop sizing procedure. In this application, it often happens that the temperatures are well beyond the catalog data. Even though the ultimate outcome is that the ground-loop heat exchanger size will be adjusted to bring the temperatures within reasonable limits, it is helpful to have a model that does not catastrophically fail when the temperatures are too high or too low.

The model presented in this paper uses deterministic models of each heat pump component. Each of the fundamental equations describing the system components may have one or more parameters, which are estimated simultaneously using catalog data only; no other experimental data are required. The parameter estimation is done with a multi-variable optimization method. Once the parameters have been estimated, the heat pump model may be used as part of a multi-component system simulation.

This modeling approach has the advantage of not requiring experimental data beyond what is published in the manufacturer's catalog. Yet, its predictions are of similar or better accuracy than previously published deterministic models that required additional experimental data. Unlike the equation-fit models, the model domain may be extended beyond the catalog data without catastrophic failure in the prediction.

LITERATURE REVIEW

Simulation models of vapor-compression refrigeration and air-conditioning systems, such as heat pumps and chillers, have been the topic of numerous papers. The models can generally be classified in terms of the degree of complexity and empiricism. A review of the literature reveals a few limitations on existing models. For the more deterministic models, there is a gap between what data are provided by manufacturers' catalogs and what data the simulation models require. For equation-fit models, the valid application is limited to the manufacturer-supplied data range and conditions.

Stoecker and Jones (1982), Allen and Hamilton (1983), and Hamilton and Miller (1990) have presented steady-state equation-fit models of vapor compression refrigeration systems with reciprocating compressors. The Allen and Hamilton (1983) model utilizes overall system data, e.g., entering and leaving water temperatures and flow rates. The models of Stoecker and Jones (1982) and Hamilton and Miller (1990) require more detailed data, such as internal refrigerant pressures and temperatures. Consequently, the latter two models will be difficult to use for engineers who only have access to catalog data.

Gordon and Ng (1994) proposed a simple thermodynamic model for reciprocating chillers that they suggest might be valuable for diagnostic purposes. The model predicts the COP over a wide range of operation conditions from the inlet fluid temperatures and the cooling capacity, using three fitted parameters. The prediction of COP is remarkably good for a range of different chillers. However, the model does not predict the cooling capacity; it is required as an input. A chiller model presented by Shelton and Weber (1991), is similar in approach and also has the same limitation of requiring the cooling capacity as an input.

The quasi-static reciprocating chiller model developed by Bourdouxhe et al. (1994) is characterized by the authors as being part of a toolkit "oriented toward simple solutions with a minimum number of parameters" and being somewhere between "curve-fitting, the traditional way to describe the input-output relationships, and deterministic modeling, which is an exhaustive description of the physical phenomena." Their approach is to utilize a "conceptual schema" as a modeling technique to represent the unit as an assembly of classical and elementary components. The behavior of each component is then modeled by a deterministic approach. This approach requires fewer parameters and experimental data compared with the models developed previously. In the parameter identification procedure, the "available experimental data," such as the condensing and evaporating temperatures, the possible subcooling, and, superheating, are required. Based on these experimental results, the parameters of the compressor are identified. Then the whole chiller is considered to identify the evaporator and condenser heat-transfer coefficients. However, those experimental data are normally not available from manufacturers' catalogs.

Parise (1986) developed a vapor compression heat pump simulation model to predict the overall performance of a system. Simple models for the components of the heat pump cycle were employed. Input data include compressor speed, displacement volume, clearance ratio, and other parameters for a comparatively detailed description of each component.

Cecchini and Marchal (1991) proposed a computer program for simulating refrigeration and air-conditioning equipment of all types: air-to-air, air-to-water, water-to-water, and water-to-air. Some parameters characterizing the components require experimental data from equipment testing, such as the heat exchanger mean surface temperatures, the saturation pressures in both evaporator and condenser, and superheating and subcooling. Again, these data are not typically provided in manufacturers' catalogs.

Fischer and Rice (1983) developed an air-to-air heat pump model to predict the steady-state performance of conventional, vapor compression heat pumps in both heating and cooling modes. The motivation for the development of this model is to provide an analytical design tool for use by heat pump manufacturers, consulting engineers, research institutions, and universities in studies directed toward the improvement of heat pump efficiency. The compressor was modeled using the data "curve-fits" and "compressor maps" provided by the manufacturers. Modeling of the other components employed many fundamental correlations, and detailed design data are required. Hence, while this model may be useful in the heat pump design process, it is difficult or impossible for an engineer working from catalog data to use. Similarly, the water-to-air heat pump model developed by Greyvenstein (1988) is also based on detailed component information, including the fan curve, compressor characteristics, heat exchanger geometry, etc.

The superheat–controlled water-to-water heat pump model developed by Stefanuk et al. (1992) may be the most detailed model presented to date. The authors claim, "the model is derived entirely from the basic conservative laws of mass, energy, momentum and equations of state as well as fundamental correlations of heat transfer." Values of the parameters that describe the behavior of the individual components are assumed to be available. For example, the parameters of the compressor are selected by "fitting the model to manufacturer-supplied performance curves that related mass flow rate and input electrical power to evaporation temperature and the compressor discharge pressure." However, they are not normally available in the heat pump manufacturers' catalogs. Comparisons between the experimental measurements and model predictions for the evaporating and condensing pressures, the heat transfer rates in the evaporator and the condenser, and the COP of the heat pump are given. Except for a few points with errors beyond ±10%, most of the results are generally acceptable. The predictions of the heat transfer rates in both heat exchangers are consistently too high. The authors explain that the cause for this phenomenon is the overestimated predictions of heat transfer coefficients since heat transfer coefficients used in the model are only known to within ±20%.

Damasceno et al. (1990) compared three steady-state air-to-air heat pump computer models. Two of them are available in the open literature—the third one was developed in-house. These are (1) the MARK III model, which is an updated version of an earlier program developed at Oak Ridge National Laboratory by Fischer and Rice (1983) and Fischer et al. (1988); (2) HPSIM, developed at NBS by Domanski and Didion (1983); and (3) HN, developed by Nguyen and Goldschmidt (1986) and updated by Damasceno and Goldschmidt (1987). All three models require extensive test data for calibration. A more extensive comparison and summary of the heat pump and chiller models developed so far are shown in Table 1.

Domanski and McLinden (1992) presented a simulation model called "Cycle-11." "Cycle-11" and its derived versions are models targeted at the preliminary evaluation of performance of refrigerant and refrigerant mixtures in the vapor compression cycle. Hence, to facilitate this function, the input to the program normally includes the outlet temperature for the heat-transfer fluids. In the UA version of "Cycle-11" (Domanski 2000), the author simplified the heat exchanger model to use a constant heat-transfer coefficient UA. However, the approach to obtain the value of this constant UA is not addressed explicitly. The input to the program also includes internal specifications for the components, such as the compressor swept volume, compressor speed, and electric-motor efficiency. The authors assumed that this information is already available. Some other internal pressure and temperature changes are calculated using simplified pressure-drop and heat-transfer correlations. They are obtained from a separate program and specified as inputs for the model.

