

MODELING OF ROOFTOP PACKAGED
AIR CONDITIONING EQUIPMENT

by

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Abstract

A large amount of the operating costs of a building is determined by the energy requirements and the fuel consumptions of its heating, ventilation and air conditioning system (HVAC-system). Simulation programs are used to determine the energy use in buildings. These simulation programs are based on models for the type of HVAC equipment used and models for determining the internal and external loads of the building. In this project a model for rooftop packaged air conditioning units has been developed.

The model uses generally available manufacturer's catalog data information to determine the model parameters. Thus, the model is easy applicable. The system is modeled by describing the performance of each component with mechanistic models. This method has been successfully used to predict the performance of rooftop packaged air conditioning systems and allows certain extrapolation of performance data.

The model has been developed with the Engineering Equation Solver (EES) software package. A program has been written that allows determining the characteristic model parameters based on manufacturer's performance data and also allows creating extrapolated performance data for different operating conditions. Originally developed with EES the program can be also called directly by TRNSYS, a transient simulation program.

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Special thanks go to my advisors Professor William Beckman and Professor John Mitchell, who always kept me motivated during slow times of the project. They were always very helpful and had many great ideas for this project. This work could not have been accomplished without Prof. Sandy Klein and his Engineering Equation Solver (EES). Thus, even for a student coming from the “SI-world”, as Professor John Mitchell used to say in his lectures, the difficulties of doing calculations in the English unit system were overcome. Meanwhile, for myself 1 ton rather represents 12,000 Btu/hr of refrigeration capacity, which is approximately 3.5 kW, than representing the mass of 2000 pounds.

I am really thankful to the professors of the Solar Energy Laboratory for letting me further pursue my education at the Solar Lab. Not only did I further improve my engineering skills, but when preparing for the seminars I also significantly enhanced my baking skills. Also, I have greatly enjoyed the friendly environment in the Solar Lab, having met many

students from the country and around the world, too many to mention them all. I will definitely take many good memories of that time with me.

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Nomenclature

a	parameter for compressor efficiency
A	area
A_o	outer heat transfer area
b	parameter for compressor efficiency
C	clearance factor
C_0	coefficient in heat transfer correlation
$c_{1..3}$	coefficients in polynomial expression E
$C_{1..9}$	heat exchanger parameter
$\bar{c}_{p, fictitious}$	fictitious specific heat
C_f	coefficient in heat transfer correlation
C_{max}	maximum capacitance rate
C_{min}	minimum capacitance rate
c_p	specific heat at constant pressure
C_r	capacitance rate ratio
c_s	saturation specific heat
c_v	specific heat at constant volume
D	tube diameter
D_h	hydraulic diameter
Δh_r	enthalpy difference in condensing region

Δp	pressure drop parameter
E	polynomial expression
ε	heat exchanger effectiveness
F	coefficient in heat transfer correlation for condensation
η_0	fin efficiency
η_{comb}	combined efficiency
η_v	volumetric efficiency
g	gravitational constant
$h_{1..4}$	enthalpies for state points
h_a	enthalpy of air
h_c	convection coefficient
h_{fg}	enthalpy of vaporization
h_{sat}	enthalpy of saturated air
hA	overall conductance area product
hA_i	inner conductance area product
hA_o	outer conductance area product
k	thermal conductivity
m	exponent in heat transfer correlation
\dot{m}	mass flow rate
m^*	capacitance rate ratio for wet coil analysis
μ	dynamic viscosity

n	polytropic exponent
N	number of data points
NTU	number of transfer units
NTU^*	number of transfer units used in the wet coil analysis
NTU_i	number of transfer units for inner heat transfer
NTU_o	number of transfer units for outer heat transfer
ν	kinematic viscosity
Nu_D	Nusselt number
$OF_{\dot{m}}$	objective function for mass flow rate
OF_P	objective function for power
$OF_{\dot{Q}}$	object function for heat transfer rate
ω	humidity ratio of air
P	power
p	pressure
Pr	Prandtl number
\dot{Q}	heat transfer rate
R_{tube}	thermal resistance of the tube
R_{com}	compression ratio
Re_D	Reynolds number
ρ	density
RPM	motor speed

T	temperature
u	velocity
U^*A	overall heat transfer area coefficient product based on enthalpies
UA	overall heat transfer area coefficient product
UA_i	inner overall heat transfer area coefficient product
v	specific volume
$v_{a..d}$	specific volume at state point a to d
$V_{a..d}$	volume at state point a to d
V_{disp}	displacement volume
u_{std}	face velocity
W	work
x	refrigerant quality
z	parameter in expression for compressor power draw of scroll compression process

Additional Subscripts

a	air
act	actual
c	cold
calc	calculated
cat	catalog

comp	compressor
cond	condenser
db	dry bulb
dis	discharge
dp	dew point
evap	evaporator
f	liquid
g	vapor
h	hot
i	at inlet state
max	maximum
mean	mean
min	minimum
o	at outlet state
r	refrigerant, at refrigerant temperature
s	at surface temperature
sat	saturated
sens	sensible
suc	suction
th	theoretical
tot	total
w	water

wb

wet bulb

Chapter 1

Introduction

1.1 The Need for Air Conditioning

Although some people might regard air conditioning as a luxury, in certain areas having air conditioning is a must. One does not feel comfortable if the temperature and humidity level is too high. Air conditioning affects not only personal comfort, but also economics. If people feel comfortable, their productivity is generally better than if they work under uncomfortable conditions. Thus, air conditioning is and will be more and more needed in our world.

1.2 Project Outline

When designing an air conditioning system for a new building or replacing an air conditioning system in an old building, it is necessary to know which air conditioning system or which products would operate best and be most cost effective. One convenient way to figure out the best solution for the system design is to run computer simulations, in which the whole building with its equipment is modeled. Therefore, models have to be available that predict the performance of air conditioning equipment under different operating conditions.

The goal of this project is to develop a model of a rooftop packaged air conditioning system suitable for use in simulations. The model uses available manufacturer's catalog data only. The model should fit the catalog data accurately and allow a satisfactory extrapolation over a wide range of operating conditions.

To develop the model, the different components of the system such as compressor, evaporator, condenser and expansion valve have to be modeled. Different approaches exist to model the behavior of those components.

One method for modeling the performance of a component is to use a curve fit approach using a polynomial representation. It is easy to represent the characteristics of a component, given that there are enough catalog points to get a good curve fit. The resulting equations, however, have no physical meaning, and a prediction of the component performance outside the range of the fitted data may be very inaccurate.

Another modeling technique is to describe the behavior of a component using mechanistic models. This requires a detailed knowledge of characteristic parameters such as geometric specifications, fin efficiencies, or other quantities. Such detailed information is generally not found in manufacturer's catalogs.

To avoid the problems of uncertain extrapolation with a polynomial representation and to accommodate the lack of specific information about the system's components for a complex model, a semi-mechanistic modeling technique is used in this project. The characteristic performance parameters for each component will be first defined. Those performance parameters contain all specific characteristics of the system's components. Parameter values are then obtained by fitting the model to the catalog data. The resulting model should allow reasonable extrapolation over a wide operating range.

Listed below are the steps that were followed to achieve the project goal:

- Performance data of rooftop packaged air conditioning units from different manufacturers were collected. Available manufacturer's data were checked for use in the model. The

manufacturer's data should be given over a wide range of operating condition to allow performance prediction of the model and for the extrapolation ability of the model to be checked.

- The literature was searched for modeling techniques for the whole system as well as for the single components used in a rooftop packaged unit.
- Performance data were collected for the individual components (compressor, evaporators, and condensers) used in rooftop packaged units. The developed models for these components were evaluated by using these data. Thus, models could be used in the rooftop packaged air conditioning unit model.
- A general model for rooftop packaged units based on manufacturer's catalog data to predict performance was developed. Each of the system components is modeled.
- The model of the rooftop packaged air conditioning unit was evaluated over the whole operating range and extrapolation ability of the model was tested.
- The number of data points used to predict system's performance accurately was determined.

A software package was developed with the Engineering Equation Solver that allows easy computation of extrapolated catalog data for rooftop packaged air conditioning units with different capacities. Also, it allows this model to be called by TRNSYS, which is a transient simulation program to perform energy calculations.

Chapter 2

System description

2.1 Air Conditioning Systems

Air conditioning systems can be divided into packaged air conditioning systems and central air conditioning systems. Usually, central air conditioning systems are used for entire buildings where the total cooling load is very high. Packaged units are often used if cooling demand is lower and cooling is needed for just a few rooms.

In 1995, a survey was made by the Energy Information Administration of the Department of Energy of the United States, in which the end-use equipment for space cooling was recorded for commercial buildings. Figure 2.1 shows the end-use equipment for space cooling as a percentage of the total cooled floor area in commercial buildings.

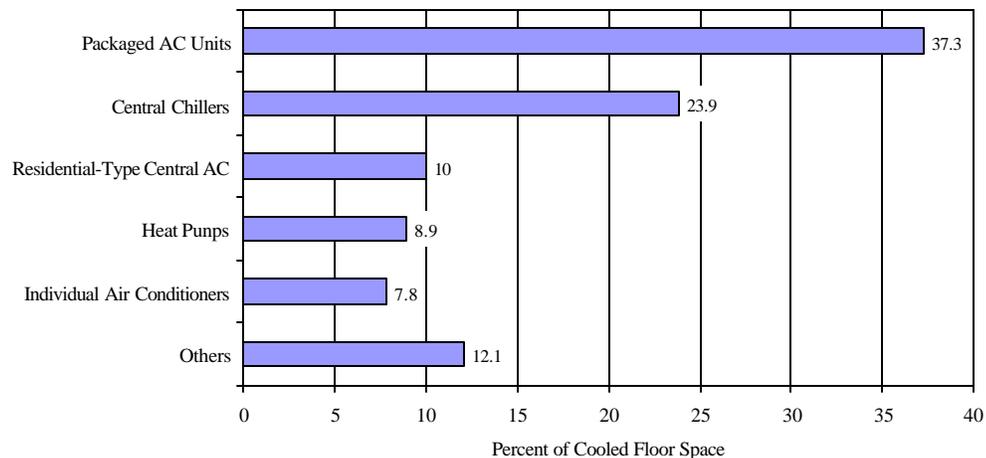


Figure 2.1: Survey of cooling equipment used in commercial buildings 1995

According to Figure 2.1, unitary packaged systems are by far the most commonly used air conditioning equipment in commercial buildings in the United States. 1,242,000 buildings out of 3,381,000 buildings were cooled by packaged air conditioning units. This corresponds to 37.3 % of the total cooled floor area of 3344 million square meters (36,001 million square feet) in commercial buildings. 23.9 % of the total floor area was cooled by central chillers. Only 96,000 commercial buildings had a central chiller as cooling equipment, which reflects that central chillers are mainly used if higher cooling capacity is required.

2.2 Rooftop Packaged Units

A packaged air conditioning unit is a self-contained air conditioner. It provides the cooling, heating and the motion of the air. Packaged units can be classified into three general types of units, the indoor packaged units, the split packaged units and the rooftop packaged units. The indoor and split packaged units differ from rooftop packaged units in arrangement and kind of equipment used. They consist of two main modules, an indoor air handling unit and an outdoor condensing unit. The condensing units are often water-cooled. This project is concentrated on rooftop packaged air conditioning units only. Thus, a generalized model can be developed for this type of unit, since the components used in rooftop packed units, such as compressor or condenser types, are mostly the same types.

Rooftop packaged units come as one single package that is ready to be mounted on a rooftop. All the components needed for cooling, heating, and air movement are assembled in

a steel casing. Rooftop packaged units can be classified according to the type of heating they supply. There are rooftop packed units with either electric heating or gas-fired heating. The heating can also be provided by a heat pump. However, electric heat and gas-fired furnaces are mainly used. The available cooling capacity of common packaged rooftop units ranges from 10 kW (3 tons) to 850 kW (241 tons). The air flow rate covers a range from 400 l/s (850 ft³/min) to 37,800 l/s (80,000 ft³/min).

Rooftop packaged units are equipped either with a supply fan only or, if needed, an additional return air fan. As the air is delivered by the return fan, the air is mixed with outdoor air. Then the mixed air is drawn through filters, which are usually low or medium efficiency air filters. By passing through the cooling coils, the air stream is cooled down to some set point temperature. Often dehumidification occurs. Then, the air enters the heating section, where it is heated up to the set point temperature, if needed. As mentioned before, the heating is done by either electric heating, gas fired heating or a heat pump. After drawn through the heating section the supply air stream leaves the packaged unit and enters the ductwork of the building.

The cooling capacity is provided by a vapor compression cycle, with reciprocating or scroll compressors as the central component. The cooling coils are always direct expansion coils in rooftop packaged units. Units with a higher cooling capacity often have two separate refrigerant circuits in the evaporator for better capacity control. The condenser is designed of fin air-cooled condensing coils in which the refrigerant vapor is liquefied.

Figure 2.2 illustrates the arrangement of the main components of a rooftop packaged unit described above. Also, the air flow through the unit is shown schematically.

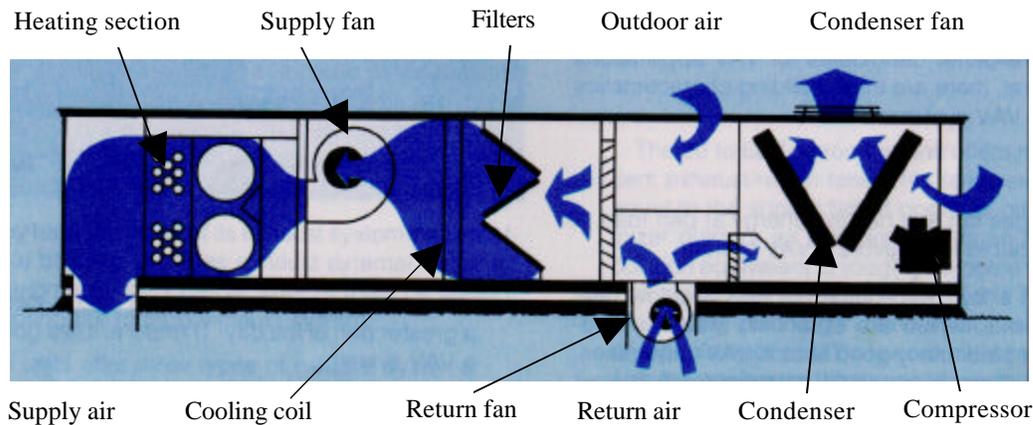


Figure 2.2: Components of a rooftop packaged unit and schematic air flow [Trane, 1981]

2.3 Available manufacturer data

Almost all large manufacturers of rooftop packaged air conditioning units have catalogs available for their products. To build a model that is generally applicable, catalog information from different manufacturers (Trane, Carrier, AAON) was collected and available inputs were determined.

In all catalogs the total cooling capacity as well as the sensible capacity is given as a function of volume flow rate, ambient temperature and return air conditions. Carrier also provides the compressor power draw for these operating points, which is a necessary input for the model. In addition, the compressor type and its performance in form of a few operating points or a compressor map has to be known. Knowing this information, the model developed in this project can be applied to any rooftop packaged unit. Table 2.1 presents the

data that is used as input for the rooftop packaged air conditioning model. Also, it shows the availability of those data for three different manufacturers.

Table 2.1: Available catalog information on rooftop packaged units from different manufacturers

Available Catalog Data	Manufacturer		
	Trane	Carrier	AAON
UNIT			
Entering Wet Bulb (evaporator)	X	X	X
Entering Dry Bulb (evaporator)	X	X	X
Ambient Temperature	X	X	X
Total Cooling Capacity	X	X	X
Sensible Cooling Capacity	X	X	X
Compressor Power Input		X	
COMPRESSOR			
Compressor Type	X	X	
Quantity	X	X	X
CONDENSER FAN			
Volume Flow Rate	X	X	X
Power Input	X	X	X
EVAPORATOR COIL			
Face Area	X	X	X
EVAPORATOR FAN			
Volume Flow Rate (evaporator)	X	X	X
Power Input	X	X	X

Chapter 3

Compressor Modeling

3.1 Compressors

The refrigeration compressor is the central component of a vapor compression cycle. It raises the pressure and provides the circulation of the refrigerant. In rooftop packaged units either reciprocating compressors or scroll compressors are used. Thus, a model for these both compressor types is necessary. A compressor model will be developed for reciprocating compressors and will then be applied to scroll compressors.

3.1.1. Reciprocating Compressors

Reciprocating compressors are positive displacement compressors. They are used for applications such as rooftop packaged units with a cooling capacity of up to about 210 kW (60 tons). Reciprocating compressors are classified as hermetic, semi-hermetic and open compressors. The number of cylinders ranges from one up to eight or more cylinders. Often more than one compressor is coupled together to achieve higher cooling capacities for large systems. If more than one compressor is used in rooftop packaged units, part load operation can be achieved by turning off one or more compressors. Often, compressors with more than one cylinder are also equipped with an unloading mechanism, which permits the unloading of one or more cylinders to achieve better part load performance. The unloading is done by leaving the suction valve of the unloaded cylinders open.

Figure 3.1 shows the three types of reciprocating compressors, (a) hermetic compressors, (b) semi-hermetic compressors, and (c) open compressors.

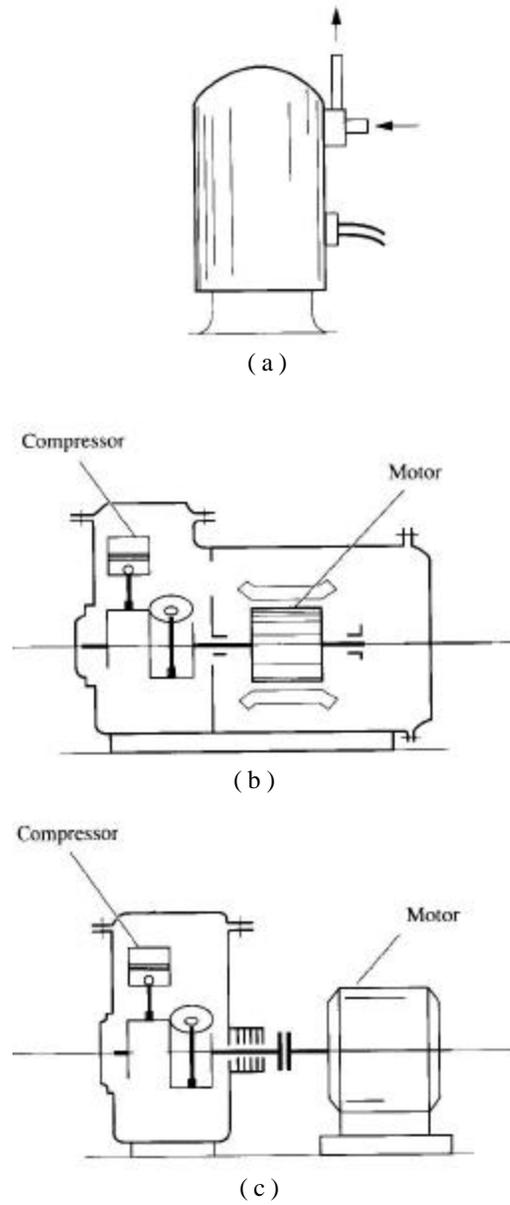


Figure 3.1: Classification of reciprocating compressors [Wang, 1993]

Most packaged units use semi hermetic compressors, which means that the motor and compressor unit are mounted in one housing. The main difference between a hermetic and a semi-hermetic compressor is that the semi-hermetic compressor can be taken apart to do maintenance work.

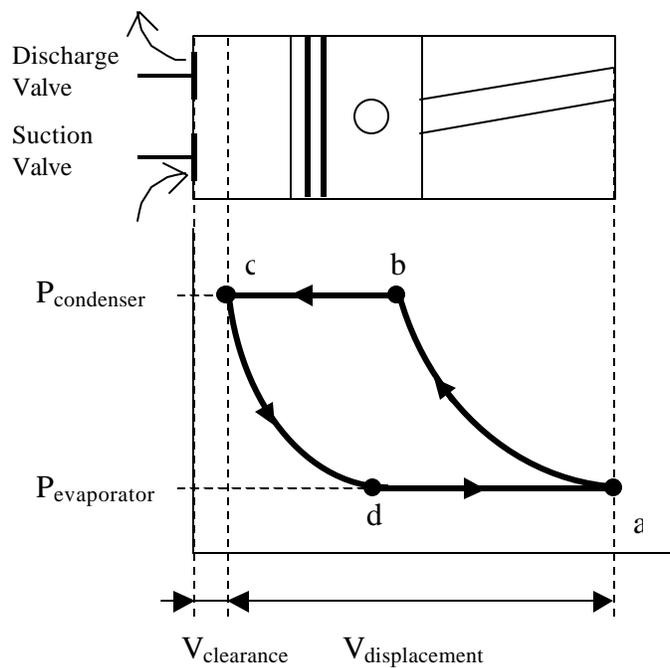


Figure 3.2: Schematic indicator diagram for a reciprocating compressor

Figure 3.2 illustrates the compressor cycle occurring in a reciprocating compressor. At point d the suction valve opens and vaporized refrigerant leaving the evaporator enters the cylinder. While the piston moves down to its bottom dead center (point a), the cylinder volume is filled with vapor. Reaching the bottom dead center the suction valve closes and the compression process starts. At point b the pressure reaches a sufficient level to open the

exhaust valve and refrigerant is discharged to the condenser. This process takes place till the piston reaches its top dead center (point c). The volume that is now left is called the clearance volume. The gas expands as the piston moves to point d. The volume from the bottom to the top dead center is called the displacement volume. These both variables, clearance volume and displacement volume, are important parameters in the compressor model used for modeling the system. The pressures shown in the schematic indicator diagram above are not the real condensing and evaporating pressures, since pressure drops occur through the suction and exhaust valves. These pressure drops, however, are relatively small compared to the dimension of the total pressure.

3.1.2. Scroll Compressors

Similar to reciprocating compressors, scroll compressors are also positive displacement compressors. They are used in rooftop packaged units with a cooling capacity in the range from 35 kW (10 tons) to 420 kW (120 tons), especially in those systems with a large cooling capacity. This is due to the fact that scroll compressors operate more efficiently than reciprocating compressors do. Also, they are more reliable, operate more quietly and smoothly, which makes them also increasingly popular for smaller units. In Figure 3.3 a scroll compressor is shown with its compression process.

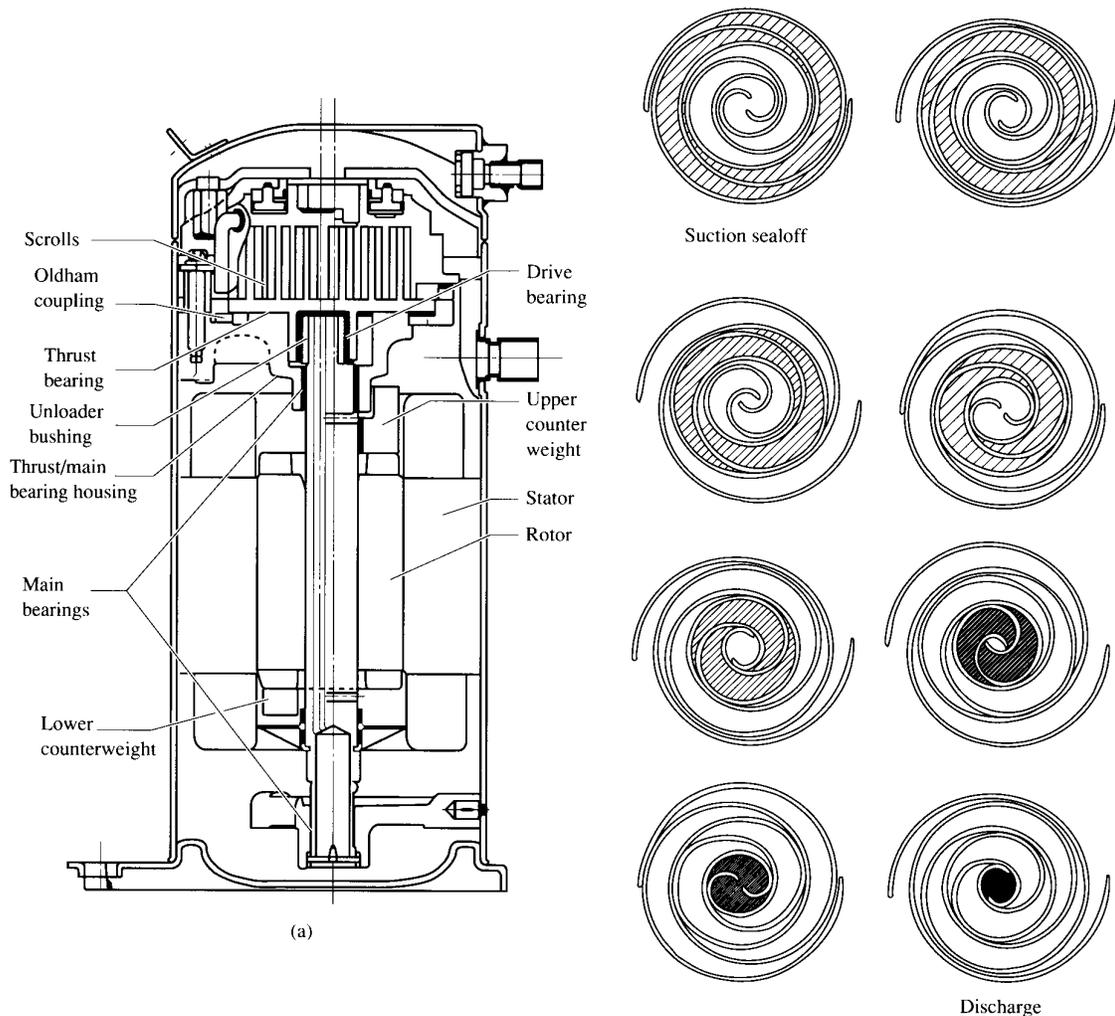


Figure 3.3: Scroll compressor with scroll compression process [Wang, 1993]

The scroll compression process takes place between two scrolls. One scroll is stationary and fixed. The other scroll is assembled with a phase difference of 180° and moves in an orbit around the center of the other scroll. The suction process takes place at the lateral openings. Vapor refrigerant enters the space between the two scrolls. When the orbiting scroll moves, the space between the two scrolls becomes sealed off and the compression process takes place. The space between the scrolls decreases continuously and by reaching

the center of the stationary scroll, the compressed hot refrigerant is discharged through a small opening in the center. Intake, compression, and discharge occur simultaneously in scroll compressors. Table 3.1 shows the characteristics of reciprocating and scroll compressors by comparison.