Clearly, while there are a number of models available for vapor compression refrigeration systems, all that provide the advantages of the deterministic approach require detailed data beyond what is typically provided in manufacturers' catalogs. However, many users of such models only have access to catalog data. Therefore, it would be useful to have a model that uses a deterministic approach but only requires catalog data. Such a model has been developed and is the topic of this paper.

WATER-TO-WATER HEAT PUMP MODEL

The objective of this research effort is to develop a water-to-water heat pump model suitable for use in energy calculation and/or HVAC system simulation programs. Furthermore, it is desired that the model accurately duplicate the water-to-water heat pump performance, utilize catalog data for parameter estimation, require a minimal number of data points, and allow extrapolation. The modeling method employs mechanistic relations of fundamental thermodynamic laws and heat-transfer correlations with parameters identified from catalog data. There are numerous options for the selection of parameters. A number of combinations of parameters have been investigated, and the scheme that provides the best results (i.e., least relative error) is presented in detail. The other schemes are discussed and their results are compared with the final one.

System Description

The heat pump configuration used in this study is presented in Figure 1. The heat pump consists of four basic components: reciprocating compressor, evaporator, condenser, and expansion device. Other components are neglected due to the comparatively small contribution to the thermodynamic analysis for the entire system. Assuming an isenthalpic process in the expansion device and no heat exchange between this system and its environment, in case of the cooling mode, we have

$$\dot{Q}_S = \dot{W} + \dot{Q}_L \tag{1}$$

where $\dot{Q}_S$ is the source-side heat-transfer rate, $\dot{W}$ is the compressor power input, and $\dot{Q}_L$ is the load-side heat-transfer rate.

Equation 1 assumes that no heat is lost from the compressor when, in actuality, there will be some heat transferred from the compressor shell. Generally, this loss will be fairly small, and heat pump manufacturers' catalog data neglect this heat loss (i.e., the catalog data are consistent with Equation 1).

Compressor Model

The thermodynamic cycle for an actual single-stage system may depart significantly from the theoretical cycle. The principal departure occurs in the compressor. Hence, it is

TABLE 1
Literature Summary for Heat Pump and Chiller Models

Author	Year	Compressor				Evaporator and Condenser			Expansion Device
		Suction & Discharge Pressure Drop	**Compression**	**Re-expansion of Clearance Vapor**	**Shell Loss or Efficiency**	**Refrigerant Pressure Drop**	**Superheating, & Subcooling**	**Heat Transfer Coefficient**	
Stoecker and Jones	1982	Non-deterministic/component level equation-fit model							
Allen and Hamilton	1983	Non-deterministic/overall system level equation-fit model							
Fischer and Rice (Mark I) Revisions of Fischer and Rice's model, (Mark III to Mark V)	1982 to 1997	Curve-fit/map-based model				Yes	Yes	Variable	1) Capillary tube: equation-fit
		Deterministic/loss and efficiency-based model							2) TXV: general model and empirical correlations
		No	Isentropic	Yes	Yes				3) Short tube orifice: empirical correlations
Domanski and Didion Revision of Domanski and Didion's model	1984 to 2000	1) No	1) Isentropic	1) Zero clearance volume	Yes	Yes Specified as input	Yes Individual sections specified as input	1) Variable	Adiabatic Detailed model for constant flow area expansion device
			2) Polytropic						
		2) Yes Specified as input	3) Either of above with volumetric efficiency	2) Typical clearance volume				2) Assumed constant	
Parise	1986	No	Polytropic	Polytropic	No	No	Yes Account for the ΔT using arithmetic average with two-phase temp.	Assumed constant specified as input	Isenthalpic
Greyvenstein	1988	Interpolation of manufacturers' performance data				Yes	Yes	Variable	Adiabatic
Hamilton and Miller	1990	Non-deterministic/component level equation-fit model							
Cecchini and Marchal	1991	No	Polytropic	Polytropic	No	No	Assumed constant	Variable	Isenthalpic
Shelton and Weber	1991	Only kW/ton is computed							
Stefanuk et al.	1992	Yes	Isentropic	Isentropic	Yes	No	Yes	Variable	Adiabatic Isenthalpic
Bourdouxhe et al.	1994	No	Isentropic	Isentropic	Yes	No	No	Assumed constant	Adiabatic Isenthalpic
Gordon and Ng	1994	Only COP is computed							

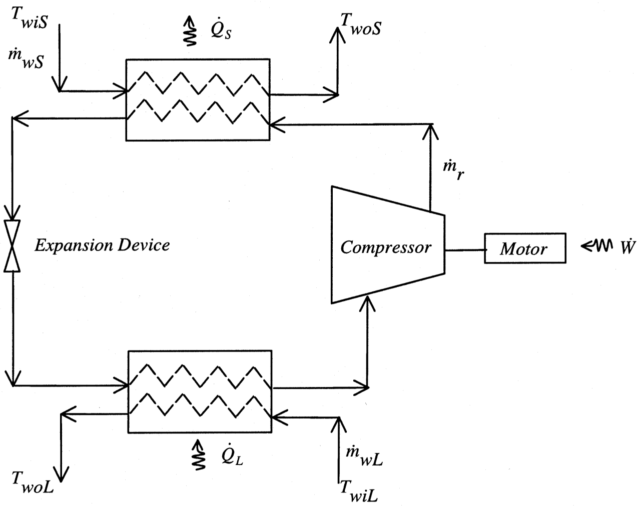

Figure 1 *Basic heat pump configuration.*

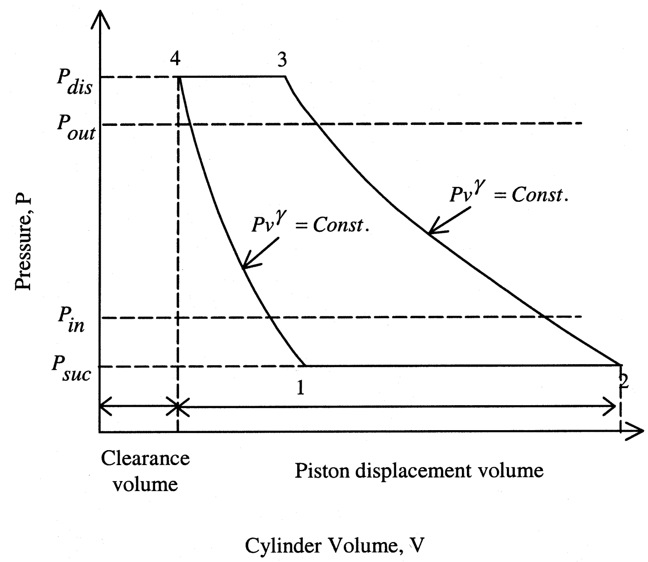

Figure 2 *Schematic indicator diagram for a reciprocating compressor.*

worthwhile to pay more attention to the thermodynamic processes occurring within a reciprocating compressor. The representation of the compressor cycle shown in Figure 2 follows that of Threlkeld (1962).

Several assumptions are incorporated into the modeling procedure.

- The modeled compressor cycle is only an approximation of the real compressor cycle.
- The compression and expansion in the compressor cycle are isentropic processes with equal and constant isentropic exponents.
- The isentropic exponent is dependent on the refrigerant type; the values of the isentropic exponents are obtained from Bourdouxhe et al. (1994), which are originally based on the study of Saavedra (1993).
- The oil has negligible effects on refrigerant properties and compressor operation.
- There are isenthalpic pressure drops at the suction and discharge valves.

For the compression process from point 2 to point 3, we will assume an isentropic process,

$$P_2 v_2^{\gamma} = P_3 {v_3}^{\gamma}, \tag{2}$$

where P is the pressure and v is the specific volume. We may further assume that the temperature change from point 3 to point 4 is negligible. With this approximation, the specific volume at point 3 is equal to the specific volume at point 4. If we denote the state of the reexpanded clearance vapor as 1, then, for the reexpansion process, we have

$$P_4 v_4^{\gamma} = P_1 v_1^{\gamma}. \tag{3}$$

Due to the reexpansion of the refrigerant vapor in the clearance volume, the mass flow rate of the compressor refrigerant is a decreasing function of the pressure ratio,

$$\dot{m}_r = \frac{PD}{v_{suc}}\left[1 + C - C\left(\frac{P_{dis}}{P_{suc}}\right)^{1/\gamma}\right], \tag{4}$$

where $\dot{m}_r$ is the refrigerant mass flow rate, PD is the piston displacement, v_{suc} is the specific volume at suction state, C is the clearance factor, P_{dis} is the discharge pressure, P_{suc} is the suction pressure, and γ is the isentropic exponent. The suction and discharge pressures play important roles in varying the magnitude of the theoretical mass flow rate. These two pressures are different from the evaporating and condensing pressures due to the pressure drop across suction and discharge valves. According to the discussion of Popovic and Shapiro (1995), the inclusion of pressure drops across the suction and discharge valves led to a more accurate prediction for their reciprocating compressor model. The pressure drop, ΔP, is also considered in the model. Results for versions of the model with and without pressure drops across the valves are presented below. Acceptable accuracy was not achieved in models that did not incorporate pressure drop.

An expression for the compressor work required may be derived subjected to the same approximations and limitations as used in the analysis above. The work is represented by the enclosed area of the diagram of Figure 2. Therefore,

$$\dot{W}_t = \int_2^3 VdP - \int_1^4 VdP. \tag{5}$$

The compressor model has a power requirement based on the thermodynamic work rate of an isentropic process.

$$\dot{W}_t = \frac{\gamma}{\gamma - 1}\dot{m}_r P_{suc} v_{suc}\left[\left(\frac{P_{dis}}{P_{suc}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \quad (6)$$

A simple linear representation has been used to account for the electrical and mechanical efficiency of the compressor. The actual power input for the compressor is calculated by the following equation:

$$\dot{W} = \eta \cdot \dot{W}_t + \dot{W}_{loss} \quad (7)$$

where $\dot{W}$ is the compressor power input, $\dot{W}_{loss}$ is the constant part of the electromechanical power losses, η is the loss factor used to define the electromechanical loss that is supposed to be proportional to the theoretical power, and $\dot{W}_t$ is the theoretical power.

The refrigerant mass flow rate is not given in heat pump manufacturers' catalogs. In this parameter estimation model, the refrigerant mass flow rate is estimated first based on the estimated values of the selected parameters for the compressor using Equation 4. The compressor power is then calculated accordingly using Equations 6 and 7. The enthalpy of the refrigerant at the suction state is determined from the evaporator model.

In addition, some superheat, ΔT_{sh}, is assumed to occur before entering the compressor, following Bourdouxhe et al. (1994). To summarize, there are six parameters that have to be identified for the compressor. They are PD, C, ΔP, W_{loss}, ΔT_{sh}, and η.

It is not expected from this model that the values of the selected parameters derived with a parameter estimation procedure will match the exact specifications of each component if they are readily available. The heat pump manufacturers' catalogs do not normally provide any compressor specifications, though they may be available from their compressor suppliers. However, proprietary information, such as clearance factor, is normally not available. If the exact values of a few of the selected parameters are available, those parameters could be eliminated from the list of the parameters to be estimated and their actual values could be used.

Due to the limitation of the available catalog data, it is virtually impossible that the estimated values of the parameters for each individual component will match the actual physical value.

Condenser and Evaporator Models

The condenser and evaporator models are developed from fundamental analysis of the counterflow heat exchangers. It is assumed that there is negligible pressure drop in the heat exchangers and, therefore, the refrigerant is at a constant temperature while changing phase. Since the temperature of the refrigerant in the two-phase region is considered constant, it is not necessary to differentiate whether the heat exchanger is actually counterflow or has some other configuration.

A more detailed approach, using the heat exchanger model of Rabehl et al. (1999) was tried, but it did not significantly improve the prediction accuracy. Instead, the heat exchanger was treated as a simple heat exchanger with phase change on one side.

$$\varepsilon = 1 - e^{-NTU} \quad (8)$$

where $NTU = \frac{UA}{\dot{m}_w C_{pw}}$, ε is the thermal effectiveness of heat exchanger, NTU is the number of transfer units, $\dot{m}_w$ is the mass flow rate of water, and C_{pw} is the specific heat of water. UA is the heat transfer coefficient, which is assumed to be a constant independent of the fluid temperatures and flow rates.

Therefore, the heat transfer coefficients UA of the condenser and the evaporator are the last two parameters to be identified. The entire cycle is shown in Figure 3.

A constant value of UA for the condenser and evaporator is clearly not physically correct, yet it seems to be a reasonable approximation given the overall goal of this model.

Furthermore, for the evaporator, the effect of refrigerant superheating is neglected in the evaporator model. If every other part of the heat exchanger model were "correct," this would result in underprediction of the evaporator heat transfer. However, this small systematic error is presumably compensated by a smaller estimated value of UA. Analogously, for the condenser, the neglect of the superheating and subcooling can also be compensated for by a smaller value of UA.