Table 3.1: Comparison of reciprocating and scroll compressors [Wang, 1993]

	Reciprocating compressors	Scroll compressors
Compression process characteristic	positive displacement	positive displacement
Refrigeration capacity	< 700 kW (200 tons)	up to 200 kW (60 tons)
Refrigerants	HCFC-22, HFC-134a	HCFC-22
Volumetric efficiency	0.92 to 0.68 (at $R_{com} = 1$ to 7)	> 0.95 (at $R_{com} = 4$)
Compressor efficiency	0.83 to 0.75 (at $R_{com} = 4$ to 7)	5 to 10 % higher
Capacity control	on-off, cylinder unloader	on-off, scroll unloader
Reliability	reliable	more reliable
Application to rooftop packaged units	up to 200 kW (60 tons)	35 kW (10 tons) up to 420 kW (120 tons)

3.2 Compressor Modeling

Many approaches for modeling refrigeration compressors have been developed. Modeling techniques for reciprocating and scroll compressors are of special interest for this project, since these are the types of compressors used in rooftop packaged units. Simple compressor models, which are found in thermodynamic textbooks such as Moran and

Shapiro (1992), often assume the compressor to be adiabatic. Inefficiencies are taken into account by defining an isentropic compressor efficiency. More detailed compressor models such as presented by Shapiro (1995) also take heat transfer into account by developing expressions for the heat loss of the compressor.

Another possibility to describe the performance of compressors is to use polynomial expressions for the mass flow rate and the compressor power draw. Stoecker (1971) proposed polynomial expressions for the compressor power draw and the capacity. Each of these expressions contains nine coefficients that have to be determined. Thus, an accurate performance prediction is possible if one has enough data points to determine these coefficients. However, extrapolation outside of the range of the data that were used for determining model parameters is uncertain with this method.

For this project a semi-empirical compressor model presented by Jähnig (1999) is used. This model allows certain extrapolation outside of the range of the fitted data, and is described in the following section.

3.2.1. Mass Flow Rate

For predicting the mass flow rate of reciprocating compressors, a model, known as the volumetric efficiency model presented by McQuiston and Parker (1988), is used. The volumetric efficiency is defined as the ratio of the actual mass flow rate of the refrigerant to the theoretical possible mass flow rate of the compressor. Theoretical possible mass flow rate

means that the displacement volume is fully filled with refrigerant vapor at the suction inlet state. Thus, using the notation of Figure 3.2, the theoretical mass flow rate can be written as:

$$\dot{m}_{r,th} = \frac{(V_a - V_c)}{v_{suc}} \cdot RPM \quad (3-1)$$

The actual mass flow rate is given by equation (3-2):

$$\dot{m}_{r,act} = \frac{(V_a - V_d)}{v_a} \cdot RPM \quad (3-2)$$

Consequently, the volumetric efficiency can be written as:

$$\mathbf{h}_v = \frac{\dot{m}_{r,th}}{\dot{m}_{r,act}} = \frac{(V_a - V_c) \cdot v_{suc}}{(V_a - V_d) \cdot v_a} \quad (3-3)$$

The compression and the expansion can be described by equations (3-4) and (3-5) as polytropic processes between two states:

$$P_{suc} \cdot V_a^n = P_{dis} \cdot V_b^n \quad (3-4)$$

$$P_{suc} \cdot V_c^n = P_{dis} \cdot V_d^n \quad (3-5)$$

The polytropic exponent n determines the process occurring between two states. For $n=0$ the process is an isobaric, for $n=1$ the process is isothermal. For the compressor model the compression process is assumed to be reversible and adiabatic. Thus, the exponent n can be calculated as the ratio of specific heat at constant pressure and the specific heat at constant volume.

$$n = \frac{C_{p,r}}{C_{v,r}} \quad (3-6)$$

Knowing the displacement volume of the compressor

$$V_{disp} = V_a - V_c \quad (3-7)$$

and defining the clearance factor of the compressor as

$$C = \frac{V_c}{V_a - V_c} \quad (3-8)$$

the definition of the volumetric efficiency can be written in terms of the clearance factor, the polytropic exponent, and the pressure ratio as well as the specific volume at inlet and discharge conditions:

$$h_v = \frac{\dot{m}_{r,act}}{\dot{m}_{r,th}} = \left(1 + C - C \cdot \left(\frac{P_{dis}}{P_{suc}} \right)^{\frac{1}{n}} \right) \cdot \frac{v_{suc}}{v_a} \cdot \frac{RPM}{RPM} \quad (3-9)$$

3.2.2. Compressor Power

The work per mass of refrigerant vapor required during compression process can be calculated by the following equation:

$$W_{th} = \int_a^b v dp - \int_d^c v dp \quad (3-10)$$

Performing the integration for a polytropic process, the work per compressor cycle per mass for polytropic processes is given by:

$$W_{th} = \left(\frac{n}{n-1} \right) \cdot P_{suc} \cdot v_{suc} \cdot \left[\left(\frac{P_{dis}}{P_{suc}} \right)^{\frac{n-1}{n}} - 1 \right] \quad (3-11)$$

Then the compressor power can be calculated by:

$$P_{ih} = \dot{m}_{r,act} \cdot \left(\frac{n}{n-1} \right) \cdot P_{suc} \cdot v_{suc} \cdot \left[\left(\frac{P_{dis}}{P_{suc}} \right)^{\frac{n-1}{n}} - 1 \right] \quad (3-12)$$

3.2.3. Parameter Identification

The expressions above for the volumetric efficiency and the work per compressor cycle can be used to determine the mass flow rate as well as the compressor power draw. However, they do not take inefficiencies into account such as friction losses, heat loss, or pressure drops. Consequently, using these equations for modeling the compressor performance is not applicable to a system. By introducing parameters that account for inefficiencies, the model can describe the compressor performance accurately. Thus, the compressor model can be used for system modeling. Jähnig (1999) proposed to introduce a pressure drop parameter and parameters for the compressor efficiency. The following parameters are used to model the compressor performance. The values were determined by fitting the model to catalog data.

- C - clearance factor
- Δp - pressure drop parameter for suction pressure
- a - parameter for compressor efficiency
- b - parameter for compressor efficiency

The expression for the mass flow rate is written in terms of the clearance factor. The clearance factor is a function of the geometry only and is therefore a fixed number for each compressor. The clearance volume, however, is usually not provided by the manufacturer and was determined by fitting compressor performance data to the compressor model.

The pressure drop parameter accounts for pressure drops across the valves. Pressure drops can be found on the suction line as well as on the discharge line. The following expression was found to work best. Although it accounts only for the pressure drop on the suction line, it provides a good fit to the catalog data.

$$p_{suc} = p_{evap} \cdot (1 - \Delta p) \quad (3-13)$$

Jähnig (1999) defined a combined efficiency as the ratio of the theoretical required compressor power draw to the actual power draw. It accounts for inefficiencies such as motor inefficiencies, mechanical inefficiencies, and heat transfer in the compressor. The combined efficiency is defined by the following equation:

$$\mathbf{h}_{comb} = \frac{P_{th}}{P_{act}} = a + \frac{b}{p_{suc}} \quad (3-14)$$

where a and b need to be determined by fitting to catalog data.

3.3 Validation of the Compressor Model

Compressor data were collected from different manufacturers. The compressor model was validated by using the given manufacturer's catalog data. Having developed the model

originally for reciprocating compressors, the capability of predicting the performance of scroll compressors was also tested.

3.3.1. Objective Function for Parameter Fitting Procedure

The parameter values are obtained by fitting the compressor model to manufacturer's catalog data. The objective function is a mean weighted RMS error, which measures the mean percent deviation of the calculated data points from the mean value of all catalog data points used for fitting.

Thus, the objective functions for determining the parameters used to predict the mass flow rate and the power are given by equation (3-15) and (3-16).

For mass flow rate:

$$OF_m = \sqrt{\frac{\sum_{i=1}^N \left(\frac{m_{r,cat} - m_{r,calc}}{m_{r,mean}} \right)^2}{N}} \quad (3-15)$$

For compressor power:

$$OF_p = \sqrt{\frac{\sum_{i=1}^N \left(\frac{P_{cat} - P_{calc}}{P_{mean}} \right)^2}{N}} \quad (3-16)$$

First the parameters for the mass flow rate were determined in the fitting process. Having determined these parameters, they were used in the fitting process to determine the parameters used in the model for the power prediction. The four parameters could be fitted at

the same time. Parameter values for the mass flow rate, however, also affect the power prediction. Parameter values for the mass flow rate could be found, which would yield to a better power fit, while the mass flow rate fit would be worse if all parameters were fitted at the same time.

3.3.2. Model Validation for Reciprocating Compressors

The collected compressor data for reciprocating compressors covers a capacity range from 3.5 kW (1 ton) up to 53 kW (15 tons). Table 3.2 lists some of the compressors modeled in this project to validate the compressor model.

Table 3.2: Compressor data for five different reciprocating compressors

Compressor	(1)	(2)	(3)	(4)	(5)
Manufacturer	Trane	Trane	Trane	Carlyle	Carlyle
Model No.	16 BH 143 YZ 1	16 GH 673 LM 1 G	14 CRHC 100	06 DE 8241	06 DE 5371
Type	hermetic	hermetic	hermetic	semi-hermetic	semi-hermetic
Capacity kW (tons)	4 (1)	18 (5)	35 (10)	33 (9)	52 (15)
Displacement [ft ³ /min]	3.148	12.534	25.5	23.9	37.1
Motor speed [min ⁻¹]	3500	3500	3450	1750	1750
Cylinder	1	2	3	6	6

For determining the compressor parameters C , Δp , a , and b compressor data were used in which the compressor power draw and capacity were given as a function of the evaporating temperature and the condensing temperature. Available compressor data refer to rating conditions according to ARI Standard 520. Thus, compressor performance data refer to $-9.4\text{ }^{\circ}\text{C}$ ($15\text{ }^{\circ}\text{F}$) subcooling and $-6.7\text{ }^{\circ}\text{C}$ ($20\text{ }^{\circ}\text{F}$) superheat. Data of compressors (4) and (5) refer to $-9.4\text{ }^{\circ}\text{C}$ subcooling and superheat.

Table 3.3 lists the calculated model parameters for compressors (1) to (5). Fourteen data points covering the catalog range of evaporating and condensing temperature were chosen for calculating the model parameters. For these compressors, the clearance factor, which is used as a curve fit parameter, is in the range of 0.071 to 0.121. Realistic values, which are difficult to obtain from manufacturers, are probably around 0.05. Manufacturers try to minimize the clearance volume as far as possible. This is due to the fact that the volumetric efficiency increases with decreasing clearance volume, which means consequently an enhanced compressor performance.

Table 3.3: Compressor parameters for compressors (1) to (5)

Compressor	(1)	(2)	(3)	(4)	(5)
Number of data points	16	16	14	14	14
C	0.121	0.084	0.097	0.076	0.071
Δp	0.075	0.050	0.068	0.091	0.094
a	0.735	0.727	0.682	0.743	0.716
b	-0.482	-0.257	-0.104	-0.261	-0.191
OF _m [%]	6.4	2.0	0.9	1.4	2.0
OF _P [%]	10.6	3.1	4.3	2.0	2.9

3.3.2.1 Required Number of Data Points for Fitting

From Table 3.2, compressor (5) was chosen to analyze the effect of the number of data points used in the fitting process on the compressor performance. Table 3.4 contains the data sets (a, b, c and d) used for determining the compressor parameter for four different numbers of given operating points. Compressor parameters were determined using five, eight and fourteen data points. Table 3.5 lists the calculated compressor parameters for data sets a to d. As a general rule data points should be selected as a combination of low and high values for each cataloged operating variable. In the case of the compressor, the operating variables are the condensing and evaporating temperatures. Thus, at least four data points should be selected for the fitting process to get a good curve fit. By using additional data points in the fitting process, model prediction can be improved.

Table 3.4: Data sets a to d used for determining compressor parameters for compressor (5)

Data point	$T_{\text{evaporator}}$ [°C]	$T_{\text{condenser}}$ [°C]	Mass flow [kg/hr]	Power [kW]	Set a	Set b	Set c	Set d
1	-23.3	32.2	367.4	8.5	X	X	X	
2	-23.3	40.6	328.9	8.8		X	X	
3	-23.3	48.9	277.2	8.9	X	X	X	
4	-12.2	37.8	617.6	10.9		X	X	X
5	-12.2	60.0	505.2	12.5		X	X	X
6	-6.7	32.2	811	11.1			X	X
7	-6.7	48.9	721.4	13.5			X	X
8	-6.7	62.8	647.9	14.8			X	X
9	-1.1	37.8	978.1	12.9	X	X	X	
10	-1.1	60.0	873.4	16	X	X	X	
11	4.4	32.2	1211	12.6			X	
12	4.4	48.9	1123	16.1			X	
13	4.4	62.8	1017	18.3			X	
14	10.0	48.9	1347	17.1	X	X	X	

Table 3.5: Compressor parameters for data sets a to d

Parameters	Set a	Set b	Set c	Set d
Clearance factor C	0.0701	0.0702	0.0706	0.0694
Pressure drop coefficient Δp	0.1034	0.0948	0.09419	0.0912
Compressor efficiency parameter a	0.7046	0.7217	0.7160	0.7003
Compressor efficiency parameter b	-0.1778	-0.2044	-0.1905	-0.0966
Objective function mass flow rate [%]	2.1	2.7	2.0	0.5
Objective function power [%]	2.0	2.4	2.9	1.0
Objective function mass flow rate (14) [%]	2.4	2.0	2.0	2.2
Objective function power (14) [%]	3.0	3.0	2.9	3.2

The objective function in Table 3.5 shows the normalized RMS error as a result of the parameter fitting procedure. The last two rows (objective function (14)) of Table 3.5 contain the RMS errors as a result of predicting the compressor performance for all 14 data points. These values are all below 3.2 %. Using only five data points (set a) over a wide range of operating conditions already works well for predicting the compressor performance and results with a normalized RMS error of 3.0 %. The results of using set b (8 data points) or set c (14 data points) are similar. The compressor parameter as well as the resulting errors in predicting the compressor performance are almost the same. Consequently, for determining the parameters of the compressor model it is sufficient to use five data points with high and low values of evaporating and condensing temperatures, though model accuracy can be improved by using more data points. Figure 3.4 illustrates the predicted compressor performance for the different sets of data points.

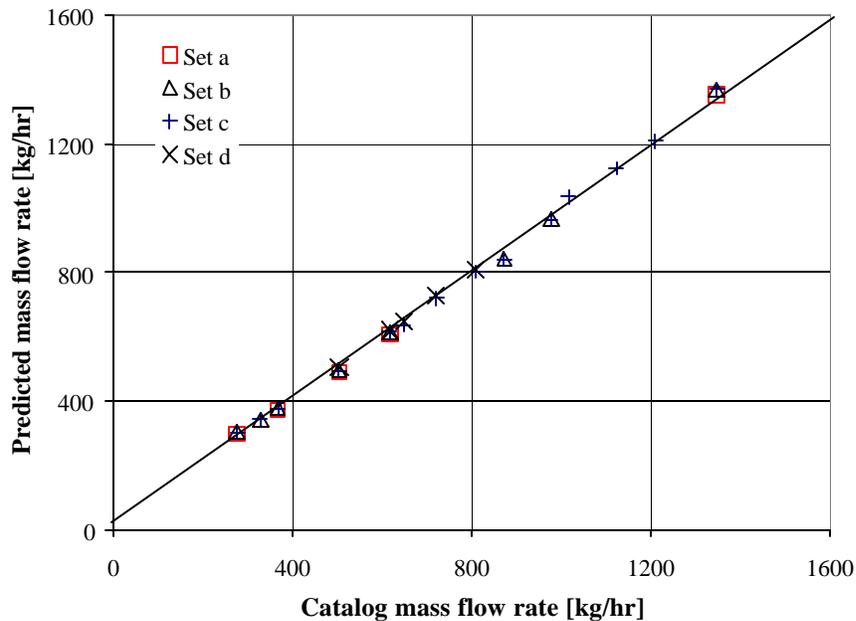


Figure 3.4: Predicted mass flow rate for data sets a to d

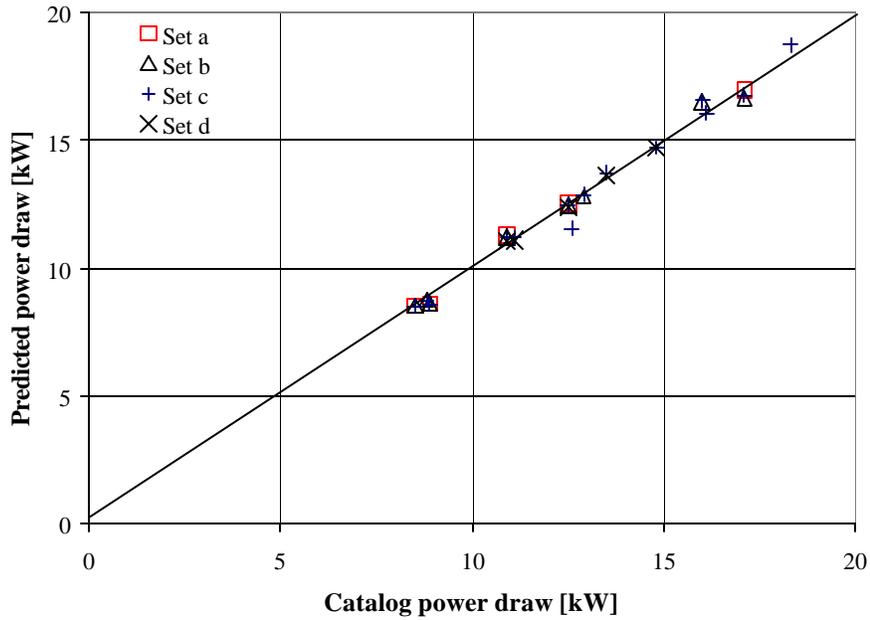


Figure 3.5: Predicted power draw for data sets a to d

3.3.2.2 Extrapolation Ability of the Compressor Model

The extrapolation ability of the compressor model was tested with compressor (5) and data set d. Compressor data and calculated parameters are given in Table 3.2 to Table 3.5. Data points used for determining the compressor parameter were picked in a limited range of evaporating temperatures to test the extrapolation ability of the compressor model. After having determined the compressor parameters by using these five data points (data set d), the mass flow rate prediction as well as the prediction of the compressor power draw was tested for the whole operating range given in the catalog. Figure 3.6 shows that the predicted mass flow rate is in very good agreement with the catalog mass flow rate over the whole operating range. The figure also depicts the extrapolation ability for the mass flow rate

model. The normalized RMS error is 2.6 % for the mass flow rate and 3.9 % for the compressor power draw:

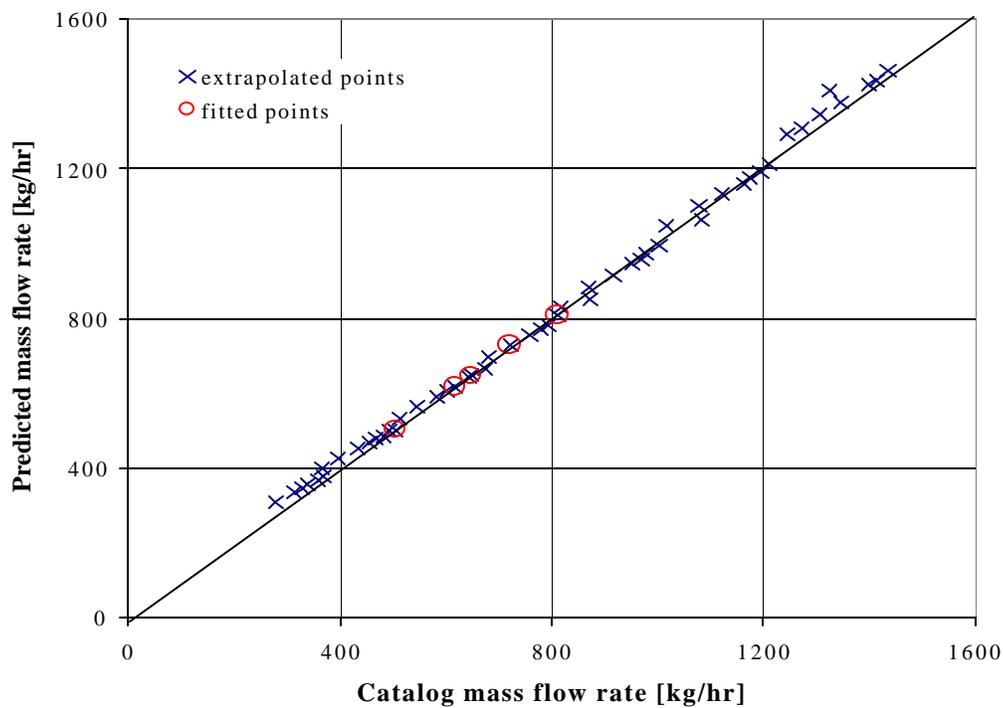


Figure 3.6: Predicted mass flow rate with extrapolated data points using data set d

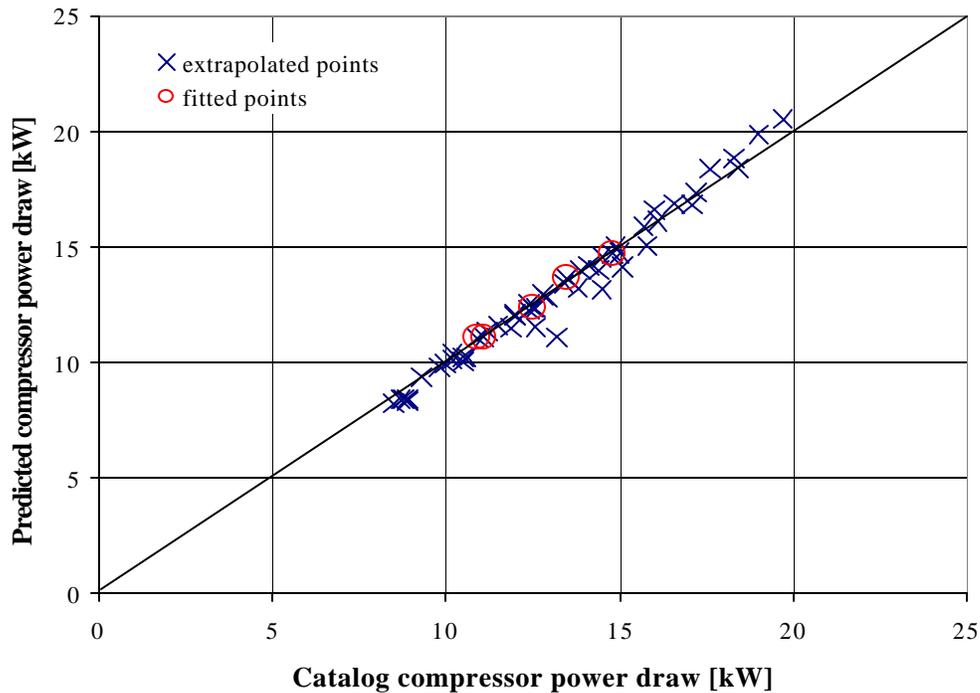


Figure 3.7: Predicted compressor power draw with extrapolated data points

The extrapolation ability of the compressor power model is not as accurate as that for the mass flow rate model. However, the evaporating temperatures used for determining the parameters of the compressor model were in a very narrow range of -12.2 to -6.7 °C, while the catalog data range is from -23.3 to 10 °C. Extrapolation to data points with high suction temperatures and low suction discharge pressure ratio results in a considerable underestimation of the compressor power draw, which can be seen in Figure 3.7 and Figure 3.9.