Expansion Device

The heat pump model does not explicitly model the expansion device. Rather, the amount of superheat is held constant and the refrigerant mass flow rate is determined by the compressor model. This is a round-about way of modeling (or assuming the presence of) a thermostatic expansion valve. All of the heat pumps investigated with this model do utilize thermostatic expansion valves. To the best of the authors'

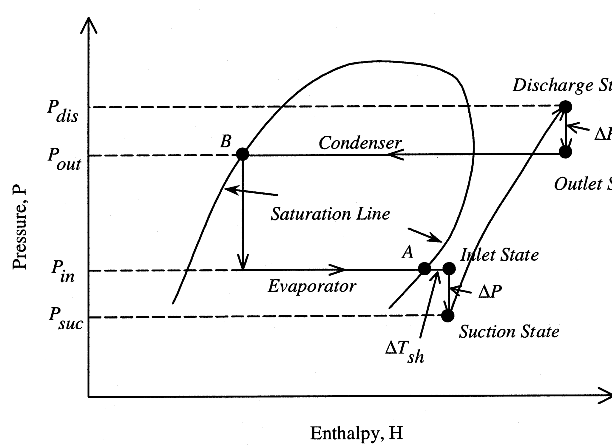

Figure 3 *Pressure-enthalpy diagram for the refrigeration cycle.*

knowledge, this is true for all water-to-water heat pumps manufactured in North America. Therefore, the model has not been tested with any other expansion devices, such as a capillary tube, and it may not be applicable for heat pumps with capillary tubes.

Parameter Estimation Procedure

The values of the parameters are estimated using the available catalog data. One set of parameters is estimated for the heating mode, and one set is estimated for the cooling mode. For each operating point, the data needed are:

- Source-side entering water temperature, flow rate, and heat rejection (cooling mode) or heating extraction (heating mode)
- Load-side entering water temperature, flow rate, and cooling capacity (cooling mode) or heat capacity (heating mode)
- Compressor power consumption

The parameter estimation procedure minimizes the difference between the model results and the catalog data by systematically adjusting the values of the parameters. The difference between the model results and the catalog data is quantified in the form of an objective function. For any given set of parameters, $PD, C, \Delta P, W_{loss}, \Delta T_{sh}, \eta, (UA)_L$, and $(UA)_S$, in the case of the cooling mode, the objective function is calculated as follows:

1. Calculate the evaporator and condenser effectiveness by Equations 9 and 10.

$$\varepsilon_L = 1 - \exp\left(-\frac{(UA)_L}{C_{pw}\dot{m}_{wL}}\right) \tag{9}$$

where ε_L is the thermal effectiveness of the evaporator and $(UA)_L$ is the heat-transfer coefficient for the evaporator.

$$\varepsilon_S = 1 - \exp\left(-\frac{(UA)_S}{C_{pw}\dot{m}_{wS}}\right) \tag{10}$$

where ε_S is the thermal effectiveness of the condenser and $(UA)_S$ is the heat-transfer coefficient for the condenser.

2. Calculate the evaporating and condensing temperatures of the refrigerant.

$$T_e = T_{wiL} - \frac{\dot{Q}_L}{\varepsilon_L C_{pw}\dot{m}_{wL}} \tag{11}$$

where T_e is the evaporating temperature and T_{wiL} is the evaporator entering water temperature.

$$T_c = T_{wiS} + \frac{\dot{Q}_S}{\varepsilon_S C_{pw}\dot{m}_{wS}} \tag{12}$$

where T_c is the condensing temperature and T_{wiS} is the condenser entering water temperature.

3. When the condensing and evaporating temperatures are obtained, the corresponding pressures and enthalpies can be derived using a refrigerant property subroutine. We used subroutines provided with an HVACSIM+ system simulation program (Clark and May 1985).

4. Identify the refrigerant state at the compressor suction port by adding the superheat to the evaporating temperature. The refrigerant enthalpy at this point is determined using the refrigerant property subroutines.

$$T_{icom} = T_e + \Delta T_{sh} \tag{13}$$

where T_{icom} is the compressor inlet temperature and ΔT_{sh} is the superheat.

5. Identify the compressor suction and discharge states by adding or subtracting the pressure drop. The specific volume at the suction state is determined by the refrigerant property subroutines.

$$P_{suc} = P_e - \Delta P \tag{14}$$

where P_e is the evaporating pressure and ΔP is the pressure drop across the suction valve.

$$P_{dis} = P_c + \Delta P \tag{15}$$

where P_c is the condensing pressure and ΔP is the pressure drop across discharge valve.

6. Calculate the refrigerant mass flow rate by Equation 4, the theoretical value of isentropic compression power by Equation 6, and the total power input by Equation 7.

7. Calculate the new value of the cooling capacity for cooling mode.

$$\dot{Q}_L = \dot{m}_r(h_A - h_B) \tag{16}$$

where h_A is the enthalpy of the refrigerant leaving the evaporator and h_B is the enthalpy of the refrigerant entering the evaporator. Or, for heating mode, the heating capacity is calculated using Equation 17.

$$\dot{Q}_L = \dot{W} + \dot{Q}_S \tag{17}$$

where $\dot{Q}_L$ is the heating capacity, and $\dot{Q}_S$ is the heat extraction that is determined through the same procedure as the cooling capacity $\dot{Q}_L$ in the cooling mode.

Based on the given values of the parameters, the power consumption for the compressor and the cooling capacity (cooling mode) or heating capacity (heating mode) are calculated for each operation point. Then it is possible to compare the calculated results with the catalog performance data. The relative error between the catalog data and the calculated results for the power consumption and the cooling capacity or the heating capacity should be small. This is achieved by searching for the minimum value of the following objective

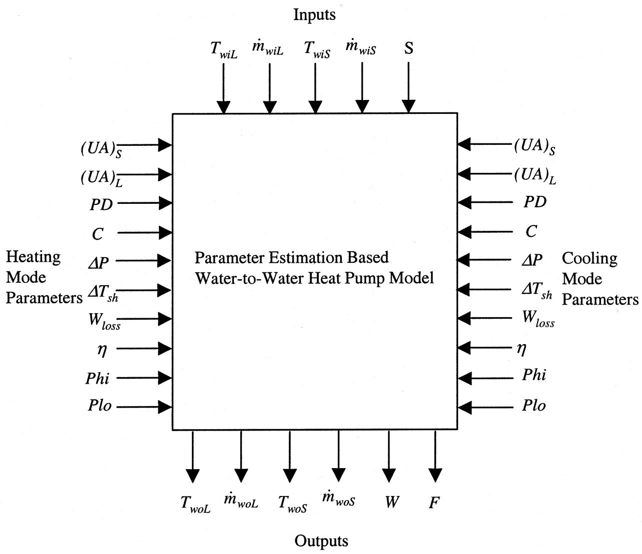

Figure 4 *Information flow chart for model implementation.*

function, which is the sum of the squares of the relative errors for both power consumption and load side heat transfer rate.

$$SSE = \sum_{i=1}^{N}\left[\left(\frac{(\dot{W}_{cat})_i - (\dot{W})_i}{(\dot{W}_{cat})_i}\right)^2 + \left(\frac{(\dot{Q}_{Lcat})_i - (\dot{Q}_{Li})_i}{(\dot{Q}_{Lcat})_i}\right)^2\right] \quad (18)$$

where $\dot{W}_{cat}$ is the catalog power consumption, $\dot{W}$ is the calculated power consumption, $\dot{Q}_{Lcat}$ is the catalog load-side heat transfer rate, and $\dot{Q}_{Li}$ is the calculated load-side heat-transfer rate.