The following two graphs show the predicted mass flow rate and the compressor power draw as a function of the evaporator and condensing temperatures.

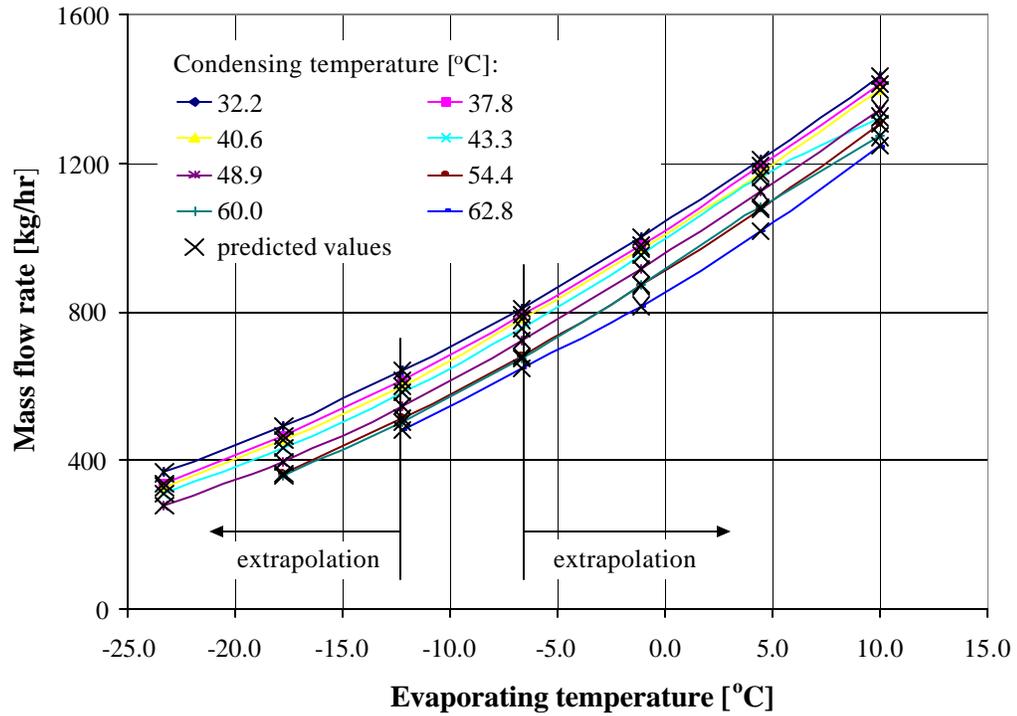


Figure 3.8: Compressor mass flow rate map with catalog and predicted mass flow rate

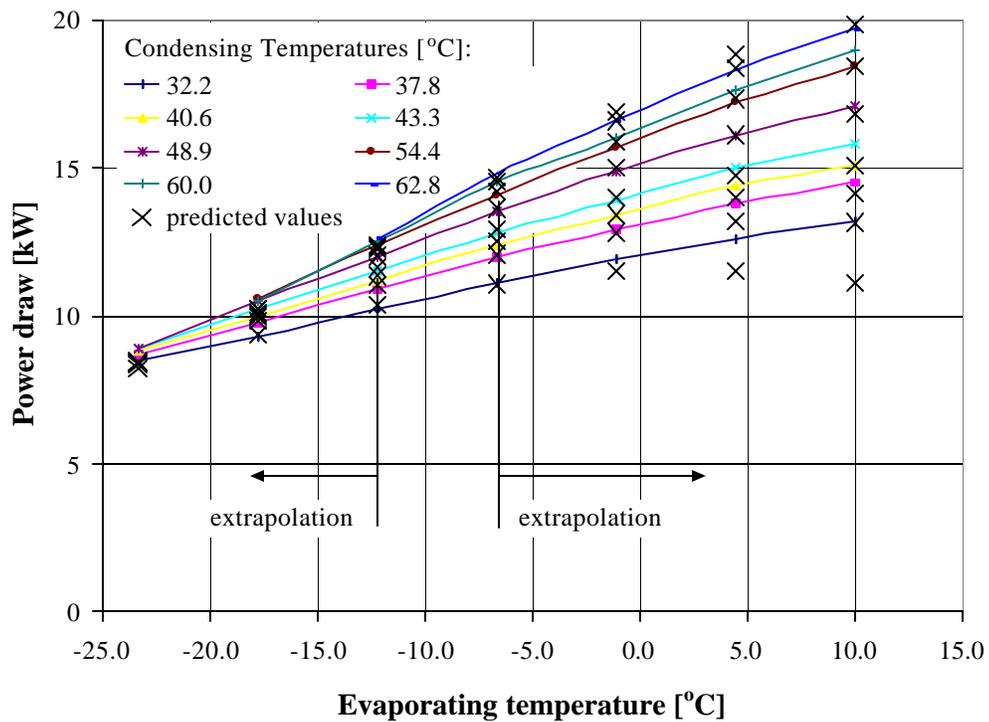


Figure 3.9: Compressor power draw map with catalog and predicted compressor power draw

3.3.3. Model Validation for Scroll Compressors

Scroll compressors are often used in rooftop packaged units with high capacities. Although, the compressor model was developed originally for reciprocating compressors, the model may also be used for scroll compressors.

Table 3.6 lists the scroll compressors that were used for validation of the compressor model. The capacities range from 35 kW (10 tons) to 70 kW (20 tons). Table 3.7 shows, that the compressor model can also be used to predict the compressor mass flow rate for scroll compressors. The error for the mass flow rate prediction is lower than 2 %. The predicted compressor power draw, however, is off by about 8 %. Figure 3.10 and Figure 3.11 show the performance prediction for compressor (8) obtained with the compressor model originally derived for reciprocating compressors.

Table 3.6: Scroll compressor data

Compressor	(6)	(7)	(8)
Manufacturer	Trane	Trane	Trane
Model No.	CSHA 100	CSHA 150	CSHA 200
Type	hermetic	hermetic	hermetic
Capacity [kW] ([tons])	35 (10)	53 (15)	70 (20)
Displacement [ft ³ /min]	20.60	30.89	41.20
Motor speed [1/min]	3500	3500	3500

Table 3.7: Scroll compressor parameters

Compressor	(6)	(7)	(8)
Number of data points	14	14	14
Clearance factor C	0.028	0.033	0.034
Pressure drop coefficient Δp	0.000	0.000	0.000
Compressor efficiency parameter a	0.740	0.741	0.737
Compressor efficiency parameter b	-0.376	-0.411	-0.408
Objective function mass flow rate [%]	1.6	1.5	0.9
Objective function power [%]	7.5	7.1	7.6

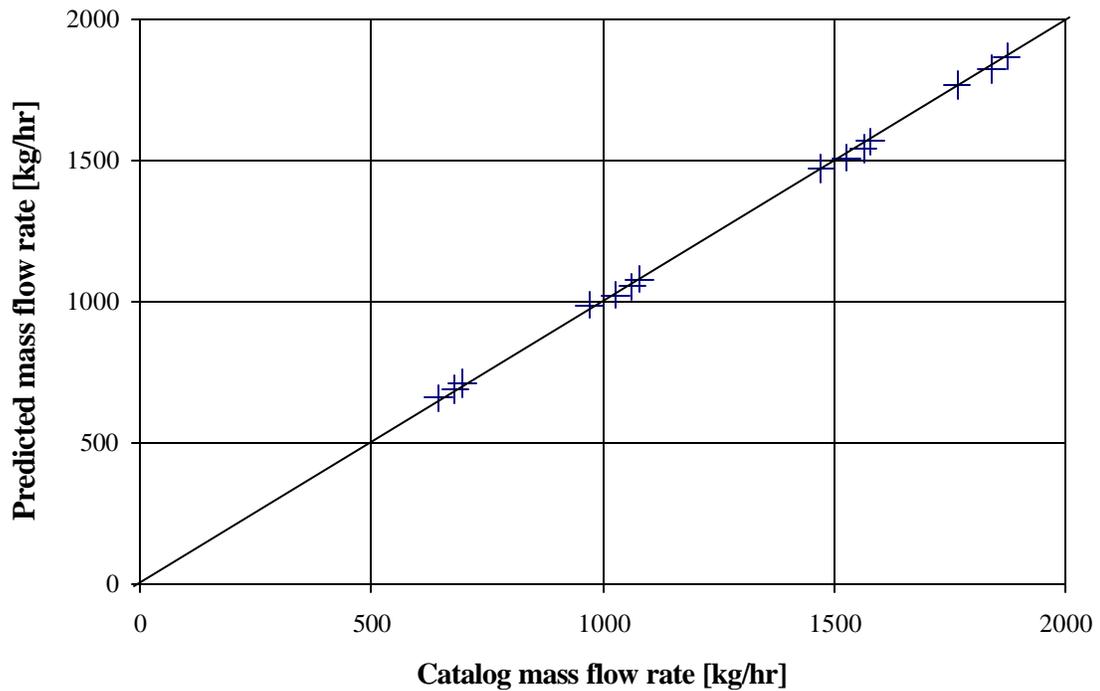


Figure 3.10: Predicted mass flow rate for scroll compressor (8)

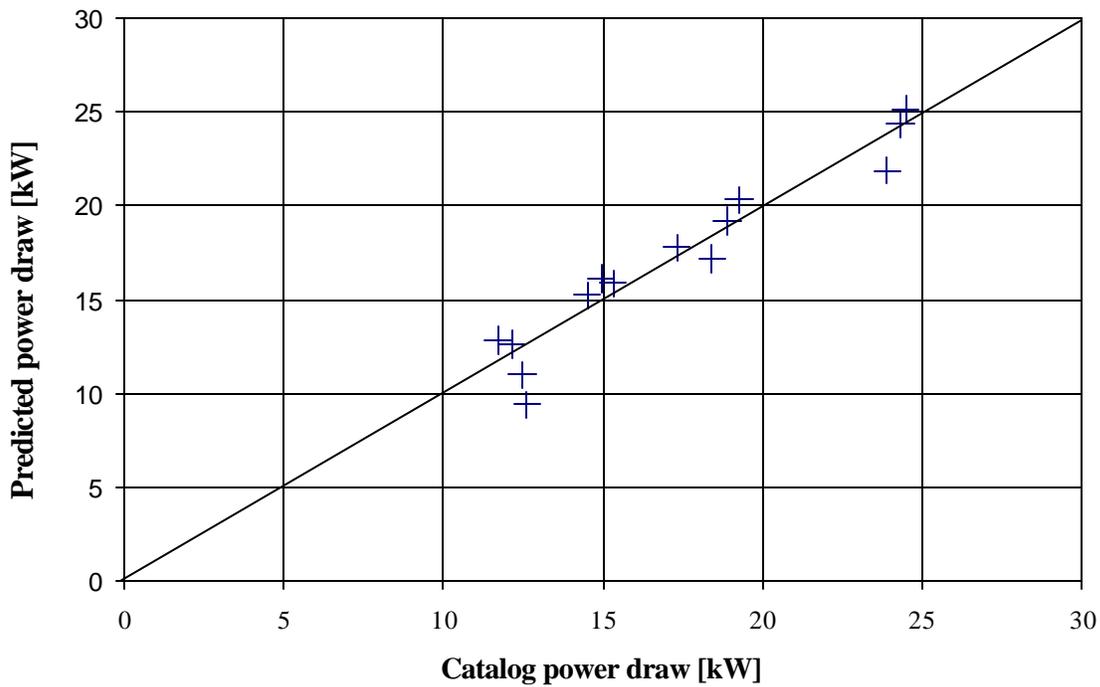


Figure 3.11: Predicted compressor power draw for compressor (8)

For very low discharge suction pressure ratios, compressor power draw prediction results in a considerable underestimation of the compressor power draw. Therefore, a different model was developed to predict more accurately the power draw of scroll compressors, which is described below.

For different scroll compressors the dependence of the power draw and the discharge suction pressure ratio was examined. For all compressors a strong linear dependence was found between power draw and discharge suction pressure ratio for a fixed evaporating temperature. The lines of constant evaporating temperature, however, have different slopes. This fact is shown for compressor (8) in Figure 3.12.

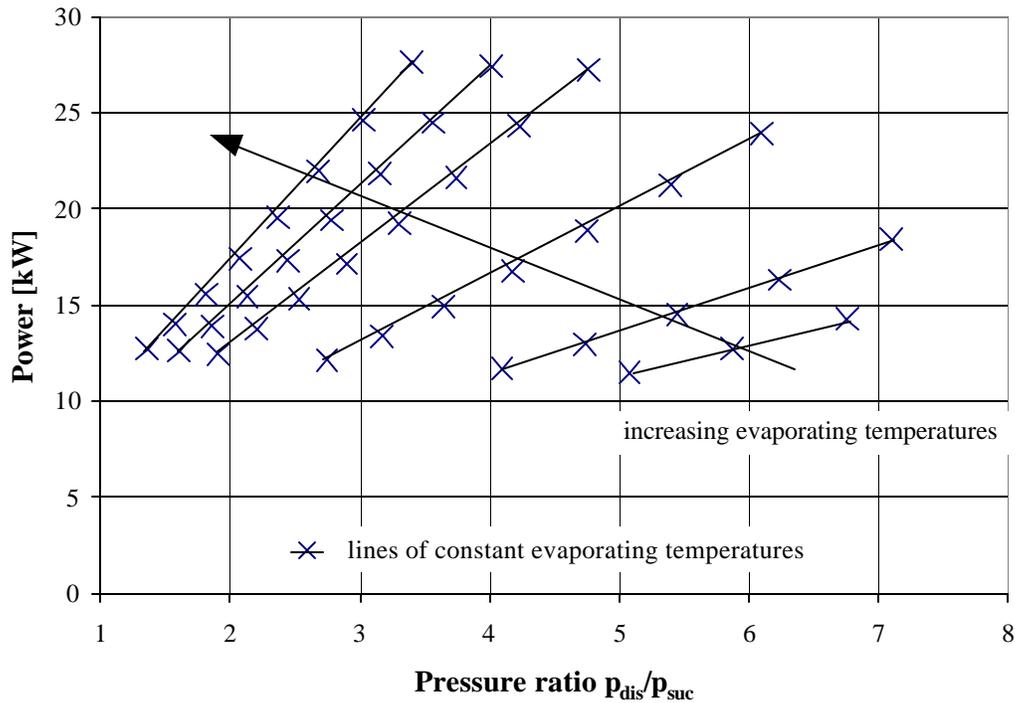


Figure 3.12: Power versus pressure ratio for compressor (8)

In Figure 3.12 the catalog compressor power is shown as a function of the pressure ratio. Figure 3.12 illustrates that the compressor power draw increases with the growing pressure ratio. The compressor power draw also increases with higher evaporating temperatures. This is due to the fact that the mass flow rate through the compressor increases with higher evaporating temperatures or pressures due to a higher density of the refrigerant at the suction inlet of the compressor. In Figure 3.13 the catalog mass flow rate is shown versus pressure ratio for different evaporating temperatures.

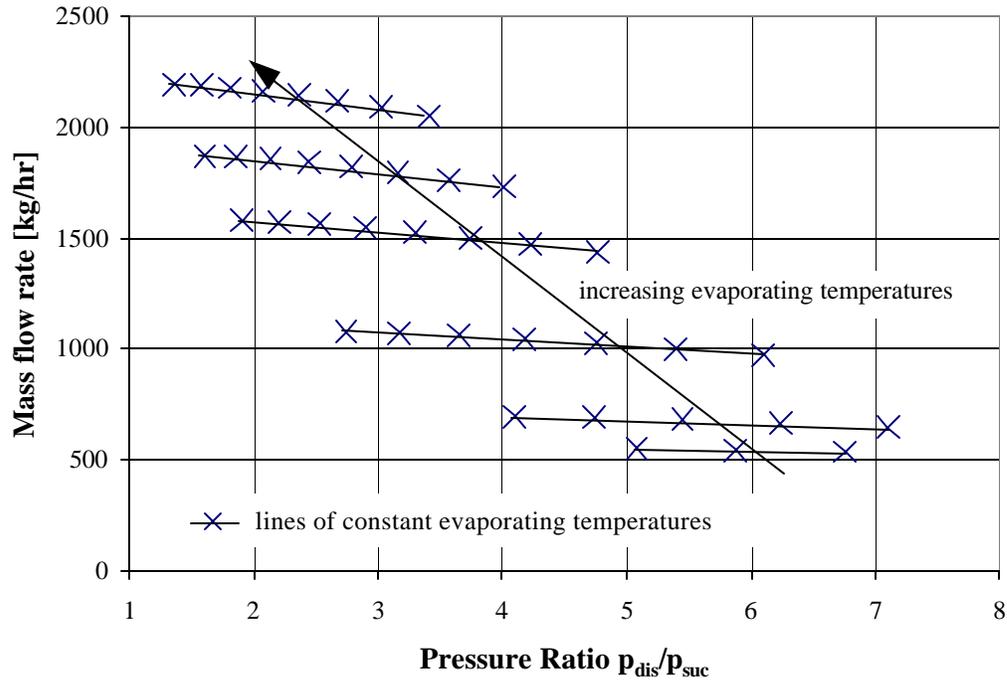


Figure 3.13: Mass flow rate versus pressure ratio for compressor (8)

Figure 3.13 shows that the mass flow rate versus pressure ratio is almost constant. This fact can be used to account for the different slopes in Figure 3.12 and used to develop an expression for the compressor power draw as the product of a proportionality constant times the product of the mass flow rate with the pressure ratio. The following expression for the power prediction for scroll compressors evolved from these circumstances:

$$P_{comp} = z \cdot \dot{m}_r \cdot \frac{P_{dis}}{P_{suc}} \quad (3-17)$$

where the parameter z is determined by fitting this expression to catalog compressor data.

This expression was tested for a number of scroll compressors and was found to be able to predict the compressor power draw accurately. Table 3.8 shows the errors for the power draw prediction for the compressor (6) to (8) by using 14 data points for fitting.

Table 3.8: Resulted errors in power draw prediction for compressors (6) to (8)

	(6)	(7)	(8)
Manufacturer	Trane	Trane	Trane
Model No.	CSHA 100	CSHA 150	CSHA 200
Capacity [kW] (tons)	35 (10)	53 (15)	70 (20)
Number of data points	14	14	14
Objective function power [%]	1.8	1.4	1.7

Compressor (8) was chosen as a reference compressor to show the effect of the number of data points used for fitting. Table 3.9 lists the results for this compressor. It shows, that even with few fitting points the curve fit parameter z can be obtained, allowing an accurate power draw prediction.

Table 3.9: Effect of number of fitting points on curve fit parameters and power draw prediction

Data set	(e)	(f)	(g)
Number of data points	5	8	14
Curve fit parameter z [kWh/kg]	0.003855	0.003898	0.003933
Objective function power [%]	0.7	0.9	2.2
Objective function power (14) [%]	2.3	2.0	2.0

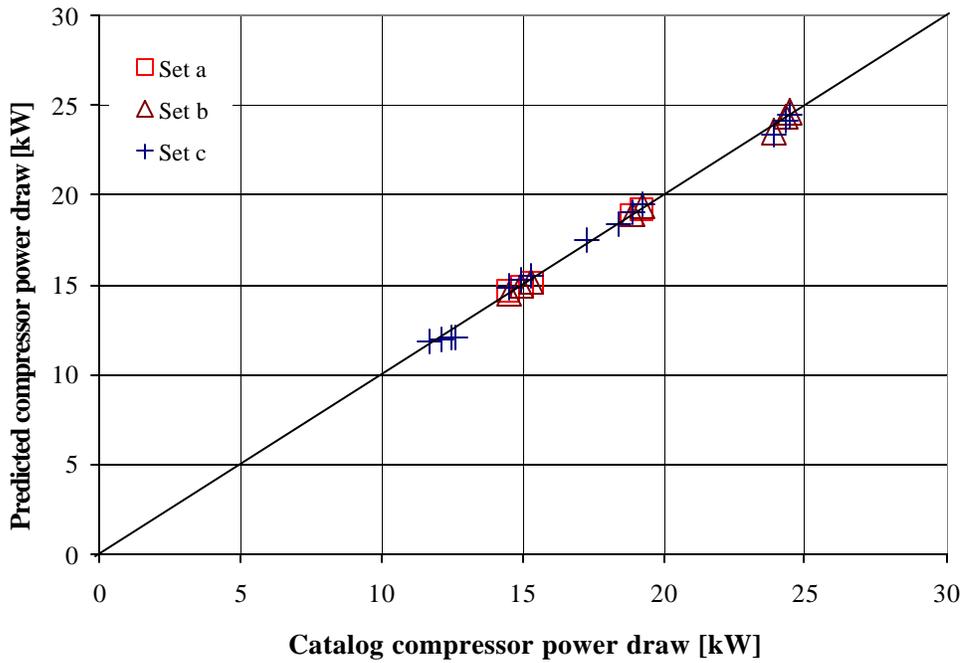


Figure 3.14: Predicted power draw with modified scroll compressor model

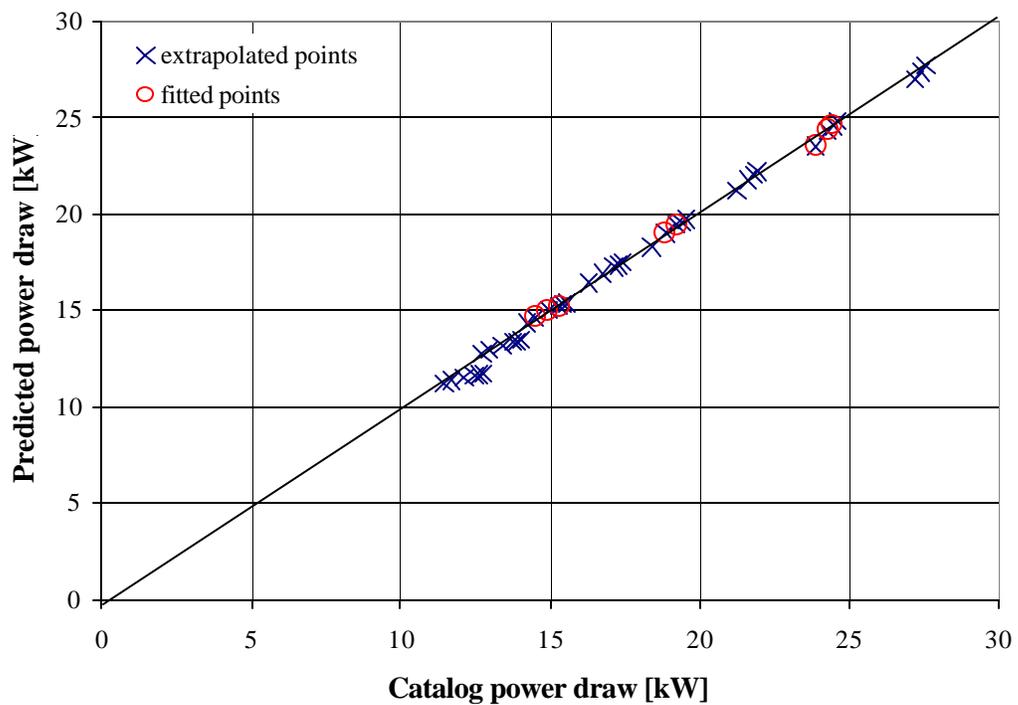


Figure 3.15: Predicted compressor power draw with extrapolated data points for data set b

The expression for the power draw of scroll compressors was validated with catalog compressor performance data. However, this expression was not tested in combination with the model for rooftop packaged air conditioning units due to the lack of performance data for packaged units with scroll compressors.

3.4 Conclusions

The performance of a vapor compression cycle depends strongly on the performance of the compressor. Thus, for modeling a vapor compression cycle, a model is required that accurately describes the performance of the compressor.

For reciprocating compressors an existing model was chosen for performance prediction. This model, which is based on the volumetric efficiency model, allows good performance prediction over a wide range of operating conditions. For scroll compressors however, which are often used in rooftop packaged units, this compressor model does not predict the compressor power draw with sufficient accuracy. Thus, an alternate expression for the power draw of scroll compressors was developed.

Both models were evaluated with manufacturer's compressor data and found to represent satisfactorily the compressor performance. The error for compressor performance prediction, mass flow rate and power, can be expected to be generally less than 5 % using the compressor model presented in this chapter. Also, the extrapolation ability was tested and found to allow certain extrapolation outside the range of the fitted data. In addition, the

compressor model presented in this thesis requires fewer data points for determining model parameters than polynomial modeling techniques.

Chapter 4

Heat exchanger modeling

4.1 Introduction

Heat exchangers are used for a variety of engineering applications in air conditioning, space heating, power production, chemical processing and waste heat recovery. Thus, it is of great interest to predict the performance of heat exchangers. Many models have been developed for predicting the performance of different kinds of heat exchangers by using different modeling methods.

A simple way of modeling heat exchangers is to fit a polynomial expression to measured performance data. The form of these polynomial curve-fits depends on the type of heat exchanger. Stoecker (1971) proposed polynomials for a variety of heat exchangers such as condensers and cooling coils. Depending on the type of heat exchanger, the form of the polynomial expression and the number of curve fit parameters varies significantly. Using polynomial expressions often requires a large number of known operating points in order to predict accurately the performance. In addition, performance prediction outside the range of the data used for fitting is often uncertain using polynomial models.

Common methods to describe heat exchanger performance are the log-mean-temperature-difference method and the effectiveness-NTU method, which can be found in most heat transfer textbooks such as Incropera and de Witt (1998). These methods can take the design and the flow arrangement of the heat exchanger into account. To predict

performance, the overall heat transfer coefficient is needed, which can be calculated from fundamental heat and mass transfer relations, knowing the geometry of the heat exchanger.

Often, when predicting the performance with either the LMTD method or the effectiveness-NTU method, the overall heat transfer coefficient is assumed to be constant. This is done to allow a heat exchanger performance prediction with a low computational effort. However, assuming the overall heat transfer coefficient to be constant leads to considerable deviation in performance prediction if mass flow rates in the heat exchanger vary significantly. Thus, modeling approaches should be used that vary the overall heat transfer coefficient according to the operating conditions. In the HVAC1KIT (1995) a method for predicting heat exchanger performance assuming a constant overall heat transfer coefficient as well as a method that relates the overall heat transfer coefficient to the mass flow rates or capacitance rates are presented.

For modeling the heat exchanger in this thesis, a modeling technique was chosen that was presented by Rabehl (1997). In this method, the overall heat transfer coefficient is related to the mass flow rates or capacitance rates using the form of fundamental heat and mass transfer correlations. Knowledge of the heat exchanger geometry is required in order to calculate the overall heat transfer coefficient with the fundamental heat and mass transfer correlations. However, available catalog information about the geometry of a heat exchanger such as fin spacing, tube diameter or other variables, is often not detailed enough to calculate the overall heat transfer coefficient based on fundamental heat transfer correlations. Thus, Rabehl (1997) proposed to modify the fundamental heat transfer correlations and introduce heat exchanger specific parameters. These parameters are constants depending on heat

exchanger geometry and flow arrangement. They are determined by fitting them to catalog data. Thus, the overall heat transfer coefficient is determined and can be used in either the effectiveness-NTU or LMTD approach for the heat exchanger performance prediction. A very good prediction of performance is obtained with this method according to Rabehl's study (1997). Extrapolation of performance data can be performed, since this modeling technique uses generally valid fundamental heat and mass transfer relations.

4.2 Heat Exchangers for Refrigeration Applications

For modeling a vapor compression cycle, which is used in many refrigeration applications such as in rooftop packaged units, models for the evaporator and the condenser are required. Due to different heat transfer mechanisms in the condenser and the evaporator, two different models were used for performance prediction of these two heat exchangers.

Evaporators operate at the lower temperature level of the vapor compression cycle and pick up the refrigeration load. Generally, two kinds of evaporators are used in air conditioning systems. These two types are direct expansion coils and shell-and-tube water chillers.

In air conditioning systems that are equipped with air cooled direct expansion coils the supply air is directly cooled down by flowing across the finned tubes in which refrigerant evaporates. Two kinds of direct expansion coils exist. The flooded coil and the dry type. In the case of a flooded coil, the vertical tubes of the coil are filled with refrigerant up to a

certain level. Above this level the vaporized refrigerant accumulates and is delivered to the compressor. Dry type direct expansion coils are characterized by a flow of refrigerant through the horizontal tubes that evaporates continuously on its way through the coil.

Shell-and-tube water chiller have refrigerant and water as working fluid. Just as for the direct expansion coil, two different types exist. The refrigerant either flows through the tubes (direct expansion) or the shell is filled with refrigerant (flooded type) and water passes through the tubes that are flooded by the refrigerant.

Dry type direct expansion coils are used in rooftop packaged units. Figure 4.1 shows schematically a dry type direct expansion coil.

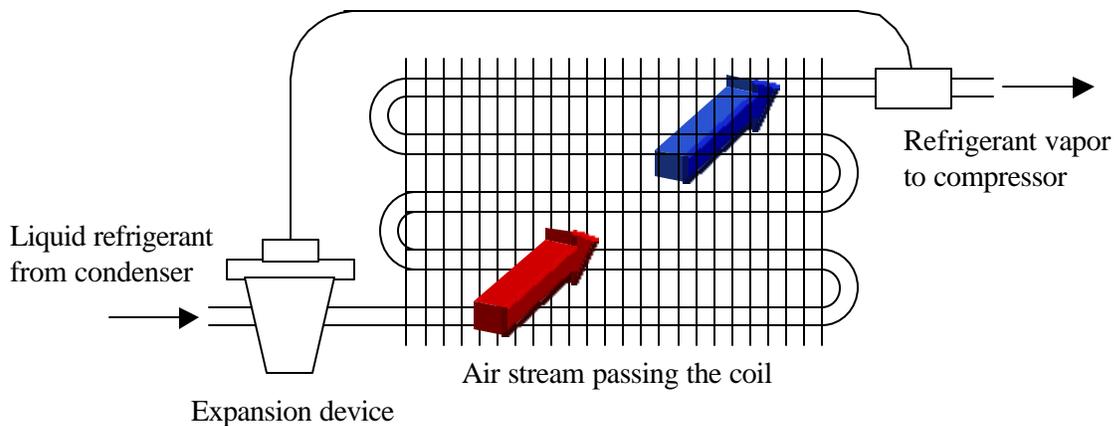


Figure 4.1: Schematic drawing of a dry type direct expansion coil

In these direct expansion coils the warm air flows across the fins and tubes of the direct expansion coil and is cooled down. Dependent on the coil surface temperature and the air entering conditions, dehumidification might occur. The refrigerant enters the cooling coil through an expansion device. This expansion device can be a capillary tube or a

thermostatic expansion valve. The latter is typically found in rooftop packaged units. The thermostatic expansion valve senses the temperature of the refrigerant as it leaves the evaporator coil and it meters just the correct amount of refrigerant to maintain the predetermined amount of superheat. In units with a low cooling capacity, the expansion valve is replaced by the capillary tube. The refrigerant enters the direct expansion coil as liquid. As it travels through the coil, picking up heat from the air flowing across the tubes, more and more refrigerant evaporates. Through the action of the expansion device, the refrigerant leaves the coil slightly superheated and enters the compressor.

Condensers operate at the higher temperature level and reject the heat gained in the vapor compression cycle, which consists of the refrigeration load and the compressor power, to the ambient. Condensers are either water-cooled condensers, evaporative cooled condensers or air cooled condensers. In rooftop packaged units air condensers are used with few exceptions. Air cooled condensers are used due to the fact that the air is readily available, first costs are lower, and maintenance costs are reduced.

The refrigerant is discharged from the compressor at high pressure and temperature, enters the condensing coil as vapor. As the vapor passes through the inside of the tubes, heat is rejected to the ambient, which is at a lower temperature level, and condensation of the refrigerant vapor occurs. Often, subcooling of the fully condensed refrigerant occurs in condensing coils. To increase the heat transfer rate between the air stream and the refrigerant vapor stream, finned copper tubes are used as condensing coils.

4.3 Heat Exchanger Modeling

The model for the evaporator and for the condenser model is based on the effectiveness-NTU method. However, different approaches are needed for these two heat exchangers due to different heat transfer mechanisms. For sensible heat exchangers such as air-cooled condensers an effectiveness-NTU method is used that is based on temperatures. For heat exchangers where latent heat transfer occurs, such as in cooling coils, an effectiveness-NTU method is used that is based on enthalpies. Both methods are described in the following sections.

4.3.1. Effectiveness-NTU Method for Dry Cooling Coils and Condensers

Condensers as well as evaporators for which water does not condense are modeled using an effectiveness-NTU method, which is usually more convenient for heat exchanger performance prediction than the log-mean-temperature-difference approach (LMTD). This is due to the fact that the LMTD method requires the fluid inlet and outlet temperatures, whereas the effectiveness-NTU method can be applied by just knowing the fluid inlet temperatures. Consequently, obtaining the numerical solution using the LMTD is often difficult. The effectiveness-NTU method as illustrated below is valid for sensible heat exchangers, only.

The maximum possible heat transfer rate is defined as

$$\dot{Q}_{\max} = C_{\min} \cdot (T_{h,i} - T_{c,i}) \quad (4-1)$$

where the minimum capacitance rate is defined as $C_{\min} = \dot{m}_a \cdot c_{p,a}$, since the capacitance rate of the air is always smaller than the capacitance rate of the refrigerant for a condensing or a direct expansion coil.

The effectiveness of the heat exchanger is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate.

$$\mathbf{e} = \frac{\dot{Q}_{act}}{\dot{Q}_{max}} \quad (4-2)$$

The capacitance rate ratio is defined as:

$$C_r = \frac{C_{\min}}{C_{\max}} \quad (4-3)$$

For evaporators and condensers the capacitance ratio is zero, since the specific heat is infinite for an evaporating or condensing fluid. The following expression for the heat exchanger effectiveness for a capacitance ratio of zero is used:

$$\mathbf{e} = 1 - e^{-NTU} \quad (4-4)$$

This expression for the heat exchanger effectiveness is generally valid for all flow configurations. In the equation above NTU stands for number of transfer units, which is a dimensionless parameter. The NTU is calculated by dividing the overall heat transfer area coefficient product UA by the minimum capacitance rate.

$$NTU = \frac{UA}{C_{\min}} \quad (4-5)$$

The UA product can be calculated by using fundamental heat and mass transfer relations. Finally, with the overall heat transfer area coefficient product determined the heat exchanger performance can be predicted.

4.3.2. Effectiveness-NTU Method for Wet Cooling Coils

Describing the performance of cooling coils is of great interest, since, in all air conditioning systems, cooling coils are used to cool a warm air stream. If the surface temperature of the cooling coil is lower than the dew point temperature of the entering air stream condensation occurs. In this case, the effectiveness-NTU method for sensible heat exchangers that is based on temperature cannot predict the performance accurately, since the latent heat transfer due to condensation is not taken into account.

Different approaches exist to predict the performance of cooling coils. Empirical models, such as proposed by Stoecker (1971), as well as fundamental models exist. Stoecker (1971) proposed equations (4-6) to (4-8) for describing the performance of direct expansion cooling coils where c_1 , c_2 and c_3 represent curve fit parameters.

$$\dot{Q}_{tot} = \dot{m}_a \cdot E \cdot (h_{a,i} - h_{a,sat}) \quad (4-6)$$

$$\dot{Q}_{sens} = \dot{m}_a \cdot E \cdot c_{p,a} \cdot (T_{db,i} - T_{r,i}) \quad (4-7)$$

where

$$E = c_1 + c_2 \cdot \dot{m}_a + c_3 \cdot \dot{m}_a^2 \quad (4-8)$$

The curve fit parameters c_1 , c_2 and c_3 are determined by fitting them to catalog data.

In the HVAC1KIT (1995), performance of direct expansion coils is predicted by replacing the air stream by a fictitious water stream. Two models are used. One model describes the coil performance by treating the coil as a heat exchanger where no condensation occurs. The other model describes the performance of the wet coil where condensation occurs. Both models use effectiveness-NTU relationships. The model that predicts the higher heat transfer is chosen for actual performance prediction. For the dry coil, the air stream is replaced by a water stream, which is calculated by the equation below.

$$\dot{m}_w = \frac{c_{p,a}}{c_{p,w}} \cdot \dot{m}_a \quad (4-9)$$

The fictitious water mass flow rate for the wet coil is given by equation (4-10) and equation (4-11) .

$$\dot{m}_w = \frac{\bar{c}_{p,fictitious}}{c_{p,w}} \cdot \dot{m}_a \quad (4-10)$$

$$\bar{c}_{p,fictitious} = \frac{h_{a,i} - h_{a,o}}{T_{wb,i} - T_{wb,o}} \quad (4-11)$$

Among the fundamental modeling approaches for cooling coils, a mechanistic method proposed by Braun et al. (1989) exists, which uses the formulation of the effectiveness-NTU method and applies it to cooling coils. The method is very accurate in performance prediction for cooling coils. In addition, the computational requirements are significantly less compared to methods where differential equations have to be solved in order to predict the cooling coil performance. This method will be referred to as the analogy approach, and is outlined below as it is used for describing the direct-expansion coil performance. A more detailed description of the method can be found in the work of Braun et

al. (1989) and Mitchell (1997). Figure 4.2 shows schematically the air flow and refrigerant flow for a cooling coil.

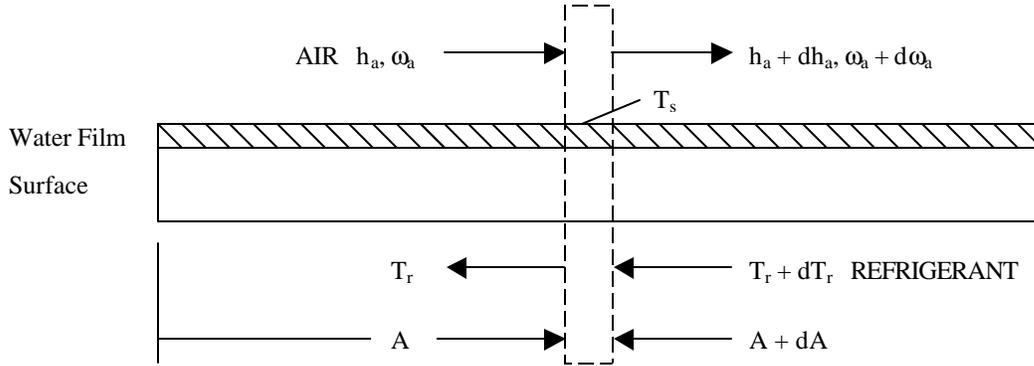


Figure 4.2: Schematic of a counterflow cooling coil

Based on Figure 4.2, differential equations can be formulated for the heat-and mass transfer. Assuming the Lewis number to be unity equations (4-12) and (4-13) are obtained.

$$\frac{d\mathbf{w}_a}{dA} = -\frac{NTU_o}{A_o} \cdot (\mathbf{w}_a - \mathbf{w}_{a,sat,s}) \quad (4-12)$$

$$\frac{dh_a}{dA} = -\frac{NTU_o}{A_o} \cdot [(h_a - h_{a,sat,s})] \quad (4-13)$$

$$\frac{dT_r}{dA} = NTU_i \cdot (T_r - T_s) \quad (4-14)$$

where

$$NTU_i = \frac{UA_i}{\dot{m}_r \cdot c_{p,r}} \quad (4-15)$$

$$NTU_o = \frac{hA_o}{\dot{m}_a \cdot c_{p,a}} \quad (4-16)$$

Equation (4-13) is written in terms of enthalpies, whereas equation (4-14) is written in terms of temperatures. By introducing an air saturation specific heat c_s (equation (4-18)), which is defined as the derivative of the saturated air enthalpy with respect to the temperature evaluated at the refrigerant temperature, equation (4-14) can be written in terms of saturated air enthalpy. Thus, equation (4-17) is obtained.

$$\frac{dh_{a,sat,r}}{dA} = \frac{\dot{m}_a \cdot c_s \cdot (dh_a / dA)}{\dot{m}_r \cdot c_{p,r}} \quad (4-17)$$

where

$$c_s = \left(\frac{dh_{a,sat}}{dT_r} \right)_{T=T_r} \quad (4-18)$$

In equation (4-19) the air saturation specific heat is determined as the ratio of the saturated air enthalpy difference to the temperature difference between the entering dew point and the refrigerant temperature. This is a good approximation for the air saturation specific heat used for the wet coil in the evaporator model.

$$c_s = \frac{h_{a,sat} \big|_{T=T_{dp}} - h_{a,sat} \big|_{T=T_{r,i}}}{T_{dp} - T_{r,i}} \quad (4-19)$$

In contrast to the evaporator model presented in the HVAC1KIT (1995) where the air stream is replaced with a water stream, the analogy approach for wet cooling coils replaces the refrigerant stream with a saturated air stream.

For the differential equations on the refrigerant side and the air side solutions can be found that are similar to the effectiveness-NTU relationships. Table 4.1 shows the analogy

between the characteristic variables of the effectiveness-NTU method for sensible heat exchangers and for heat exchangers with condensation.

Table 4.1: Effectiveness-NTU relations for sensible heat exchangers and wet cooling coils

Parameter	Effectiveness-NTU Method for Sensible Heat Exchangers	Effectiveness-NTU Method for Wet Cooling Coils
Capacitance ratio	$C_r = \frac{C_{\min}}{C_{\max}}$	$m^* = \frac{\dot{m}_a}{\dot{m}_r \cdot (c_{p,r}/c_s)}$
Number of transfer units	$NTU = \frac{UA}{C_{\min}}$	$NTU^* = \frac{U^* A}{\dot{m}_a}$
Maximum heat flow	$\dot{Q}_{\max} = C_{\min} \cdot (T_{h,i} - T_{c,i})$	$\dot{Q}_{\max} = \dot{m}_a \cdot (h_{a,i} - h_{a,sat,r})$
	$\dot{Q}_{act} = \mathbf{e} \cdot \dot{Q}_{\max}$	$\dot{Q}_{act} = \mathbf{e} \cdot \dot{Q}_{\max}$
Heat flow	$\dot{Q}_{act} = \dot{m} \cdot c_{p,h} \cdot (T_{h,i} - T_{h,o})$	$\dot{Q}_{act} = \dot{m}_a \cdot (h_{a,i} - h_{a,o})$
	$\dot{Q}_{act} = \dot{m} \cdot c_{p,c} \cdot (T_{c,i} - T_{c,o})$	$\dot{Q}_{act} = \dot{m}_r \cdot (h_{r,o} - h_{r,i})$
Effectiveness	$\mathbf{e} = 1 - e^{-NTU}$	$\mathbf{e} = 1 - e^{-NTU^*}$

The capacitance rates and temperatures in the effectiveness-NTU method for sensible heat exchangers are replaced in the analogy approach for wet coils by mass flow rates and enthalpies.

In the analogy approach the maximum possible heat transfer is calculated similar to the effectiveness-NTU for sensible heat exchanger. Thus, the maximum heat transfer rate is obtained as the product of the minimum capacitance rate, which is the mass flow rate of

the air, and the difference between the enthalpy of the entering air and the enthalpy of saturated air at the refrigerant inlet temperature.

$$\dot{Q}_{\max} = \dot{m}_a \cdot (h_{a,i} - h_{a,sat,r}) \quad (4-20)$$

Subsequently, the NTU^* and the effectiveness for the wet coil can be calculated by using the effectiveness-NTU relationships from Table 4.1:

$$NTU^* = \frac{U^* A}{\dot{m}_a} \quad (4-21)$$

$$\mathbf{e} = 1 - e^{-NTU^*} \quad (4-22)$$

The enthalpy of the outlet air can be calculated from the expression for the actual heat transfer rate. However, this is not enough to completely determine the outlet state of the air. Thus, a second analysis is done using the effectiveness-NTU method for the heat transfer between the condensate layer on the outside of the evaporator coil and the air. For this computation the condensate layer is assumed to be at uniform temperature. Consequently, the leaving dry bulb temperature is calculated and the humidity ratio, which fixes the state of the leaving air, can be calculated with the following equation:

$$\mathbf{w}_{a,o} = \frac{h_{a,o} - h_{dry\ air} \quad T=T_{a,o}}{h_{water\ vapor} \quad T=T_{a,o}} \quad (4-23)$$

Having determined the outlet state of the air leaving the evaporator, the sensible heat transfer can be calculated by equation (4-24).

$$\dot{Q}_{sens} = \dot{m}_a \cdot c_{p,a} \cdot (T_{a,i} - T_{a,o}) \quad (4-24)$$

Using the effectiveness relationships given in Table 4.1, a performance prediction can be performed knowing the inlet states and mass flow rates of the fluids entering the heat exchangers as well as the overall heat transfer area coefficient product of the heat exchanger. Dependent on whether there is sensible heat transfer only or sensible and latent heat transfer, the effectiveness-NTU method for sensible heat exchanger or the analogy approach for wet coils is used for predicting the cooling coil performance.

Determining the overall conductance area product is required to predict the coil performance. Figure 4.3 shows the thermal circuit for a coil, which consists of three thermal resistances. Inside the tube, where the refrigerant passes through, the heat transfer mechanism is convection. The same is true for the air that flows across the finned tubes. The thermal resistance due to conduction through the tube will be neglected in calculating the UA product. This is a reasonable assumption, since heat exchanger coils are manufactured of materials with a high thermal conductivity such as copper to decrease the thermal resistance due to conduction.

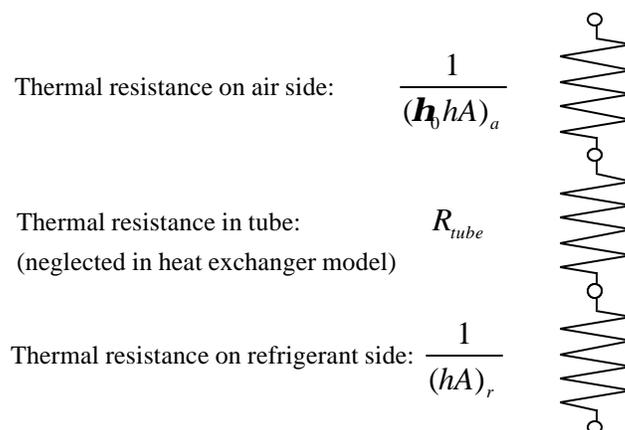


Figure 4.3: Generalized thermal circuit for a coil

If the coil surface temperature is higher than the dew point temperature of the air entering the evaporator, there is sensible heat transfer only in the evaporator and the effectiveness-NTU method for sensible heat exchangers is used. The UA product can then be calculated by the following equation:

$$UA = \frac{1}{\frac{1}{(\mathbf{h}_0 hA)_a} + \frac{1}{(hA)_r}} \quad (4-25)$$

The heat transfer coefficients will be obtained by fundamental heat transfer relationships for the refrigerant side and for the air side.

Condensation will occur if the coil surface temperature is below the dew point temperature of the air entering the evaporator. Consequently, there are two cases to consider; either the coil surface is partially wet or it is totally wet. In the case of a partially wet coil, the heat exchanger could be treated as a sensible heat exchanger up to the point where the coil surface temperature equals the dew point temperature. From then on, the coil is treated as a wet coil and the wet coil analogy approach is used. However, to determine the point where the coil surface temperature is equal to the dew point temperature requires a detailed analysis. A simpler approach is to treat the coil either as totally dry (effectiveness-NTU method for sensible heat exchangers) or as totally wet coil (effectiveness-NTU method for wet coils). Both methods slightly underpredict the actual heat transfer rate. Therefore, the higher heat transfer rate is used as the predicted performance. According to Braun et al. (1989), the error associated with this method is generally less than 5 %.

Using the wet coil analogy approach, the $U^* A$ product used in the wet coil analysis is calculated as shown below.

$$U^* A = \frac{1}{\frac{c_{p,a}}{(h_0 hA)_a} + \frac{c_s}{(hA)_r}} \quad (4-26)$$

Thus, the enthalpy based overall heat transfer area coefficient product can be calculated if the heat transfer coefficients for the refrigerant side and the air side are known. Subsequently, the heat exchanger performance can be performed by using the effectiveness-NTU relationships of the analogy approach for wet coils.