The optimal parameter values for a particular heat pump will be those associated with the minimum value of function SSE. Hence, the search for the optimal values of the selected parameters becomes a multi-variable optimization problem. To solve this problem, the widely used Nelder-Mead simplex (Kuester and Mize 1973) method is employed. A multistart random sampling strategy was added to ensure the global minimum has been obtained.

Model Implementation

The model is implemented in nearly the same way as the objective function evaluation described above. However, a thermostat signal is used as an input parameter to tell the model which set of parameters (heating mode or cooling mode) should be used. Also, the objective function evaluation takes advantage of the fact that the heat transfer rates are known, using the catalog data as an initial guess, then minimizing the difference between the predicted and measured heat-transfer rates. However, for the model implementation, the heat-transfer rates are solved simultaneously with successive substitution, and this introduces an iterative loop not present in the objective function evaluation.

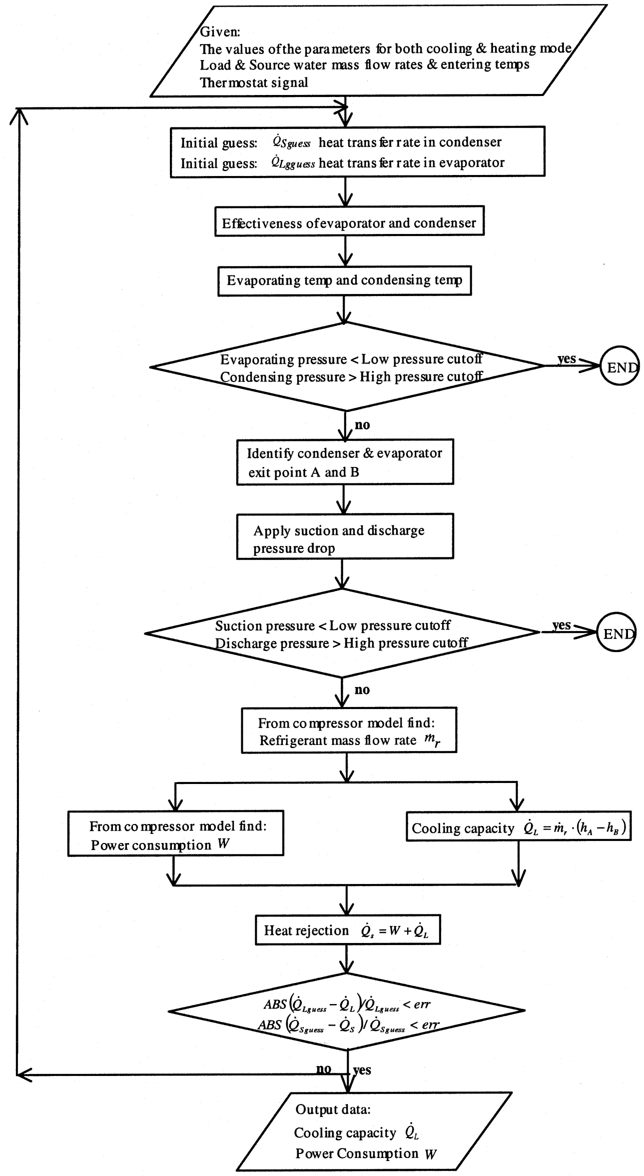

Figure 5 *Flow diagram for model implementation computer program.*

The model then determines the outlet temperatures for each of the fluid streams. Other information, such as cooling and heating capacities, COP, etc., may be reported if desired. An information flow chart of the model implementation, is presented in Figure 4. Figure 5 shows the cooling mode of the model implementation. This modeling approach has the advantage of not requiring experimental and component specification data beyond what are published in heat pump manufacturers' catalogs. Since the objective of this model is to eliminate the requirement for any internal measurements, such as the refrigerant mass flow rate, temperature, and pressure, all the comparisons are made based on the external measurements of water flow rates and temperatures for both source and load sides. These data are readily available in the heat pump manu-

TABLE 2
Range of Water Flow Rates and Entering Water Temperatures

No.	Load Side		Source Side	
	EWT	Flow Rate	EWT	Flow Rate
1 (cooling)	25ºF (–3.89ºC) to 65ºF (18.33ºC)	4 GPM (0.25 kg/s) to 7 GPM (0.44 kg/s)	55ºF (12.78ºC) to 95ºF (35ºC)	4 GPM (0.25 kg/s) to 7 GPM (0.44 kg/s)
2 (cooling)	25ºF (–3.89ºC) to 65ºF (18.33ºC)	20 GPM (1.26 kg/s) to 36 GPM (2.27kg/s)	55ºF (12.78ºC) to 95ºF (35ºC)	20 GPM (1.26 kg/s) to 36 GPM (2.27kg/s)
3 (heating)	80ºF (26.67ºC) to 120ºF (48.89ºC)	4 GPM (0.25 kg/s) to 7 GPM (0.44 kg/s)	25ºF (–3.89ºC) to 65ºF (18.33ºC)	4 GPM (0.25 kg/s) to 7 GPM (0.44 kg/s)
4 (heating)	60ºF (15.56ºC) to 120ºF (48.89ºC)	45 GPM (2.84 kg/s) to 90 GPM (5.68 kg/s)	10ºF (–12.22ºC) to 70ºF (21.11ºC)	45 GPM (2.84 kg/s) to 90 GPM (5.68 kg/s)

TABLE 3
RMS Errors of the Simulations for Four Sets of Catalog Data

No.	Unit	Nominal Capacity		Number of Points	RMS	
		(W)	(Btu/h)		Capacity	Power
1	A	7,034 (cooling)	24,000 (cooling)	81	4.57%	4.77%
2	B	43,965 (cooling)	150,000 (cooling)	81	4.71%	5.44%
3	A	7,620 (heating)	26,000 (heating)	81	2.66%	1.70%
4	C	99,654 (heating)	340,000 (heating)	234	3.08%	5.76%

facturers' catalogs. The ranges of flows and inlet temperatures in the catalog data for the heat pumps investigated are presented in Table 2.

Treatment of Extreme Operating Conditions

When this model is used inside of a transient system simulation program, it may encounter conditions not intended by the manufacturer, such as low water flow rates or extreme temperatures. This may happen even when the system simulation inputs are correct, as the equation-solving process may occasionally try physically unrealistic values. Without any other checks, the model may then provide unrealistic results or crash due to errors in the property routines.