4.4 Conductance Area Product for Direct Expansion Coil

Direct expansion coils are manufactured out of material with a high thermal conductivity, such as copper, and have fins on the outside of the tubes. For flow across a bank of tubes the following equation can be found in Incropera and de Witt (1996), which was empirically found by Zhukauskas:

$$Nu_D = C_0 \cdot Re_D^m \cdot Pr^{0.36} \cdot \left(\frac{Pr}{Pr_s} \right)^{0.25} \quad (4-27)$$

This equation is generally valid for a flow across a bank of tubes independent of the fluid type. For direct expansion coils the air is this fluid. In the equation above C_0 and m are constants that depend on the heat exchanger configuration and the flow regime. A correction factor C_f for the equation above was introduced by Brandemuehl (1992) to account for condensation occurring on the outside of the finned tubes. This correction factor accounts for the additional thermal resistance due to the water film and the wet fin efficiency.

$$C_f = 0.626 \cdot u_a^{0.101}, \quad u_a \text{ in [ft/min]} \quad (4-28)$$

Thus, the Nusselt correlation becomes:

$$Nu_D = C_f \cdot C_0 \cdot Re_D^m \cdot Pr^{0.36} \cdot \left(\frac{Pr}{Pr_s} \right)^{0.25} \quad (4-29)$$

Equations (4-28) and (4-29) have to be written in terms of the thermal resistance to make use of the heat transfer correlations in the effectiveness-NTU methods. These equations can be written in terms of fluid properties, heat exchanger geometry and flow conditions using the dimensionless numbers.

$$Nu_D = \frac{h \cdot D_h}{k} \quad (4-30)$$

$$Re_D = \frac{u \cdot D_h}{\boldsymbol{n}} \quad (4-31)$$

$$Pr = \frac{\boldsymbol{m} c_p}{k} \quad (4-32)$$

The resulting expression could be used to determine the overall heat transfer area coefficient product if variables such as the heat exchanger geometry are known. However, most often these variables are not known and the overall heat transfer coefficient area product cannot be determined. This problem can be overcome by introducing new specific heat exchanger parameters. These parameters contain information about the heat exchanger such as the geometry and flow arrangement or conditions for a specific heat exchanger. Thus, the following expression is obtained for the air side of a dry coil with C_1 and C_2 as heat exchanger specific parameters.

$$(\mathbf{h}_0 hA)_a = C_1 \cdot k_a \cdot \left(\frac{\dot{m}_a}{\mathbf{m}} \right)^{C_2} \cdot \text{Pr}_a^{0.36} \cdot \left(\frac{\text{Pr}_a}{\text{Pr}_{a,s}} \right)^{0.25} \quad (4-33)$$

Parameter C_1 is a function of geometry only. Thus, it is a constant that is usually not known. Parameter C_2 is a function of flow conditions that is dependent on the heat exchanger tube configuration and the fluid velocity. The range of parameter C_2 according to Incropera and de Witt (1996) is between 0.4 for laminar flow to 0.84 for turbulent flow. The airflow is predominantly turbulent in evaporators to enhance heat transfer. Thus, operating in a turbulent flow regime, it is reasonable to assume C_2 to be constant. The values of C_1 and C_2 can be determined by fitting them to heat exchanger performance data.

While passing through the inside of the tube, the liquid refrigerant is evaporating. Finding a single correlation that can describe the heat transfer over the full quality range is difficult. In ASHRAE Fundamentals (1997) a table with heat transfer correlations for forced convection evaporation in tubes can be found. However, most of these equations are customized to certain fluid types, quality ranges, ranges of evaporating temperatures and tube geometries. Thus, it is difficult to develop a general valid expression for the heat transfer coefficient on the refrigerant side. In the work of Rabehl (1997) a refrigerant side heat transfer coefficient area product was found to work best by combining an empirical heat transfer correlation of the Trane Company with a correlation developed by Pierre (1955), which can be found in ASHRAE Fundamentals (1997). The following expression was developed and used for calculating the overall heat transfer area coefficient product for the evaporator:

$$(hA)_r = C_3 \cdot k_r \cdot \left[\left(\frac{\dot{m}_r}{\mathbf{m}} \right) \cdot \Delta h_r \right]^{0.45} \quad (4-34)$$

Again, C_3 is a parameter for the evaporator model, which is a function of geometry only, and will be determined in the fitting process.

4.5 Conductance Area Product for Condenser Coil

The condenser can be modeled by using the effectiveness-NTU method for sensible heat exchangers. Two modeling approaches are used which differ in modeling the refrigerant side of the condenser.

Inside the condenser tubes the superheated refrigerant vapor discharged from the compressor is cooled down and starts condensing when it reaches the dew point temperature. Often the fully condensed refrigerant is subcooled afterwards. In Figure 4.4, the three different regions of a condenser are shown schematically. The sensible heat transfer in the superheated region is usually small and can be taken into account by overestimating the overall heat transfer area coefficient product. However, having significant heat transfer in the subcooled region, a more accurate performance calculation can be performed if the condenser is split up into two or three different condenser regions. A separate heat exchanger calculation would be required for each of these regions (superheated, saturated and subcooled region).

In this thesis the refrigerant side was assumed to be isothermal which means the heat transfer occurs in the saturated region of the condenser only. Thus, the log mean temperature

and the effectiveness are significantly underestimated and have to be compensated by an increased overall heat transfer coefficient. According to ASHRAE Systems and Equipment (1992) this assumption is reasonable, since condensing takes place in approximately 85 % of the condenser area.

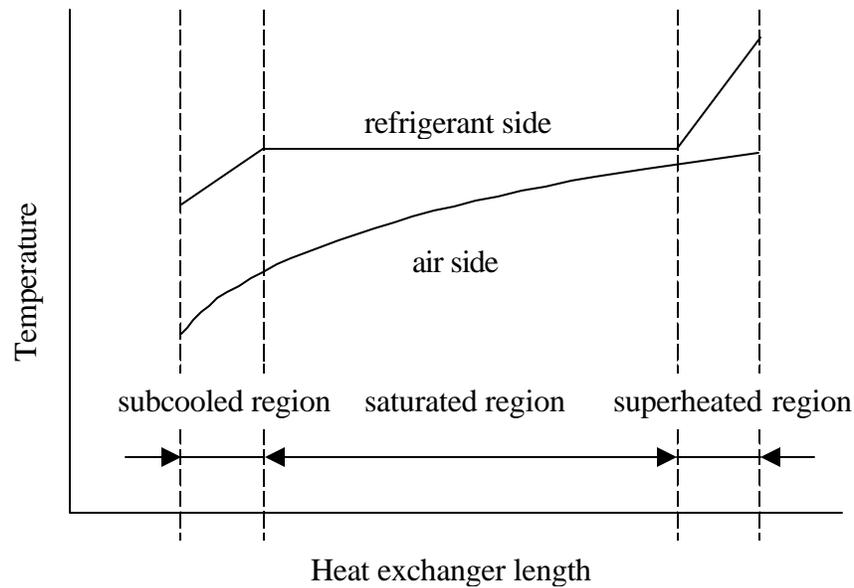


Figure 4.4: Different heat transfer regions in a condenser

The overall heat transfer coefficient can be calculated as function of the thermal resistances shown in Figure 4.3.

The same heat transfer correlation can be used for determining the air side heat transfer area coefficient product as for the direct expansion coil with a dry surface. Thus, the expression below is used.

$$(\mathbf{h}_0 hA)_a = C_4 \cdot k_a \cdot \left(\frac{\dot{m}_a}{\mathbf{m}} \right)^{C_5} \cdot \text{Pr}_a^{0.36} \cdot \left(\frac{\text{Pr}_a}{\text{Pr}_{a,s}} \right)^{0.25} \quad (4-35)$$

For condensation a relation needs to be developed. Rohsenow's book (1998) contains a correlation for condensation in horizontal tubes such as in air-cooled condensers, which is of the following form:

$$h_c = F \cdot \left[\frac{g \cdot \mathbf{r}_{f,r} \cdot (\mathbf{r}_{f,r} - \mathbf{r}_{g,r}) \cdot k_{f,r}^3 \cdot h_{fg,r}}{D \cdot \mathbf{m}_{j,r} \cdot (T_{cond,r} - T_{s,r})} \right]^{0.25} \quad (4-36)$$

In the expression above F is a coefficient, which is a function of the area of the tube circumference that is stratified. By assuming the coefficient F constant, the equation above can be modified and an expression can be developed, in which case a parameter C_6 is introduced. This parameter contains the constants from equation (4-37) for the heat transfer coefficient.

$$(hA)_r = C_6 \cdot \left[\frac{\mathbf{r}_{f,r} \cdot (\mathbf{r}_{f,r} - \mathbf{r}_{g,r}) \cdot k_{f,r}^3 \cdot h_{fg,r}}{\mathbf{m}_{j,r} \cdot (T_{cond,r} - T_{s,r})} \right]^{0.25} \quad (4-37)$$

For comparison in system modeling, a more detailed condenser model was developed in which the desuperheating region was taken into account. Thus, the condenser was split up and treated as two separate heat exchangers. In the first heat exchanger, the superheated vapor is cooled down to the point where condensation starts. The second heat exchanger covers the condensation range. By splitting the condenser up into two parts two separate heat exchanger calculations need to be performed, one for the desuperheating and one for the condensing region. The heat exchanger performance calculation for the saturated region was shown above. The overall heat transfer area coefficient product can be determined by equation (4-35) and (4-37). The UA product for the desuperheating region can be calculated by using equation (4-38) for the outer conductance area product.

$$(\mathbf{h}_0 hA)_a = C_7 \cdot k_a \cdot \left(\frac{\dot{m}_a}{\mathbf{m}} \right)^{C_8} \cdot \text{Pr}_a^{0.36} \cdot \left(\frac{\text{Pr}_a}{\text{Pr}_{a,s}} \right)^{0.25} \quad (4-38)$$

However, for the inside conductance area product a heat transfer correlation different from equation (4-36) must be used. In the ASHRAE Fundamentals Handbook (1997) a Nusselt-correlation can be found for turbulent flow for gases inside tubes. This correlation is recommended for cooling as well as for heating and is of the following form:

$$Nu_D = 0.023 \cdot \text{Re}_D^{0.8} \cdot \text{Pr}^{0.4} \quad (4-39)$$

By modifying this correlation an expression for the inner conductance area product can be obtained that is represented by the equation below:

$$(hA)_r = C_9 \cdot k_r \cdot \left(\frac{\dot{m}_r}{\mathbf{m}} \right)^{0.8} \cdot \left(\frac{\mathbf{m} \cdot c_{p,r}}{k_r} \right)^{0.4} \quad (4-40)$$

The fluid properties in equation (4-39) and (4-40) are based on the bulk temperatures. Consequently, the more detailed condenser model consists of six parameters instead of three. In Chapter 5, both condenser models will be used for modeling the system.

4.6 Heat Exchanger Model Validation

For the evaporator and condenser, the heat exchanger models were validated with manufacturer's catalog data. Performance prediction was tested within the range of the fitted data as well as outside the fitting range to determine the extrapolation capability of the models.

4.6.1. Objective Function for Parameter Fitting Procedure

The heat exchanger models for the condenser and the evaporator each contain three constant parameters that are obtained by using the method of least squares. The errors between the measured and calculated heat transfer rate are normalized by dividing each error by the mean of all measured values. Thus, the objective function for the heat exchanger parameter fitting procedure can be written as:

$$OF_{\dot{Q}} = \sqrt{\frac{\sum_{i=1}^N \left(\frac{\dot{Q}_{cat} - \dot{Q}_{calc}}{\dot{Q}_{cat,mean}} \right)^2}{N}} \quad (4-41)$$

4.6.2. Validation of the Evaporator Model

The evaporator model was validated with data from the Trane Coil-DS-1 catalog (1985), which contains data for heating and cooling coils. As a sample direct expansion coil, a PRIMA-FLO cooling coil was chosen. The catalog provides coil capacities as a function of suction temperature, face velocity as well as entering dry bulb and wet bulb temperature. Coil types and sizes are specified by the number of rows of tubes and the number of fins per length. Table 4.2 lists the characteristics of the direct expansion coil that was chosen for the evaporator model validation.

Table 4.2: Direct expansion coil used for model validation

DX-coil characteristics	
Number of rows	4
Fins per meter (fins per foot)	390 (120)
Face area [m ² (ft ²)]	1.6 (17.4)

Catalog data for the direct expansion coil in Table 4.2 were available in a flow range from 2 to 3.6 m/s (400 to 700 ft/min), in a suction temperature range from 1.7 to 7.2 °C (35 to 45 °F), and in a air entering temperature range from 21.1 to 35 °C dry bulb and 15.6 to 23.9 °C wet bulb (70 to 95 °F dry bulb and 60 to 75 °F wet bulb). For all data sets the condensing temperature was 40.6 °C (105 °F).

The three parameters (C_1 and C_2 from equation (4-33) and C_3 from equation (4-34)) were obtained for three different sets of catalog data to determine the effect on the heat exchanger parameter and to check the extrapolation capability of the model.

Each data set consists of twelve operating points. The data points of set a in Table 4.3 have a suction temperature of 4.4 °C and cover the full flow and entering air temperature range. Data set b contains data for a fixed air entering temperature of 26.7 °C dry bulb and 19.4 °C wet bulb, whereas data set c contains data in the flow range from 2.5 to 3 m/s (500 to 600 ft/min) only. Table 4.3 lists the heat exchanger parameters and the error for the fitting process as well as the error as a result of the model validation for all catalog data points. For all data sets the error in predicting the sensible heat transfer was smaller than 0.05 %.

Table 4.3: Heat exchanger parameters for direct expansion coil for data sets a to c

	C_1	C_2	C_3	Error for fitting [%]	Error for validation [%]
Set a	1358	0.5	1211	0.25	0.3578
Set b	1333	0.5	1231	0.19	0.41
Set c	959.3	0.5	1670	1.17	1.75

Parameter C_2 in Table 4.3 was set to be constant. As described in Chapter 4.4 the parameter C_2 is within the range of 0.4 to 0.84. However, parameters C_1 and C_2 affect each other considerably and make an optimization for the heat exchanger parameter difficult. Thus, by setting C_2 to a fixed number, this problem can be overcome. Figure 4.5 shows the effect of parameter C_2 on the error in the fitting process for the three data sets.

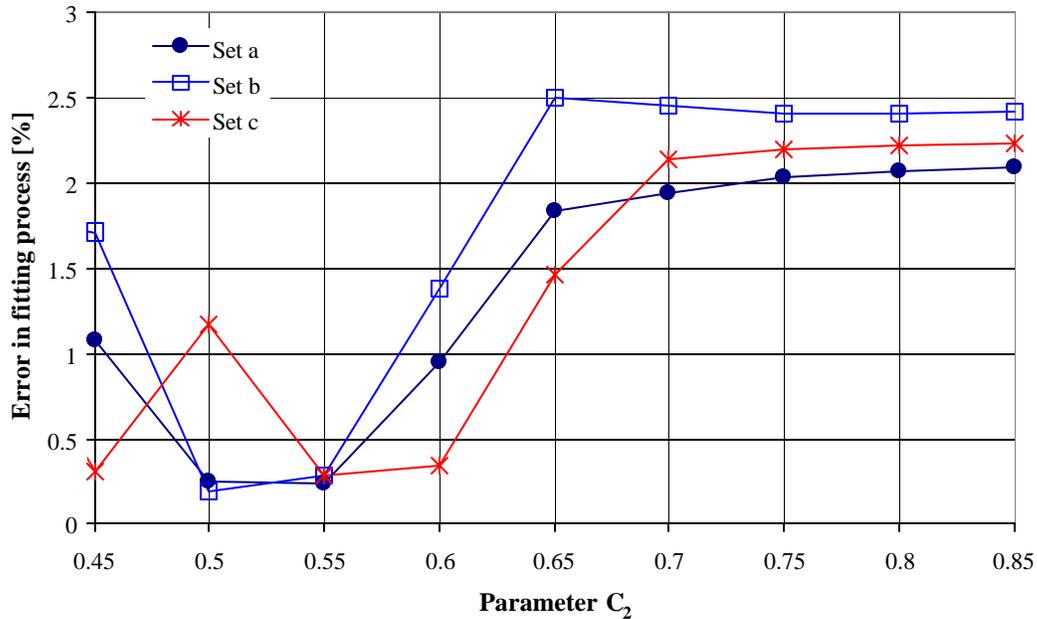


Figure 4.5: Effect of parameter C_2 on the error in the fitting procedure

From Figure 4.5 it can be seen that 0.5 as a fixed value for C_2 is a reasonable approximation and yields very small errors. Optimization of the heat exchanger parameters in the fitting procedure is difficult and depends strongly on starting guess values used for the parameter optimization. Most likely, a different set of C_1 and C_3 could be found for data set c that yields to an error of similar size as the error obtained for $C_2=0.45$ and $C_2=0.55$.

The coefficients presented in Table 4.3 for data sets a to c were used to determine the extrapolation capability of the direct expansion coil model. Figure 4.6 to Figure 4.8 show the predicted heat exchanger performance for the three data sets. Figure 4.6 shows the extrapolation to lower and higher suction temperatures (data set a), whereas Figure 4.7 show the performance prediction for different entering air temperatures (data set b). Both figures show that the model allows confident extrapolation.

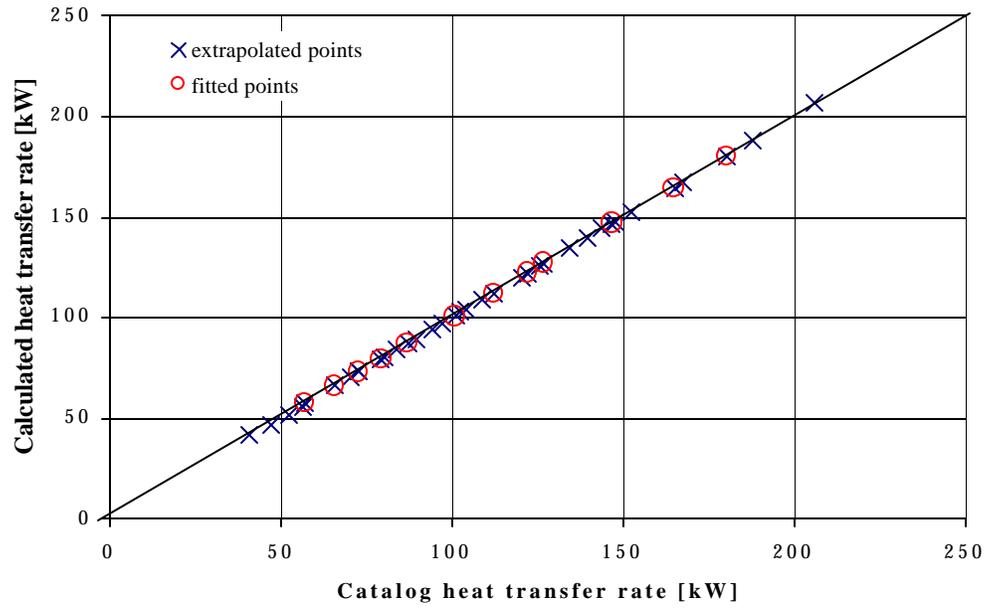


Figure 4.6: Extrapolation of data set a to lower and higher suction temperatures

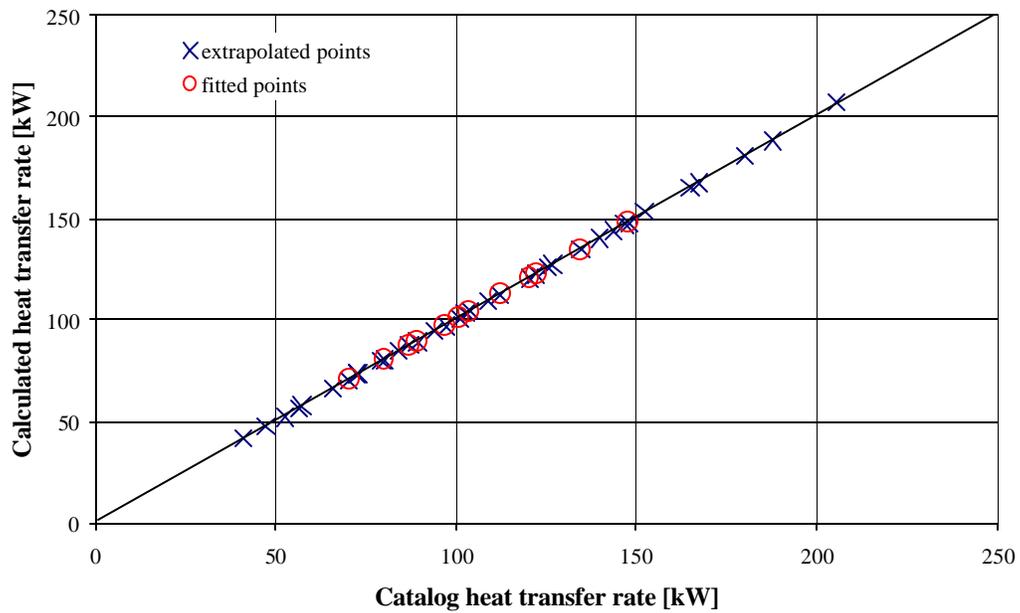


Figure 4.7: Extrapolation of data set b to lower and higher air entering temperatures

In direct expansion coils different mass flow rates on the air and the refrigerant side are often encountered. This results in different fluid velocities and affects the overall heat transfer coefficient. The variation of the overall heat transfer coefficient is taken into account by using fundamental equations for the heat transfer coefficients on the air and the refrigerant side. Figure 4.8 shows the predicted coil performance with extrapolation to lower and higher face velocities on the air side.

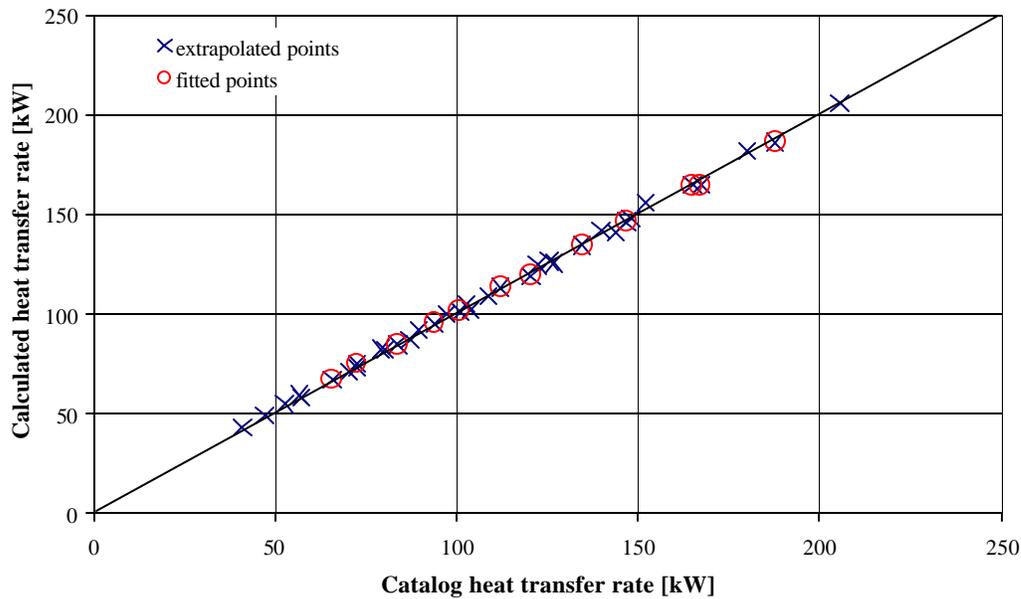


Figure 4.8: Extrapolation of data set c to higher and lower face velocities

4.6.3. Validation of the Condenser Model

The three parameter condenser model was validated by using Bohn condensing coil catalog data (1988). The validation of the condenser model was made for a condensing coil with the following characteristics:

Table 4.4: Condensing coil used for model validation

Condensing-coil characteristics	
Number of rows	6
Fins per meter (fins per foot)	51 (168)
Face area [m ² (ft ²)]	0.17 (1.8)

The performance information for the condensing coil was given in the form of a coil capacity curve that allows the creation of performance data in the range of 1 to 4 m/s (200 to 800 ft/min) for different condensing and air entering conditions.

Ten catalog data points were chosen with an air inlet temperature of 35 °C (95 °F) and a condensing temperature of 51.7 °C (125 °F). The face velocity was in the range from 2 to 3.3 m/s (400 to 640 ft/min) for the fitting points. The following values of the three parameters C_4 and C_5 from equation (4-35) and C_6 from equation (4-37) were determined.

Table 4.5: Heat exchanger parameters for condensing coil

C_4	C_5	C_6	Error for fitting [%]
1358	0.5	1211	0.67

Figure 4.9 shows the performance prediction of the condenser model for the data points used in the fitting process as well as the predicted performance for higher and lower flow rates (1 to 4 m/s, (200 to 800 ft/min)), ambient (29.4 to 40.6 °C (85 to 105 °F)) and condensing temperatures (46.1 to 57.2 °C (85 to 105 °F)). The extrapolation of the flow rate, however, is not as important as for the evaporator. The fluid velocity on the air-side is

generally constant, since the condenser fans run at constant speed, whereas on the refrigerant side the fluid velocity changes, since it is related to the actual capacity.

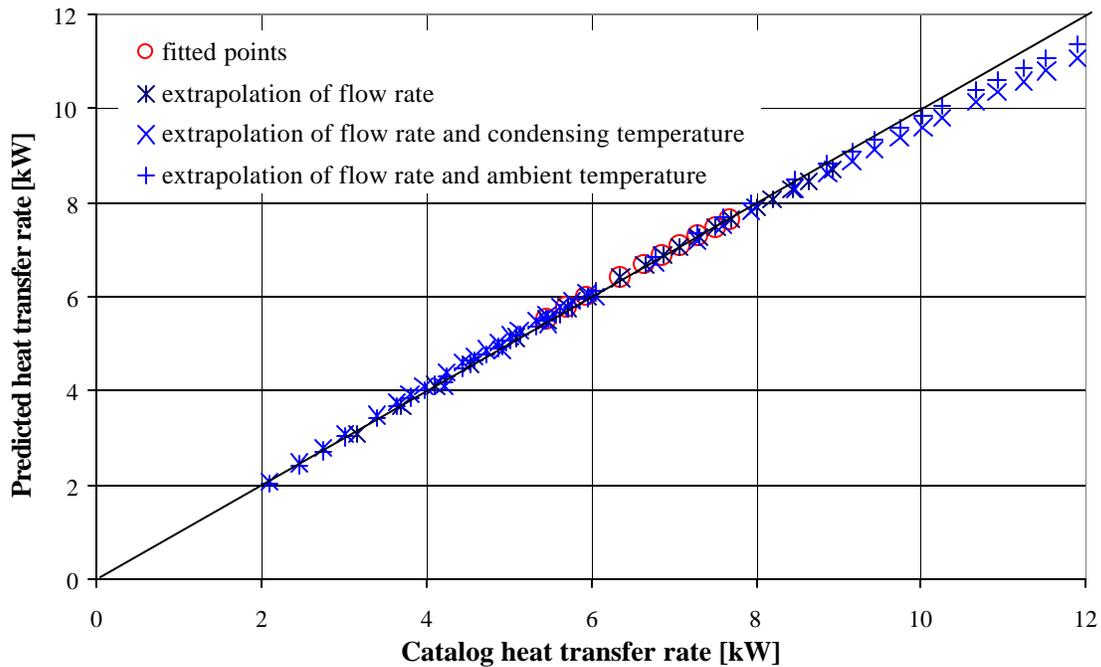


Figure 4.9: Performance prediction with the condenser model

Table 4.6 lists the mean weighted errors for the predicted condensing coil performance shown in Figure 4.9.

Table 4.6: Errors for performance prediction for the condensing coil

Mean weighted error for performance prediction	[%]
Extrapolation of flow rate	0.7
Extrapolation of flow rate and condensing temperature	3.9
Extrapolation of flow rate and ambient temperature	1.9

4.7 Conclusions

Two heat exchanger models were presented for performance prediction of the evaporator and condenser. Both models are based on effectiveness-NTU relationships. The overall heat transfer area coefficient product UA (UA^*) that is required for the performance prediction is based on fundamental heat transfer correlations. Thus, the UA product is adjusted according to the temperature level and the flow conditions in the heat exchangers. Consequently, the extrapolation ability of both models is very good as shown for the two heat exchanger models.

The fundamental heat transfer correlations were modified and heat exchanger specific parameters were introduced, which can be determined by fitting them to performance data of the heat exchanger. Thus, no detailed knowledge of the geometry of the heat exchanger is required. This allows the heat exchanger models to be used for system modeling where the specific heat exchanger geometry is not known.

Chapter 5

System Modeling

5.1 Introduction

The vapor compression cycle is widely used for air conditioning and refrigeration applications. It is used in small applications such as household refrigerators as well as in large refrigeration applications with a cooling capacity of a couple of 100 kilowatts. Basically, two different kinds of approaches exist for modeling the performance of a whole system such as a vapor compression cycle.

Polynomial expressions can be used for the prediction of the compressor power draw, the total capacity, and sensible capacity of a vapor compression cycle. The performance of a vapor compression cycle, however, is a function of four input parameters at least, since the performance depends on the entering air dry bulb temperature, the wet bulb temperature, the volumetric flow rate on the evaporator side and the ambient temperature. Thus, these polynomial expressions contain many curve fit parameters that have to be determined. Stoecker (1971) presented polynomial expressions for the capacity and power requirement of a centrifugal chiller, where 18 coefficients have to be determined in order to predict the system's performance. Also, as mentioned in Chapter 3 and 4, extrapolation of performance data using these polynomial expressions is often uncertain.

Consequently, methods for modeling primary system equipment are used that are based on fundamental physical principals rather than on polynomial expressions. Thus, confident extrapolation of performance data can be performed. In these cases, the system is

modeled by using semi-mechanistic models as presented for the single components in Chapter 3 and 4. These component models are then interconnected according to the system layout.

5.2 The Vapor Compression Cycle

Basically, the simplest vapor compression cycle consists of two heat exchangers, an expansion device, and a compressor. Often vapor compression systems have multiple stages to improve the cycle's efficiency. Rooftop packaged air conditioning units, however, use a simple single stage refrigeration cycle. Figure 5.1 shows such a single stage vapor compression cycle with its components.

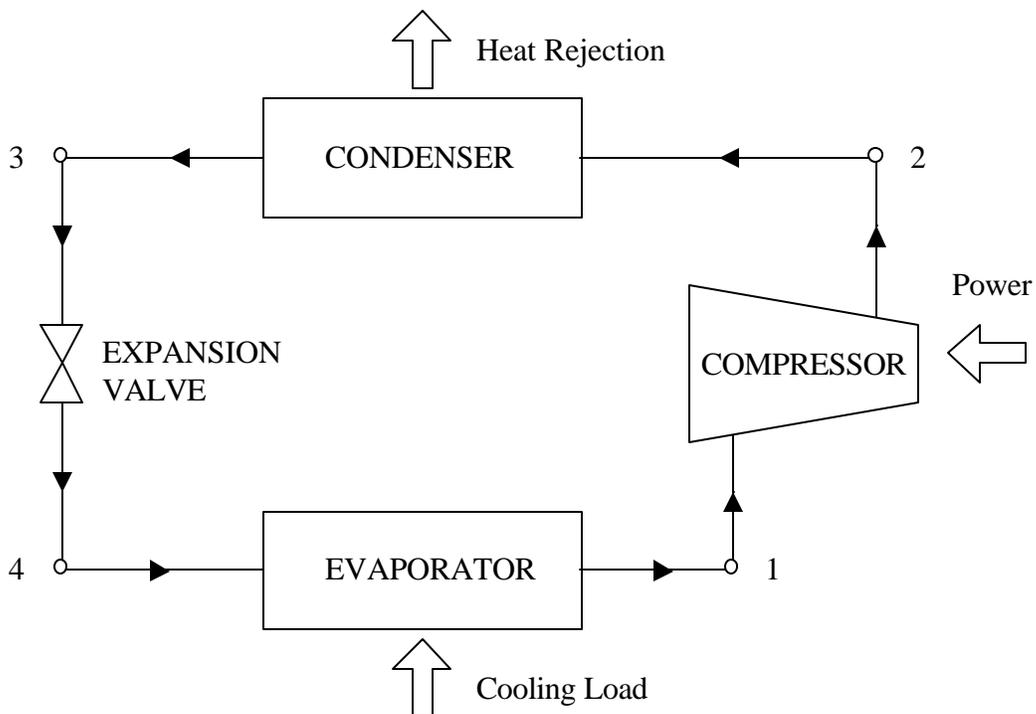


Figure 5.1: Single stage vapor compression cycle

For a rooftop packaged unit these components are a direct expansion coil as the evaporator, a reciprocating or scroll compressor, and an air-cooled condenser. These components are described earlier in Chapter 3 and Chapter 4.

The vapor compression cycle operates between two temperature levels. The evaporator operates at the lower temperature level. In the evaporator a warm air stream is cooled down to the desired temperature. The cooling load depends on the air entering conditions and the flow rate. In the evaporator the cooling load is picked up by the refrigerant that is used as the working fluid in the vapor compression cycle. The refrigerant entering the evaporator is liquid and is at low temperature and pressure. Thus, heat can be transferred from the warm air stream to the cold refrigerant. Ideally, the refrigerant evaporates in the tube at constant pressure and reaches state point 1 as shown in Figure 5.2. The liquid refrigerant is now fully evaporated and slightly superheated through the action of the expansion device (point 1'). The refrigerant vapor is superheated to prevent the compressor from liquid slugging. The slightly superheated refrigerant vapor is then compressed to a high-pressure level. At a high pressure and temperature the refrigerant is discharged from the compressor and enters the condenser. In the condenser the heat, which consists of the cooling load and the power supplied to the refrigerant during the compression process, is rejected to the ambient. Thus, the superheated refrigerant is cooled down and condenses. Often, the fully condensed refrigerant is subcooled when leaving the condenser. Leaving the condenser the refrigerant is adiabatically expanded and enters the evaporator. The thermodynamic state points of this idealized vapor compression cycle are shown in Figure 5.2. In reality pressure drops occur on the condenser and evaporator side inside the tubes. Thus, condensing and evaporating temperatures decrease with pressure drop.

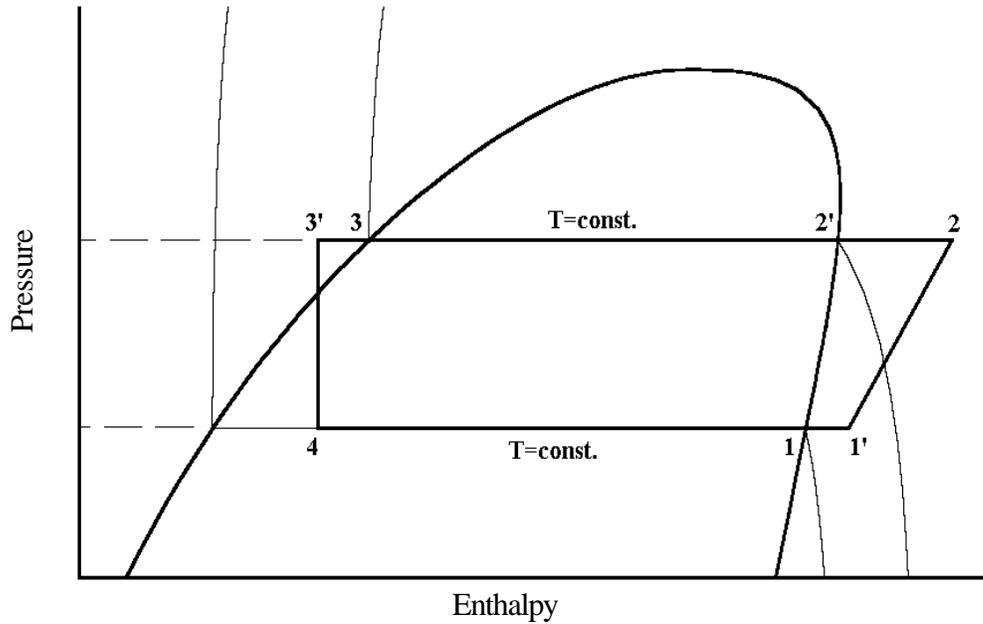


Figure 5.2: Vapor compression cycle in pressure-enthalpy diagram

5.3 System Modeling

For modeling the system, the component models described in Chapters 3 and 4 are used for the vapor compression cycle model. The models have to describe the performance of each single component used in the rooftop packaged unit in order to predict the overall performance of the system. Performance data for the condenser and evaporator are generally not available to determine the heat exchanger parameter directly. The compressor type is usually given and compressor performance data can be obtained. Thus, the parameters of the compressor model can be obtained by fitting the compressor model to the compressor

performance data. Having determined these parameters the compressor model is available for calculation of the thermodynamic state points of the vapor compression cycle.

The catalog information for rooftop packaged units provided by most manufacturers gives the capacity (sensible and total) and the compressor power draw as a function of the air temperature entering the condenser, the dry and wet bulb air temperature entering the evaporator, and the air flow rate through the evaporator. Available catalog information for different operating conditions, however, does not specify any thermodynamic state points of the vapor compression cycle. Thus, certain assumptions have to be made in order to specify the state points of the vapor compression cycle and determine the rate equations.

The cycle that is used for modeling the packaged unit is an idealized vapor compression cycle such as shown in Figure 5.2. In this case the assumption was made that the refrigerant leaving the evaporator is saturated vapor (point 1). Also the refrigerant leaving the condenser was assumed to be saturated liquid (point 3). The thermodynamic state points and the rate equations can then be calculated for the vapor compression cycle. This leads to the following computation scheme for the assumed vapor compression cycle in a rooftop packaged air conditioning unit.

The total evaporator heat transfer rate and the compressor power draw are given as catalog information for the different operating points:

$$\dot{Q}_{evap} = known \quad (5-1)$$

$$P_{comp} = known \quad (5-2)$$

The energy balance for the evaporator and the compressor can be written as:

$$\dot{Q}_{evap} = \dot{m}_r \cdot (h_{1,r} - h_{4,r}) \quad (5-3)$$

$$P_{comp} = \dot{m}_r \cdot (h_{2,r} - h_{1,r}) \quad (5-4)$$

The expansion from state point 3 to point 4 is assumed to be adiabatic and is therefore isenthalpic:

$$h_{3,r} = h_{4,r} \quad (5-5)$$

The mass flow rate as well as the power draw of the compressor is obtained from the compressor model as a function of the condensing and evaporating pressures:

$$\dot{m}_r = f(p_{cond}, p_{evap}) \quad (5-6)$$

$$P_{comp} = f(\dot{m}_r, p_{cond}, p_{evap}) \quad (5-7)$$

Assumptions for the idealized vapor compression cycle:

$$h_{1,r} = ENTHALPY(p_{evap}, x=1) \quad (5-8)$$

$$h_{3,r} = ENTHALPY(p_{cond}, x=0) \quad (5-9)$$

The 9 equations above containing 9 unknowns allow the thermodynamic state points of the idealized vapor compression cycle to be calculated. Also, the refrigerant mass flow rate is determined by the set of equations above. Thus, the state points of the thermodynamic cycle as well as the rate equations needed for the heat exchanger modeling can be determined.

The equations above contain no direct information about the temperature levels of the heat exchangers in the rooftop packaged unit, but the catalog information given as a function

of the return air temperature, the outdoor air temperature and the evaporator air flow rate contains indirectly the information to determine the temperature or pressure level in the evaporator and condenser. This can be best shown by plotting the thermodynamic state points of the assumed vapor compression cycle in a p, h-diagram. In the graph below the state points are shown as well as a sketch of the temperature levels of the air in the condenser and evaporator.

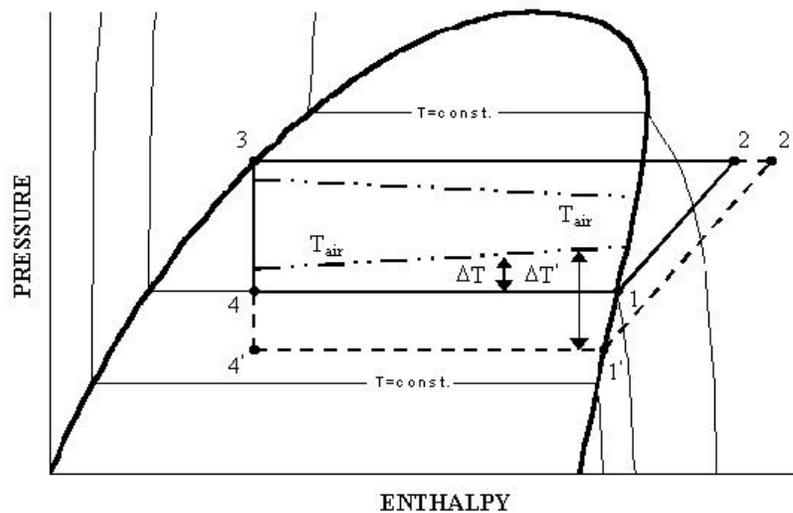


Figure 5.3: Schematic pressure-enthalpy diagram

For a specific operating point the outdoor temperature and return air temperature are fixed numbers. The catalog provides the power draw and evaporator heat transfer rate. The sizes of the heat exchangers used in the unit are reflected in the power draw for this operating point. With decreasing heat exchanger size the temperature difference ΔT increases to $\Delta T'$. Since the return air temperature is a fixed value, the evaporating temperature has to decrease

(4 → 4' and 1 → 1'), which is reflected in a higher compressor power draw (2 → 2'). In Figure 5.3 this fact is shown in a P-h diagram by the dashed line.

5.4 Model Validation

The rooftop packaged air conditioning model was validated for three different rooftop packaged air conditioning units with cooling capacities from 15 ton to 30 ton. All models use R-22 as the refrigerant.

Table 5.1: Rooftop packaged units

	(1)	(2)	(3)
Manufacturer	Carrier	Carrier	Carrier
Unit type	48 TJ 016	48 TJ 024	50 EJ 034
Cooling Capacity	53 kW (15 tons)	70 kW (20 tons)	106 kW (30 tons)
Compressor Type	06DE5371 (reciprocating)	06DE8241 (reciprocating)	06DE5371 (reciprocating)
Quantity	1	2	2
Evaporator face area	1.7 m ² (17.9 ft ²)	1.7 m ² (17.9 ft ²)	3.0 m ² (31.7 ft ²)

The validation of the model included the following steps:

1. Calculation of the parameters of the compressor model to predict the mass flow rate (C, Δp)

2. Calculation of the parameters of the compressor model to predict the compressor power draw (a, b)
3. Calculation of the thermodynamic state points and rate equations of the assumed vapor compression cycle
4. Parameter fitting for the evaporator model by using the data obtained in step three (C_1 , C_2 and C_3)
5. Parameter fitting for the condenser model by using the data obtained in step three (C_4 , C_5 and C_6)
6. Using the coefficients that were obtained in steps 1 to 5 in the rooftop packaged air conditioning model to predict the compressor power draw, the sensible capacity and the total capacity

It should be noticed that steps 3 to 5 could be performed at the same time. The parameters for the heat exchanger, however, were not fitted at the same time. Primarily, this was not done because the optimization problem is highly nonlinear and an optimization even for only three parameters at one time as done in step 4 and 5 is difficult. In addition, parameter values for the heat exchanger models affect each other. Parameter values for the condenser model could be found that would yield a better condenser fit, while the evaporator fit would be worse if all parameters were fitted at the same time.

The model validation will be shown for the rooftop packaged unit (3) in Table 5.1. This rooftop packaged unit was chosen, since the data range given in the catalog was the broadest. Thus, it allows the best opportunity to check the extrapolation ability of the complete model. The model results for units (2) and (3) can be found in Appendix B.

First, the compressor parameters C and Δp were fitted to the available compressor performance data. Next, the compressor parameters a and b for the power prediction were fitted. Thus, the following compressor parameters were obtained by using 14 catalog data points:

Table 5.2: Compressor parameters for packaged unit (3)

Compressor type	06DE5371
Clearance Factor C	0.071
Pressure drop parameter Δp	0.094
Parameter for compressor efficiency a	0.716
Parameter for compressor efficiency b	-0.191
Error for mass flow rate [%]	2.0
Error for power draw [%]	2.9

The compressor used in this rooftop packaged unit is identical to the compressor (5) in Chapter 3. In this chapter, the performance prediction and the extrapolation ability of the compressor model was demonstrated by using this compressor as a sample compressor.

The compressor model was then used in the calculation scheme for the rooftop packaged unit (equations (5-1) to (5-9)). Different data sets were chosen to allow the extrapolation ability of the model to be checked. These data sets are shown in Table 5.3.

Table 5.3: Data sets (a) to (d) for determining the model parameters

Data set	Flow rate [l/s (ft ³ /min)]	Range of		
		Ambient temperature [°C, (°F)]	Dry bulb return air temperature [°C, (°F)]	Wet bulb return air temperature [°C, (°F)]
(a)	4248-5663	23.9-46.1	26.7	16.7-22.2
	(9000–12000)	(75-115)	(80)	(62-72)
(b)	2832-7079	29.4-40.6	26.7	16.7-22.2
	(6000-15000)	(85-105)	(80)	(62-72)
(c)	2832-7079	23.9-46.1	26.7	19.4
	(6000-15000)	(75-115)	(80)	(67)
(d)	2832-7079	23.9-46.1	26.7	16.7-22.2
	(6000-15000)	(75-115)	(80)	(62-72)

Each of the data set (a) to (d) consists of eight data points. All data points refer to a dry bulb return air temperature of 26.7 °C (80 °F). For the flow rate, the ambient temperature and the return air wet bulb temperature, the extreme values for each range shown in Table 5.3 were picked. Data set (d) reflects the full range of performance information given in the catalog. Data set (a) allows the extrapolation ability to be checked for the supply air flow rate, whereas data set (b) allows the extrapolation ability to be checked for the ambient temperature. Data sets (c) and (d) allow extrapolation for the entering air dry and wet bulb temperatures.

5.4.1. Validation for Idealized Vapor Compression Cycle

Based on the idealized vapor compression cycle shown in Figure 5.3 the thermodynamic state points were calculated for data sets (a) to (d). Thus, the heat exchanger parameters could be fitted. Table 5.4 shows the heat exchanger parameter values that were obtained in the parameter fitting procedure for the evaporator. It also shows the error for the sensible and total heat transfer rate prediction as a mean weighted RMS-error. The errors for all four data sets were below 6 %.

Table 5.4: Evaporator model parameters for rooftop packaged unit (3) for data sets (a) to (d)

	(a)	(b)	(c)	(d)
Outer heat transfer parameter C_1	1536	1695	1613	1650
Outer heat transfer parameter C_2	0.5	0.5	0.5	0.5
Inner heat transfer parameter C_3	3588	3246	3599	3607
Error for sensible heat transfer rate [%]	0.6	4.4	3.3	4.0
Error for total heat transfer rate [%]	1.5	5.2	5.8	5.8

In Chapter 3 it was shown that it is a reasonable assumption to set the heat exchanger parameter C_2 to a constant value of 0.5. Thus, for the evaporator parameter fitting procedure this parameter was also set to this value. Figure 5.4 shows the error of the total heat transfer prediction for different values of C_2 . An optimization where all three parameters are fit was found to be very difficult. Due to the three-dimensional optimization problem, parameter values were found that do not lead to the global minimum of the error for the total heat transfer rate prediction. Thus, setting the heat exchanger parameter C_2 simplified the

optimization problem and still led to reasonable results for the model parameters. Setting C_2 to fixed value of 0.5 did not lead to the global minimum for all data sets. However, a value of 0.5 is physically reasonable as shown in Chapter 3 and a good performance prediction can be achieved for the whole system as it is shown later.