In order to avoid this problem, a check is provided that is analogous to what happens in real heat pumps. Real heat pumps are usually equipped with protection against overly high pressures or overly low pressures that switch the compressor off when the limits are exceeded. These have been replicated in the model by incorporating a minimum evaporator pressure P_{lo} and maximum condenser pressure P_{hi} that may be set as parameters. If either of these are exceeded, the heat pump is turned off—outlet fluid temperatures are set equal to inlet fluid temperatures and power is set to zero.

Model Validation

The water-to-water heat pump model was validated using catalog data for three randomly selected heat pumps made by two different manufacturers. In Table 3, a summary of the comparisons for units A and B, validated using the cooling mode data, and units A and C, validated using the heating mode data, is given. The heat pump capacities, number of operating points given in the manufacturer's catalog, and the root-mean-square (RMS) error for capacity and power are shown in Table 3. The comparison showed a comparatively good agreement with generally acceptable accuracy. For the model of Stefanuk et al. (1992), Stefanuk (1990) reports errors between the model and the experimental data. Measured condenser heat-transfer rates are, on average, 12% lower than the predicted rates. Measured evaporator heat-transfer rates are, on average, 17% lower than the predicted rates. However, when the model is adjusted with a physically measured refrigerant mass flow rate, the errors are significantly reduced to between 3% and 5% on average. As can be seen, the errors in Table 3 compare very favorably with the Stefanuk model, considering that making internal measurements is impractical for most engineers using simulation programs.

A comparison of the results to the catalog data for the heat pump with the least satisfactory match is shown in Figures 6 and 7. A comparison of the results to the catalog data for the heat pump with the best match is shown in Figures 8 and 9.

Discussion of Parameter Selections

The model, as presented above, represents the final step in a series of incremental modifications. With each modification, new parameters were introduced or, in some cases, old parameters were removed. Some insight as to the relative importance of the different parameters may be gleaned from Table 4, which shows the estimated parameters and the SSE for seven different versions of the model, applied to a single heat pump in cooling mode. The catalog data used for the study are for the heat pump A in cooling mode in Table 3.

It should be noted that the parameter estimation procedure can consume a significant amount of computer time. There is a trade-off between the number of parameters estimated and the required computational time. There are eight parameters selected for the final solution, scheme 7 in Table 4. For the case of heat pump C in Table 3, the parameter estimation process takes approximately 95 minutes computing time for a PC with a 333MHz CPU if all 234 operating points are used. If 16 points are used, the computing time is about 9 minutes. As will be demonstrated in the next section, 16 points should be sufficient. It might also be noted that once all the parameters are known, the model can be executed on the same machine in about 0.0025 seconds for a single operating condition.

For the compressor model, the selection of $\dot{W}_{loss}$, the constant part of the electromechanical power losses, and η, the loss factor that defines the electromechanical loss proportional to the theoretical power, contributes to the greatest improvement in the model from scheme 1 to scheme 2. An alternative improvement was the implementation of more detailed heat exchanger models (Rabehl et al. 1999), as shown in scheme 3. Since the computer time required for estimation

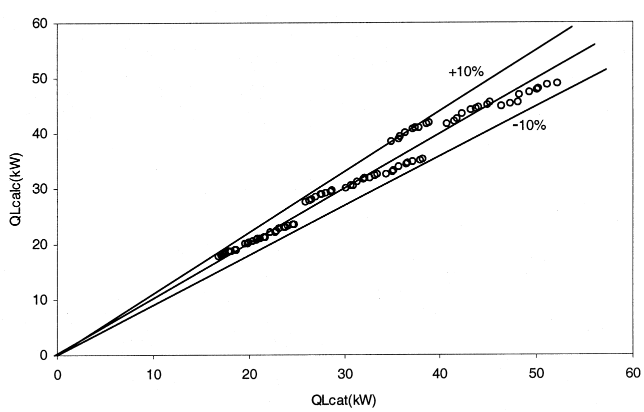

Figure 6 *Calculated cooling capacity versus catalog cooling capacity (heat pump B).*

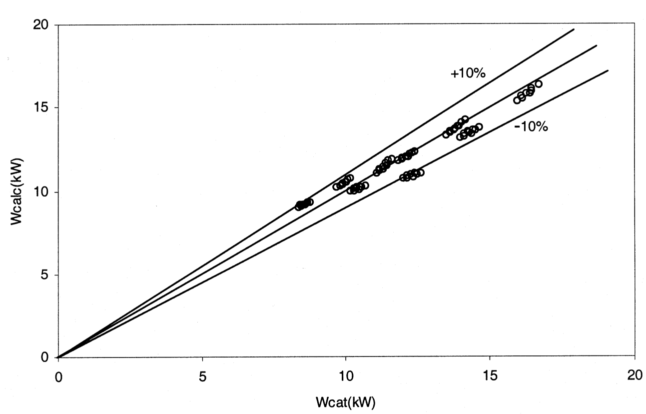

Figure 7 *Calculated power versus catalog power (heat pump B).*

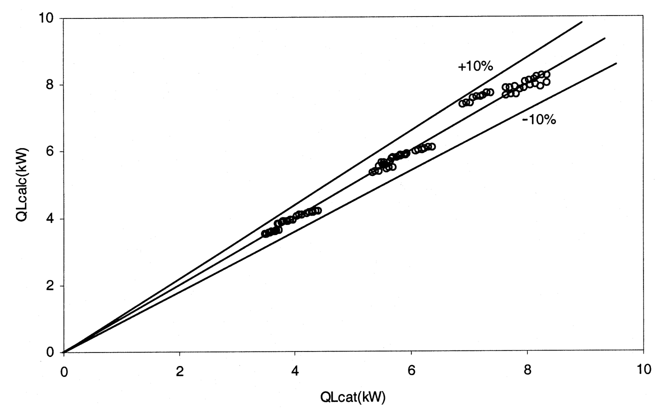

Figure 8 *Calculated heating capacity versus catalog heating capacity (heat pump A).*