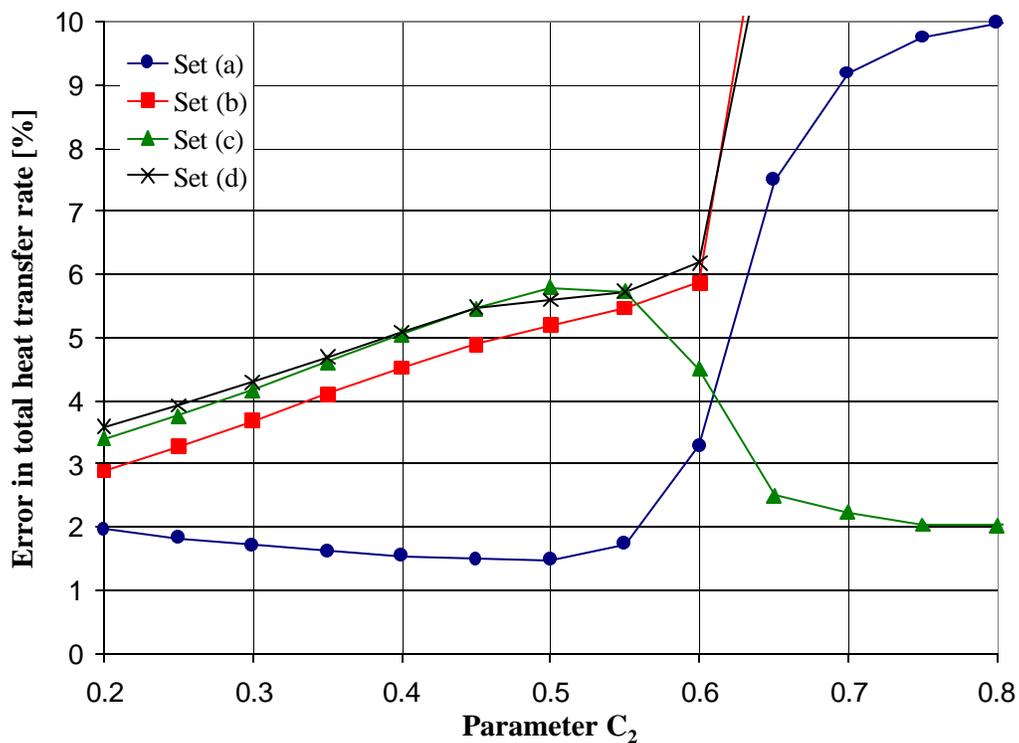


Figure 5.4: Effect of parameter C_2 on error in optimization

Figure 5.5 to Figure 5.8 show the predicted sensible and total evaporator heat transfer rate versus the heat transfer rate that was calculated based on the idealized vapor compression cycle. Figure 5.5 and Figure 5.6 show the predicted heat transfer rate versus the catalog heat transfer rate for data set (a) with the lowest error in heat transfer prediction of all four data sets, whereas Figure 5.7 and Figure 5.8 show the same graphs for data set (d) with the highest error in heat transfer prediction.

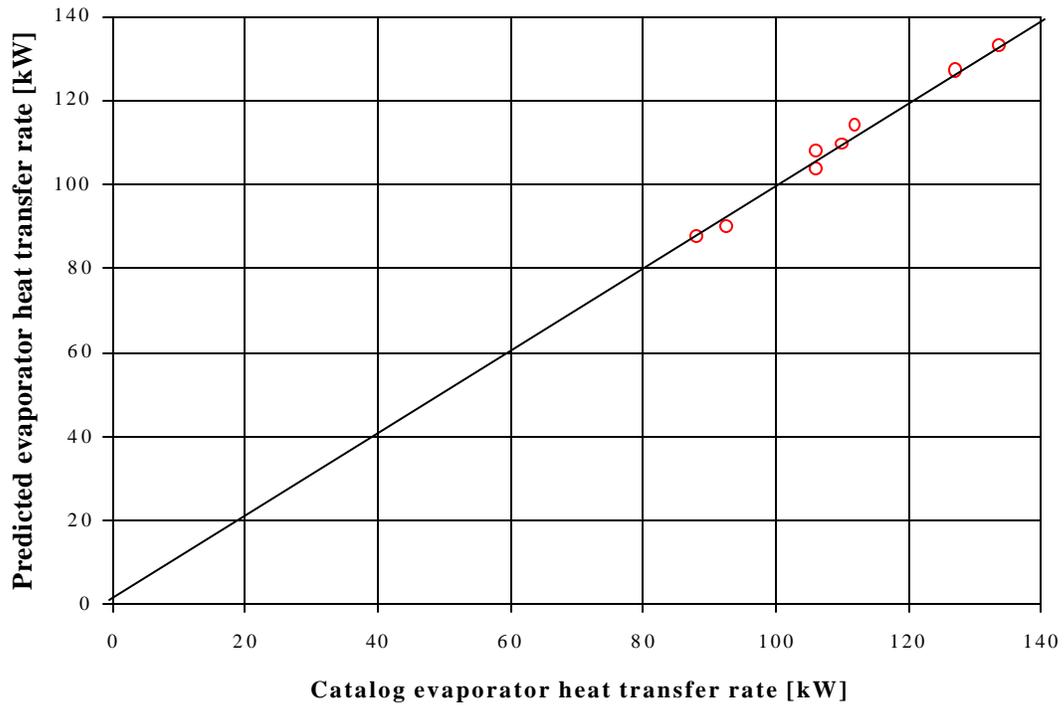


Figure 5.5: Predicted versus catalog total evaporator heat transfer rate for data set (a)

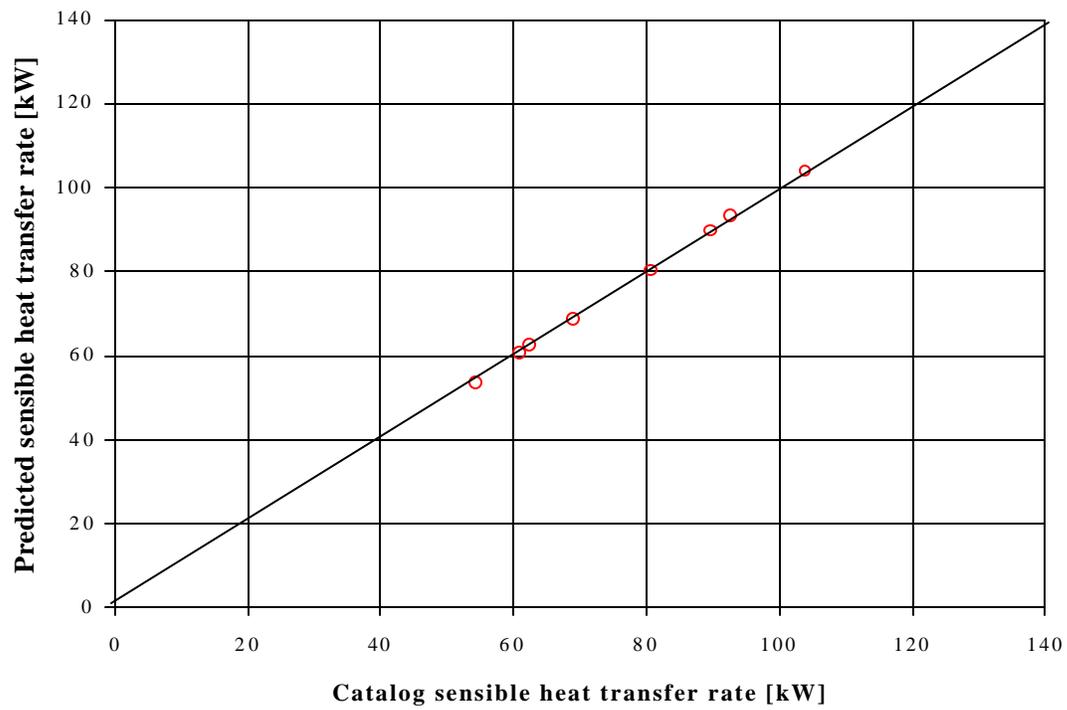


Figure 5.6: Predicted versus catalog sensible evaporator heat transfer rate for data set (a)

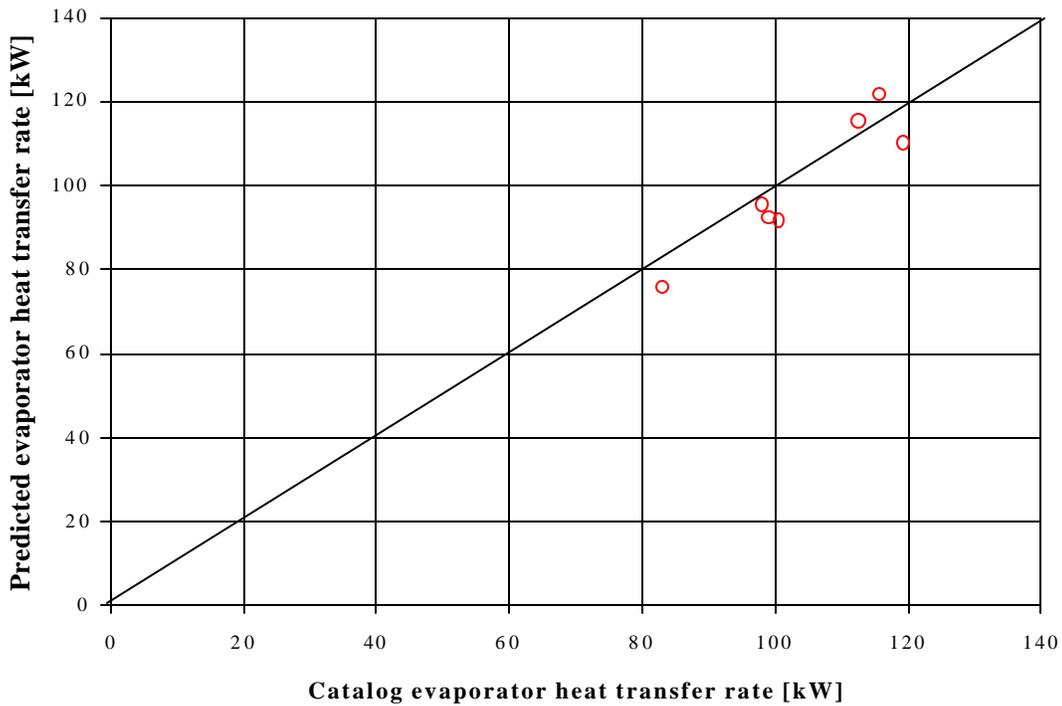


Figure 5.7: Predicted versus catalog total evaporator heat transfer rate for data set (d)

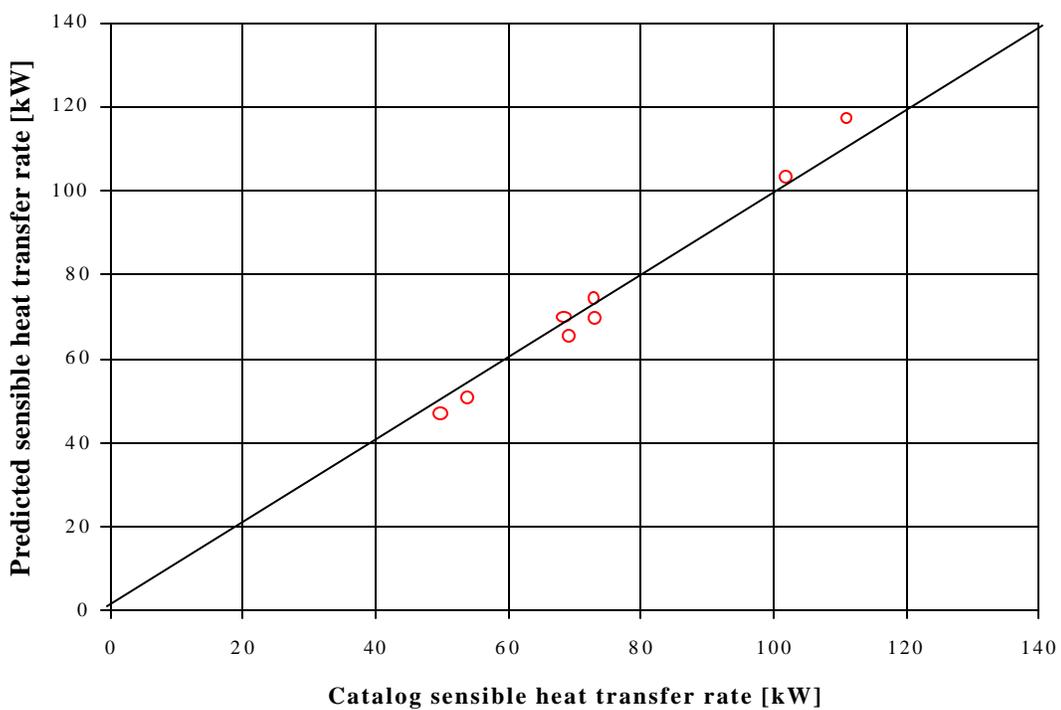


Figure 5.8: Predicted versus catalog sensible evaporator heat transfer rate for data set (d)

In contrast to the heat exchanger performance prediction in Chapter 3, the predicted performance for the heat exchanger used in the rooftop packaged unit is not as good. Since for both parameter fitting procedures the same heat exchanger model was used, the difference in performance prediction is due to the different data sets used in the fitting processes. In Chapter 3 catalog heat exchanger performance data were used. For the rooftop packed unit model, there were no catalog performance data available for the evaporator alone. Thus, for the parameter fitting procedure, data sets were used that were calculated based on the assumptions for the idealized vapor compression cycle. Consequently, the calculated data set performance deviates from the catalog performance of the rooftop packaged unit and its components. The calculated data set not only deviates due to the assumptions that there is no superheat at the exit of the evaporator and no subcooling at the exit of the condenser, but also because the compressor model, which also had some error in its performance prediction, had to be used to determine the state points of the vapor compression cycle.

Table 5.5 shows the parameter values for the condenser model for the rooftop packaged unit (3) obtained by using data set (a) to (d).

Table 5.5: Condenser model parameters for rooftop packaged unit (3) for data sets (a) to (d)

	(a)	(b)	(c)	(d)
Outer heat transfer parameter C_4	5549	6203	5783	5581
Outer heat transfer parameter C_5	0.5	0.5	0.5	0.5
Inner heat transfer parameter C_6	31893	36315	33304	32015
Error for condenser heat transfer rate [%]	9.6	4.8	7.4	9.5

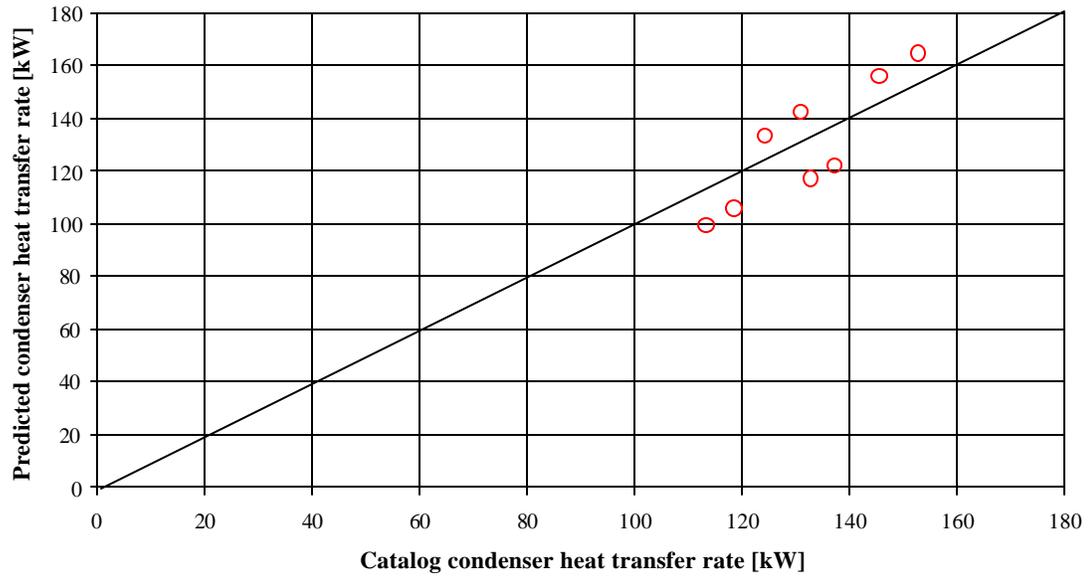


Figure 5.9: Predicted versus catalog condenser heat transfer rate for data set (a)

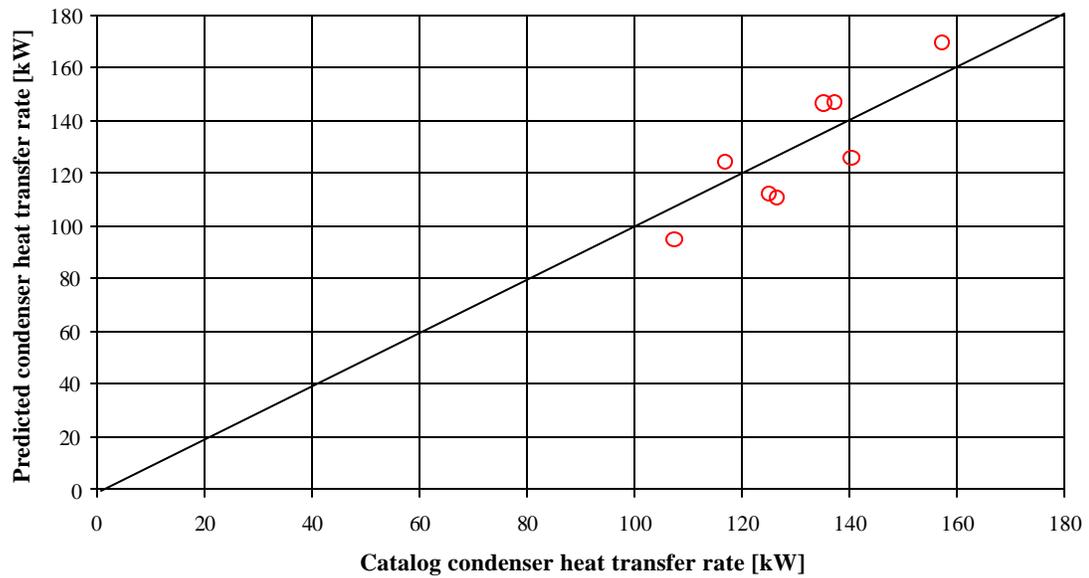


Figure 5.10: Predicted versus catalog condenser heat transfer rate for data set (d)

As shown in Figure 5.9 and Figure 5.10 the condenser fit is not as good as the fit for condenser data as presented in Chapter 3. The errors for data sets (a) to (d) were in the range of 4.8 to 9.6 %. This is also due to the assumptions of the idealized vapor compression cycle. However, the error in the heat transfer prediction is also larger because the heat transfer in the condenser was assumed to occur only in the saturated region. Thus, the effectiveness is considerably underestimated and the overall heat transfer coefficient has to compensate. By taking the desuperheating into account, the performance prediction with the condenser model can be improved by about 3 % on average as shown in Table 5.6.

Table 5.6: Parameters for detailed condenser model for rooftop packaged unit (3) for data sets (a) to (d)

	(a)	(b)	(c)	(d)
Outer heat transfer parameter C_4	3585	3694	3619	3590
Outer heat transfer parameter C_5	0.5	0.5	0.5	0.5
Inner heat transfer parameter C_6	19031	19802	19171	19027
Error for heat transfer in saturated region [%]	5.6	2.8	4.2	5.4
Outer heat transfer parameter C_7	329.5	296	296	330.9
Outer heat transfer parameter C_8	0.5	0.5	0.5	0.5
Inner heat transfer parameter C_9	479.2	475.7	475.7	479.9
Error for heat transfer in superheated region [%]	8.7	4.5	6.7	8.6
Error for condenser heat transfer rate [%]	5.9	3.0	4.5	5.8

Figure 5.11 and Figure 5.12 show the predicted condenser heat transfer rate with the detailed condenser model for data set (a) and (b).

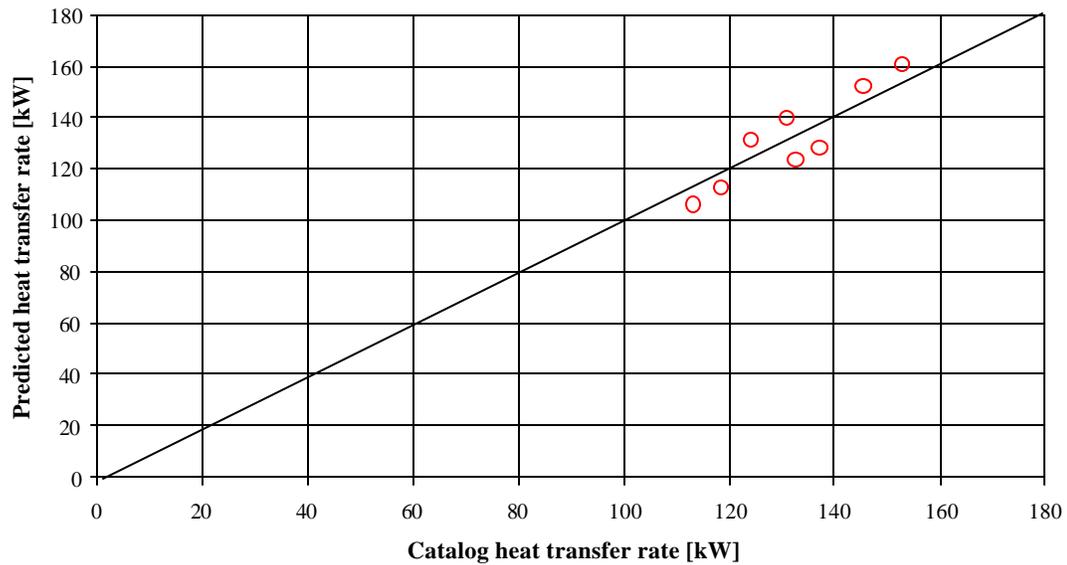


Figure 5.11: Predicted versus catalog condenser heat transfer rate for data set (a) using detailed model

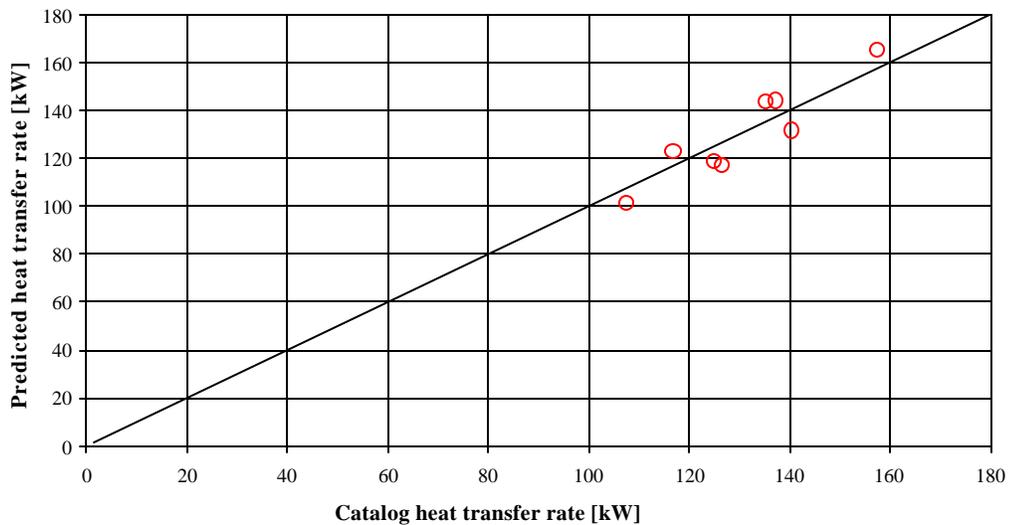


Figure 5.12: Predicted versus catalog condenser heat transfer rate for data set (d) using detailed model

After having determined the model parameters for the compressor, the evaporator and the condenser, the parameters can be used for the performance validation of the rooftop packaged air conditioning unit. Figure 5.13 shows the input variables and the parameters that have to be provided to the program as well as the output variables of the rooftop packaged unit model.