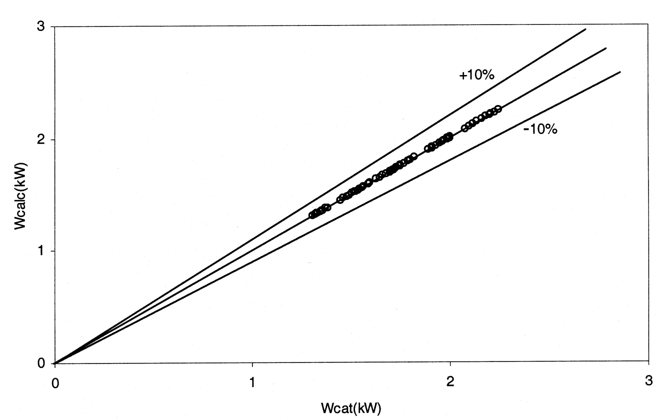

Figure 9 *Calculated power versus catalog power (heat pump A).*

TABLE 4
Comparison of the Search Results of the Objective Function

Scheme			1	2	3	4	5	6	7
	P.D.	(m^3/s)	0.00205	0.00107	0.00459	0.00176	0.00137	0.00177	0.00175
		(ft^3/h)	260.34	135.89	583.86	223.73	174.53	225.00	222.46
Parameter Estimation Model	C		0.0213	0.0499	0.0495	0.0498	0.0492	0.0482	0.0463
	η		-	0.381	-	0.541	0.470	0.575	0.824
	W_{loss}	(kW)	-	0.0000425	-	0.0129	0.00383	0.00919	0.586
		(Btu/h)	-	0.145	-	43.94	13.06	31.36	1998.29
	$(UA)_{totev}$	(kW/K)	0.239	1.770	-	2.193	2.117	1.986	2.156
		Btu/(h·°F)	452.67	3353.26	-	4156.10	4013.36	3765.42	4087.10
	$(UA)_{totcon}$	(kW/K)	0.573	0.305	-	0.730	0.671	0.803	1.469
		Btu/(h·°F)	1085.23	578.14	-	1383.03	1272.37	1521.41	2784.64
	Suction ΔP	(kPa)	-	-	-	98.414	-	103.928	97.937
		(psi)	-	-	-	14.27	-	15.07	14.20
	Discharge ΔP	(kPa)	-	-	-	-	101.123	101.693	97.937
		(psi)	-	-	-	-	14.67	14.75	14.20
	ΔT_{sh}	(ºC)	-	-	-	-	-	-	7.078
		(ºF)	-	-	-	-	-	-	44.74
	Evap.	C_1	-	-	300.139	-	-	-	-
		C_2	-	-	–0.140	-	-	-	-
		C_3	-	-	0.250	-	-	-	-
	Cond.	C_4	-	-	700.301	-	-	-	-
		C_5	-	-	0.123	-	-	-	-
		C_6	-	-	0.275	-	-	-	-
SSE			23.907	2.331	6.769	1.111	1.594	0.943	0.419

with more than eight parameters was rather high, and the benefit in terms of improving the accuracy of the model was rather low, the detailed heat exchanger representation, which requires six parameters, was abandoned.

Then, various combinations of suction and discharge pressure drops were added in schemes 4 through 6. It was noted that the separately estimated values of ΔP for both the suction side and discharge side were very nearly equal, so the final number of parameters was held to eight by estimating a single ΔP that is applied to both the suction side and discharge side.

Given the approximations in the parameter estimation procedure and the fact that the approximations are compensated by artificially high or low values of parameters, it might be expected that estimated values of parameters for a given heat pump in heating mode and in cooling mode might be significantly different. Comparing the estimated parameters for data sets 1 and 3 may be instructive, since they represent the same heat pump operating in heating mode and cooling mode. However, the estimated parameters are surprisingly close. For the load-side heat exchanger, the estimated value of *UA* for heating mode is 2.21 kW/°C (4,189 Btu/[h·°F]), and for cooling mode it is 2.10 kW/°C (3,981 Btu/[h·°F]). For the source-side heat exchanger, it is 1.54 kW/°C (2,919 Btu/[h·°F]) and 1.46 kW/°C (2,768 Btu/[h·°F]), respectively. Table 5 presents a comparison of the parameter estimation results of cooling and heating modes for the same heat pump.

It should be recognized that the development of the model is somewhat of an art. The selection of more sophisticated representation for a single component may not benefit the results in a manner proportional to the increase in computational cost. This seems to be the case for the heat exchangers.

COMPARISON TO AN EQUATION-FIT MODEL

No generally accepted equation-fit model for water-to-water heat pumps is found in the literature. Due to the similarity between heat pump and chiller models, the steady-state reciprocating chiller model recommended by Allen and Hamilton (1983) and the centrifugal and absorption water chiller model proposed by Stoecker et al. (1975) have been adapted to establish an equation-fit model for a water-to-water heat pump. Allen and Hamilton fit the cooling capacity and power consumption to a second order polynomial in two variables— evaporator and condenser leaving water temperatures. In the chiller model proposed by Stoecker et al. (1975), the cooling capacity is also a function of two variables—the evaporator leaving water temperature and the condenser entering water temperature. In the equation-fit model used here, the condenser and evaporator entering water temperatures are chosen as the variables in the polynomial representation. This is more convenient for a component model when the inlet temperatures are known. This is also more convenient for equation fitting, since the heat pump catalog data are typically given in terms of four variables: load-side entering water temperature, source-side entering water temperature, load-side water flow rate and source-side water flow rate. Accordingly, the water flow rates for both sides are included as the other two variables in the equation-fit model.

As discussed by Stoecker (1975), there is not a "best" equation for the simulation of heat pumps or chillers. A polynomial representation may be the best choice when no physical insight into the performance is available. Different combinations of variables and coefficients were tried to find the best equation form for a particular set of catalog data selected for comparison purposes. The final functional relationships for cooling capacity and compressor power are implemented with a second order polynomial in four variables, with eleven and thirteen coefficients, respectively:

$$\begin{aligned} P = {} & C_1 + C_2 \cdot T_{Li} + C_3 \cdot T_{Li}^2 + C_4 \cdot T_{Si} \\ & + C_5 \cdot T_{Si}^2 + C_6 \cdot \dot{m}_L + C_7 \cdot \dot{m}_L^2 + C_8 \cdot \dot{m}_S \\ & + C_9 \cdot \dot{m}_S^2 + C_{10} \cdot T_{Li} \cdot \dot{m}_L + C_{11} \cdot T_{Si} \cdot \dot{m}_S \end{aligned} \tag{19}$$

$$\begin{aligned} Q = {} & C_{12} + C_{13} \cdot T_{Li} + C_{14} \cdot T_{Li}^2 + C_{15} \cdot T_{Si} \\ & + C_{16} \cdot T_{Si}^2 + C_{17} \cdot \dot{m}_L + C_{18} \cdot \dot{m}_L^2 + C_{19} \cdot \dot{m}_S \\ & + C_{20} \cdot \dot{m}_S^2 + C_{21} \cdot T_{Li} \cdot \dot{m}_L + C_{22} \cdot T_{Si} \\ & \cdot \dot{m}_S + C_{23} \cdot T_{Li} \cdot T_{Si} + C_{24} \cdot \dot{m}_L \cdot \dot{m}_S \end{aligned} \tag{20}$$

where P is the power consumption, Q is the heating or cooling capacity, T_{Li} is the load-side entering water temperature, T_{Si} is the source-side entering water temperature, $\dot{m}_L$ is the load-side water mass flow rate, and $\dot{m}_S$ is the source-side water mass flow rate. C_1 through C_{24} stand for the quadratic equation-fit constants as they are in the expressions.

The parameter estimation procedure has two significant advantages over the equation-fit model—improved fidelity to the catalog data and improved extrapolation beyond the catalog data. Heat pump C in Table 3 is selected for comparison purpose. There are 234 operating points in the heating mode given in the catalog for the selected heat pump. The 234 points cover a range of entering water temperatures (EWT) and mass flow rates on both sides of the heat pump. In order to compare the parameter estimation model to the equation fit model, parameters or coefficients were determined for each model using all 234 points, all of the points except the 45 with the highest load-side EWT, all of the points except the 63 with the lowest load-side EWT, all of the points except the 45 highest and 63 lowest load-side EWT, and 16 points representing combinations of the highest and lowest values of the EWT and mass flow rates on each side. Once the parameters or coefficients were determined, each model was applied to all 234 operating points. The maximum relative error, average absolute error, and RMS error were calculated for all 234 operating points.

TABLE 5
Comparison of the Parameter Estimation Results for Heat Pump A in Cooling and Heating Modes

Parameter	Cooling	Heating
Piston Displacement	0.00175 m³/s (222.46 ft³/h)	(0.00162 m³/s) 205.93 ft³/h
Clearance Factor	0.0463	0.0690
Loss Factor	1.214	1.437
Constant Loss	0.586 kW (1,999.32 Btu/h)	0.525 kW (1,791.20 Btu/h)
Pressure Drop	97.937 kPa (14.20 psi)	99.29 kPa (14.40 psi)
Superheat	7.077°C (12.74°F)	9.82 °C (17.68°F)
Source Heat Transfer Coefficient	1.46 kW/°C (2,768 Btu/(h·°F))	1.54 kW/°C (2,919 Btu/(h·°F))
Load Heat Transfer Coefficient	2.10 kW/°C (3,981 Btu/(h·°F))	2.21 kW/°C (4,189 Btu/(h·°F))

TABLE 6
Comparison of the Relative Error of Power Consumption Simulated by Parameter Estimation and Equation-Fit Models

Catalog Data Used	Relative Error					
	Max. (abs. value)		Average (abs. value)		RMS	
	Parameter Estimation	Equation-Fit	Parameter Estimation	Equation-Fit	Parameter Estimation	Equation-Fit
Entire 234 Points	16.06%	29.12%	4.21%	6.33%	5.76%	7.84%
w/o 45 Highest Points	17.49%	25.54%	4.40%	6.73%	6.13%	8.56%
w/o 63 Lowest Points	30.82%	36.94%	4.60%	6.49%	6.93%	8.54%
w/o 45 Highest & 63 Lowest Points	30.41%	37.64%	4.56%	6.07%	6.86%	8.53%
16 Points	21.96%	33.46%	4.19%	6.85%	5.77%	8.54%

TABLE 7
Comparison of the Relative Error of Heating Capacity Simulated by Parameter

Catalog Data Used	Relative Error					
	Max. (abs. value)		Average (abs. value)		RMS	
	Parameter Estimation	Equation-Fit	Parameter Estimation	Equation-Fit	Parameter Estimation	Equation-Fit
Entire 234 Points	8.17%	38.50%	2.46%	7.41%	3.08%	9.65%
w/o 45 Highest Points	8.31%	35.72%	2.44%	7.96%	3.00%	10.81%
w/o 63 Lowest Points	8.00%	32.36%	2.71%	11.36%	3.26%	13.64%
w/o 45 Highest & 63 Lowest Points	8.00%	40.93%	2.49%	7.56%	3.03%	10.57%
16 Points	8.77%	49.24%	2.80%	12.55%	3.39%	16.45%

The comparisons of the results are summarized in Tables 6 and 7. For both heating capacity and power consumption, for every combination of operating points used to estimate parameters or coefficients, and for every characterization of the error, the parameter estimation model performs better than the equation fit model. For both results, but particularly for the heating capacity, the parameter estimation extrapolates much better (shown in the second, third, and fourth rows) and performs much better with a very limited data set (shown in the fifth row). In fact, the parameter estimation model performs almost as well with 16 data points as with 234 data points. This represents a significant advantage if the model user has to manually transcribe the data from the catalog! The idea to use 16 data points representing the combination of the lowest and highest values of the input variables came from Rabehl et al. (1999), who applied it in their heat exchanger model.

CONCLUSIONS

This paper has presented a water-to-water heat pump model suitable for use in building energy analysis and HVAC system simulation programs. The model has been developed so as to require only commonly available data from manufacturers' catalogs in order to estimate the model coefficients. As compared to more detailed deterministic models, it does not require internally measured data usually unavailable to building system designers and simulationists. It also works well with only 16 data points for each mode, making it reasonably convenient when the data must be manually transcribed from a catalog.

Furthermore, the model's performance compares favorably against the most detailed deterministic model previously published, having a similar RMS error to the model described by Stefanuk et al. (1992). As compared to equation-fit models, this model retains the physically based representation of the heat pump, which allows some extrapolation beyond the catalog data. It also performs significantly better when a limited number of operating points are utilized for estimation of parameters or coefficients.

The modeling approach taken here is readily extensible to other heat pump configurations. Future work will address water-to-air heat pumps, scroll and rotary compressors, and the effects of water-antifreeze mixtures on heat pump operation.

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NOMENCLATURE

Symbols

C = clearance factor

C_p = specific heat, J/(kg·K) or Btu/(lbm°F)

h = enthalpy, J/kg or Btu/lbm

$\dot{m}$ = mass flow rate, kg/s or lbm/h

NTU = number of transfer units

P = pressure, Pa or psia

PD = piston displacement, m^3/s or cfm

$\dot{Q}$ = heat transfer rate, W or Btu/h

S = thermostat signal

T = temperature, °C or °F

UA = heat transfer coefficient, W/K or Btu/(h·°F)

v = specific volume, m^3/kg or ft^3/lbm

$\dot{W}$ = compressor power input, W or Btu/h

γ = isentropic exponent

η = loss factor used to define the electromechanical loss that is supposed to be proportional to the theoretical power

ε = thermal effectiveness of heat exchanger

ΔT_{sh} = superheat, °C or °F

ΔP = pressure drop across suction or discharge valve, Pa or psia

Subscripts

A = A state point in refrigeration cycle

B = B state point in refrigeration cycle

c = condensing state

com = compressor

cat = catalog data

dis = discharge state

e = evaporating state

i = inlet condition or *i*th calculated result

in = compressor inlet state

L = load side

$loss$ = constant part of the electromechanical power losses

o = outlet condition

out = compressor outlet state

r = refrigerant

S = source side

t = theoretical power

suc = suction state

w = water

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