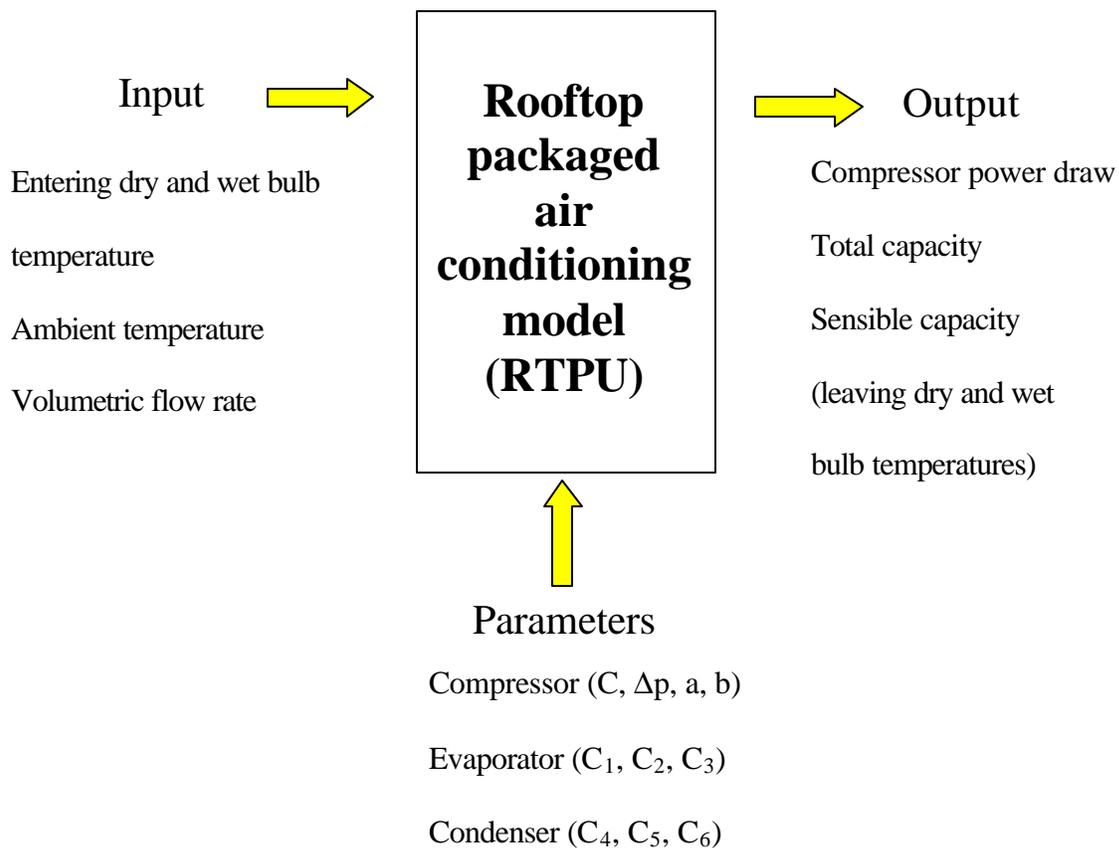


Figure 5.13: Input variables, output variables, and parameters used in rooftop packaged air conditioning model

Table 5.7 shows the errors in performance prediction for the four data sets (a) to (d) with the three parameter condenser model and with the more detailed six parameter model, where heat transfer coefficient for the desuperheating region is modeled.

Table 5.7: Performance prediction of the rooftop packaged air conditioning model

Error (fitting process)	(a)		(b)		(c)		(d)	
	normal/detailed	detailed/normal	normal/detailed	detailed/normal	normal/detailed	detailed/normal	normal/detailed	detailed/normal
Power draw [%]	2.6	1.6	1.5	1.4	2.0	1.4	2.5	1.7
Total capacity [%]	2.3	1.5	2.1	1.9	2.6	2.2	2.9	2.4
Sensible capacity [%]	1.2	0.9	3.2	3.1	1.4	1.1	2.5	2.1
Error (validation for whole catalog range)								
Power draw [%]	1.9	1.2	1.9	1.3	1.9	1.3	2.0	1.3
Total capacity [%]	2.0	1.6	1.9	1.5	1.9	1.5	2.0	1.5
Sensible capacity [%]	1.7	1.5	2.0	1.9	1.9	1.8	1.8	1.9

The mean weighted RMS errors for the predicted performance are all below 4 % in the fitting process. One sees that the rooftop packaged air conditioning model using the more detailed condenser model improves the performance prediction for the power draw prediction and the total capacity by about 0.5 % on average. The effect on the performance prediction of the sensible capacity is very small.

Figure 5.14 to Figure 5.16 show the performance prediction for the rooftop packaged air conditioning unit using the model parameters calculated with data set (a) and the simple three parameter condenser model. Shown are the eight data points that are used for fitting as well as the extrapolated data points, which cover the whole catalog range.

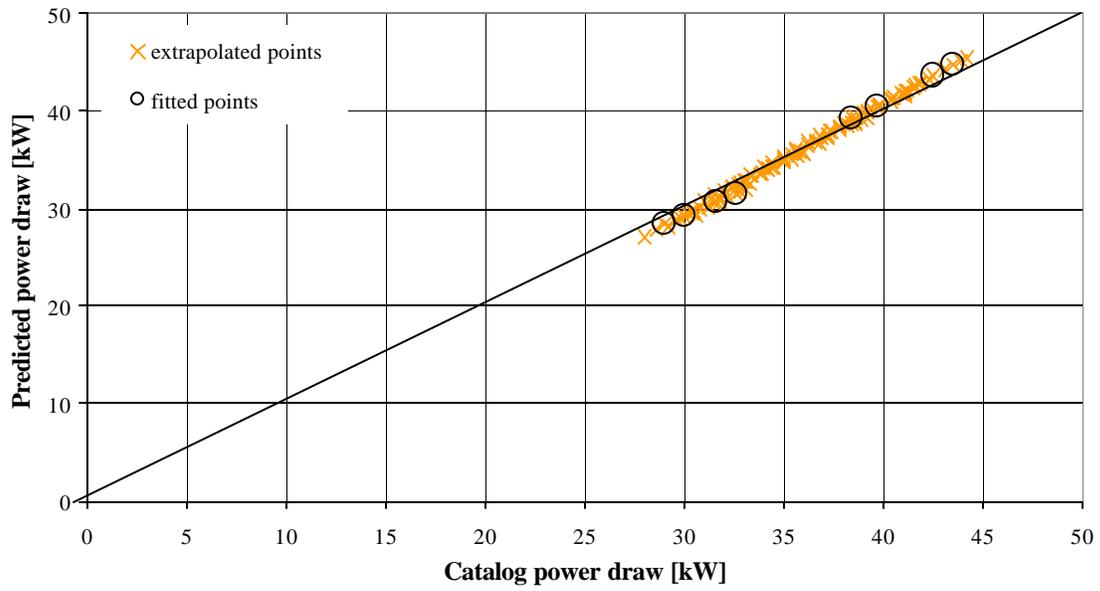


Figure 5.14: Predicted compressor power draw for model parameters calculated with data set (a)

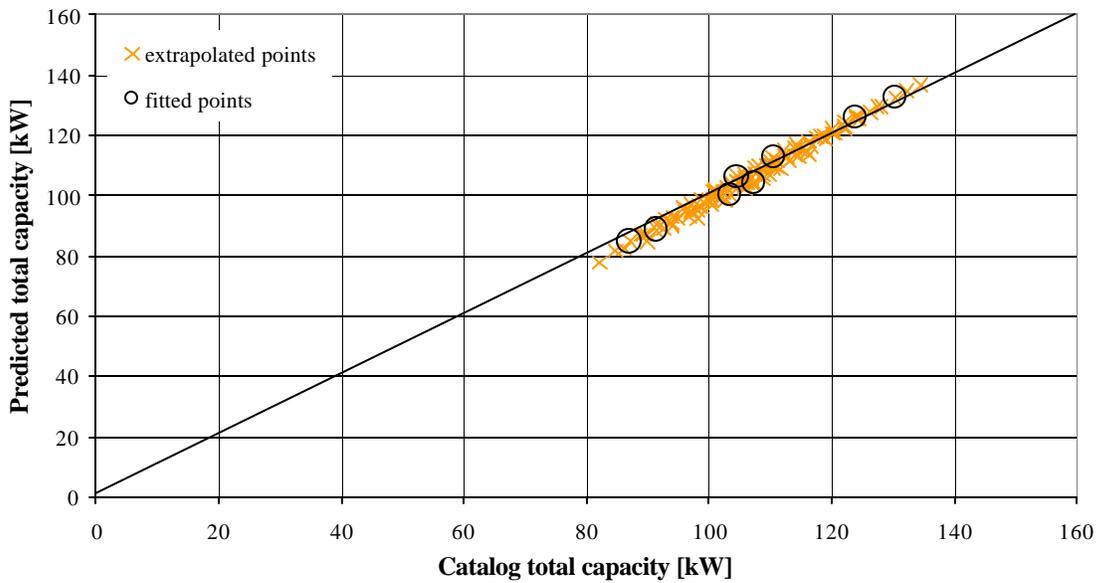


Figure 5.15: Predicted total capacity for model parameters calculated with data set (a)

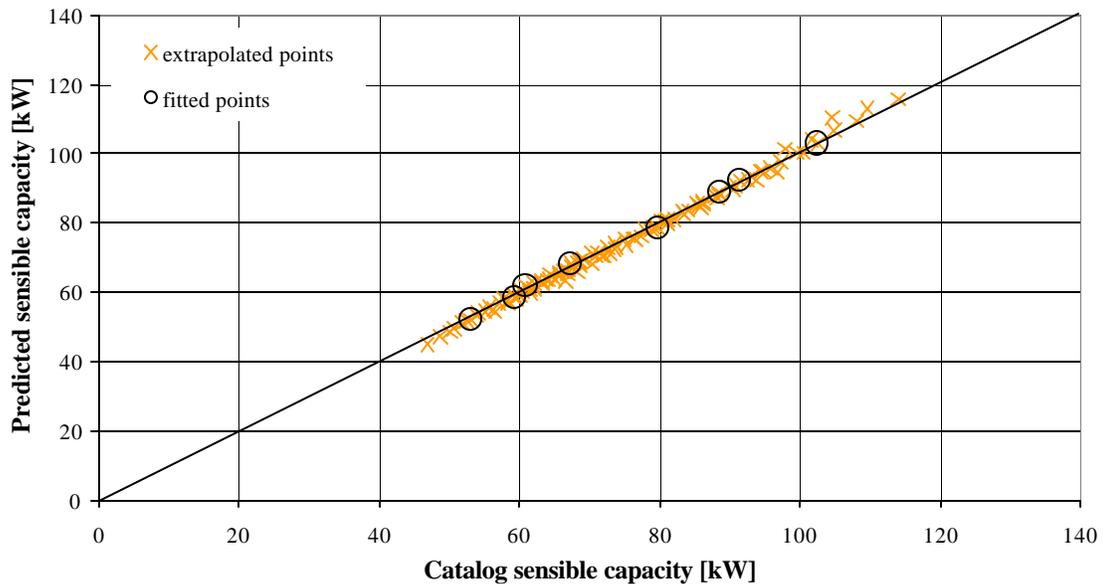


Figure 5.16: Predicted sensible capacity for model parameters calculated with data set (a)

Figure 5.14 to Figure 5.16 show that the extrapolation ability for the flow rate is in good agreement with the catalog performance. These results were obtained even though the evaporator model and the condenser model showed relative high errors in predicting the performance. This is due to the fact that the trends in performance prediction for the evaporator and condenser compensate each other, so that the error in overall performance prediction for the rooftop packaged air conditioning model is lower than for the single components.

Figure 5.17 to Figure 5.19 show the performance prediction with the model parameters calculated by using data set (d), which consists of eight data points covering a small range of ambient temperatures.

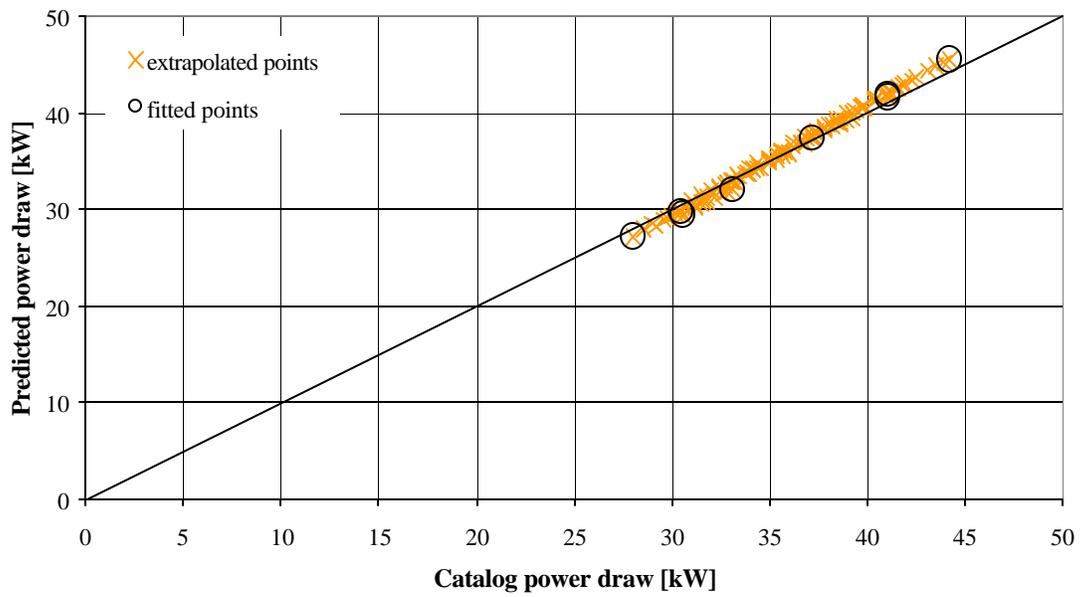


Figure 5.17: Predicted compressor power draw for model parameters calculated with data set (d)

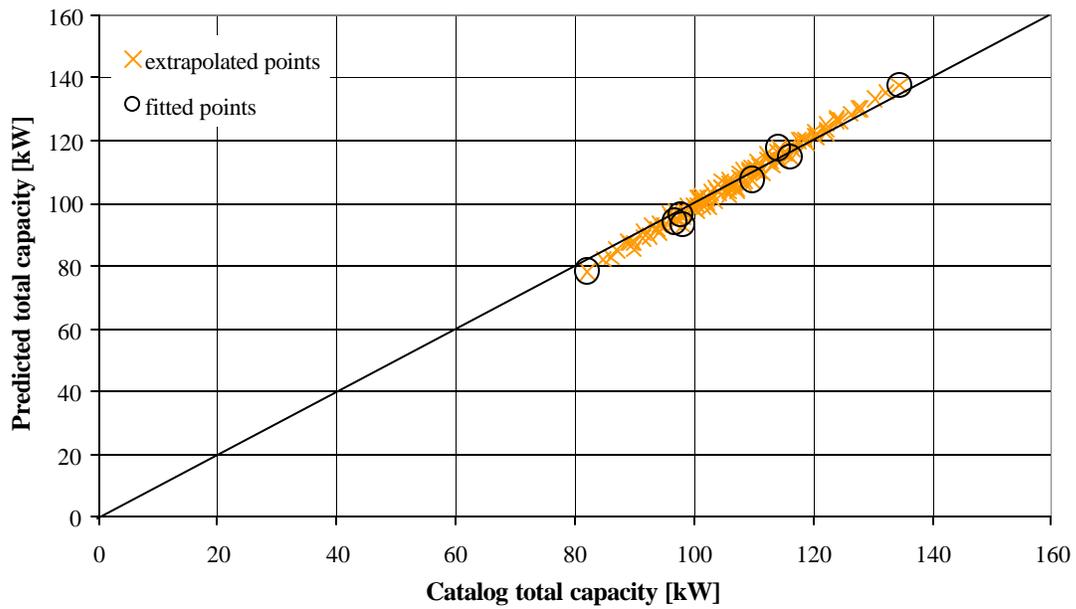


Figure 5.18: Predicted total capacity for model parameters calculated with data set (d)

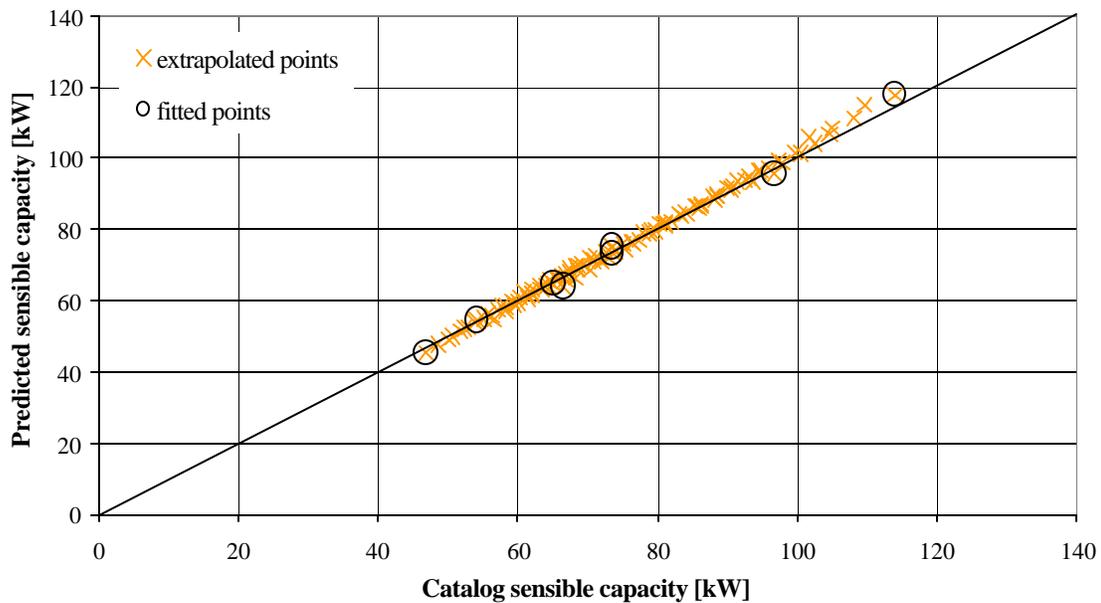


Figure 5.19: Predicted sensible capacity for model parameters calculated with data set (d)

Figure 5.17 to Figure 5.19 show that the extrapolation ability of the model for the ambient temperature is also in good agreement with the catalog data for the compressor power draw, for the total capacity and for the sensible capacity. The plots for the performance prediction for data set (c) and (d) are included in the appendix.

5.4.2. Effect of Number of Data Points

To determine the effect of the number of data points used in the fitting process three data sets were used. All three data sets covered the same range of values for the air entering conditions, the ambient temperature and the volume flow rate. The data set with only eight

data points (data set (d)) covered only the extreme values, whereas the other two data sets with twelve and fourteen data points also covered points within the range of the extreme values. Having determined the coefficients, the model was used to predict all operating points given in the catalog. The resulting errors are shown in the table below.

Table 5.8: Effect of number of data points on performance prediction

Error for fitting process	8 data points	12 data points	16 data points
Power draw [%]	2.53	2.56	2.59
Total capacity [%]	2.98	2.81	2.72
Sensible capacity [%]	2.46	2.03	1.89
Error for prediction catalog values			
Power draw [%]	1.99	1.97	1.93
Total capacity [%]	1.95	1.96	1.97
Sensible capacity [%]	1.77	1.74	1.83

It can be seen that the effect of the number of data points on the performance prediction is very small. Using only eight data points that are spread over the whole operating range in the fitting process already yields an accurate performance prediction. As a general rule data points for the fitting process should be chosen as a combination of high and low values for each cataloged operating variable.

5.4.3. Effect of Superheat

In the preceding section the model was validated for an idealized vapor compression cycle in which the refrigerant was assumed to leave the evaporator coil at saturated conditions. For real systems this, however, is not true since compressors need superheat to prevent them from liquid slugging. The amount of superheat that leads to the optimum performance prediction could be determined if heat exchanger parameters for condenser and evaporator, as well as the superheat were fitted at the same time. Optimization even for each heat exchanger separately, however, is very difficult, and in this case, seven parameters including the superheat needed to be fitted at the same time (three parameters for each heat exchanger and the superheat). Thus, an analysis was made in which the effect of different values of superheat on the performance prediction was examined for the three rooftop packaged units listed in Table 5.1.

Figure 5.20 to Figure 5.22 present the results of this analysis. The parameters for the system model were obtained by using data sets with eight data points, which cover the whole catalog data range for each rooftop packaged unit.

From Figure 5.20 and Figure 5.21 one sees that there is almost no effect of superheat on the performance prediction for the compressor power draw. The error in total capacity prediction decreases by about 0.2 to 0.3 % at high values of superheat for all three units.

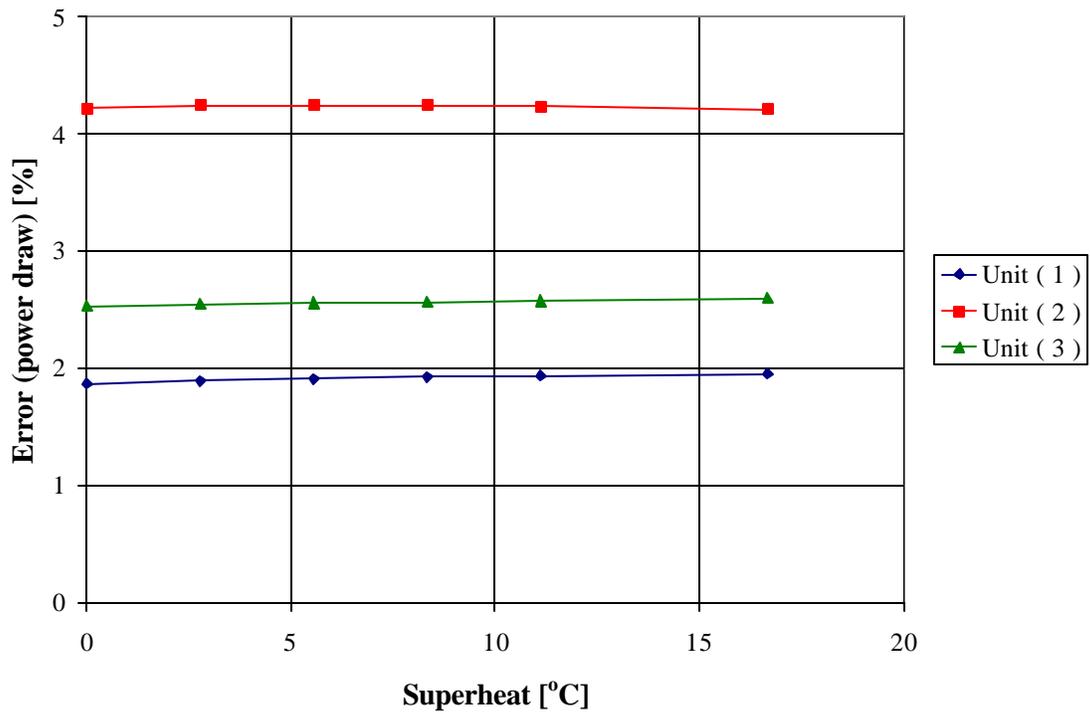


Figure 5.20: Effect of superheat on power prediction

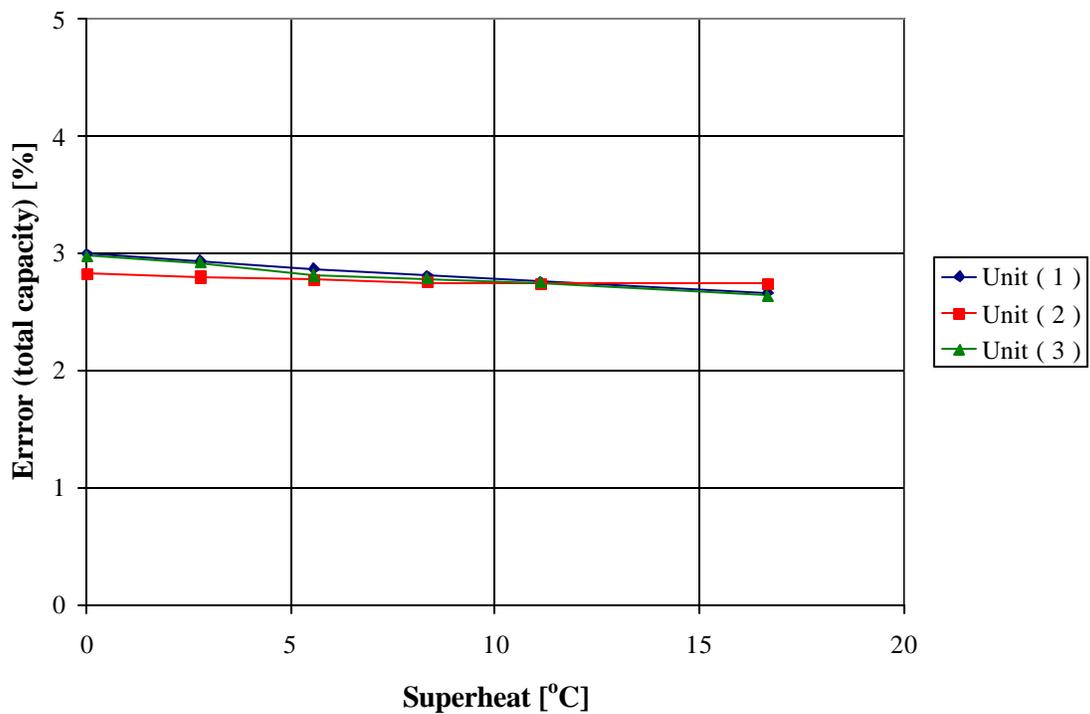


Figure 5.21: Effect of superheat on total capacity prediction

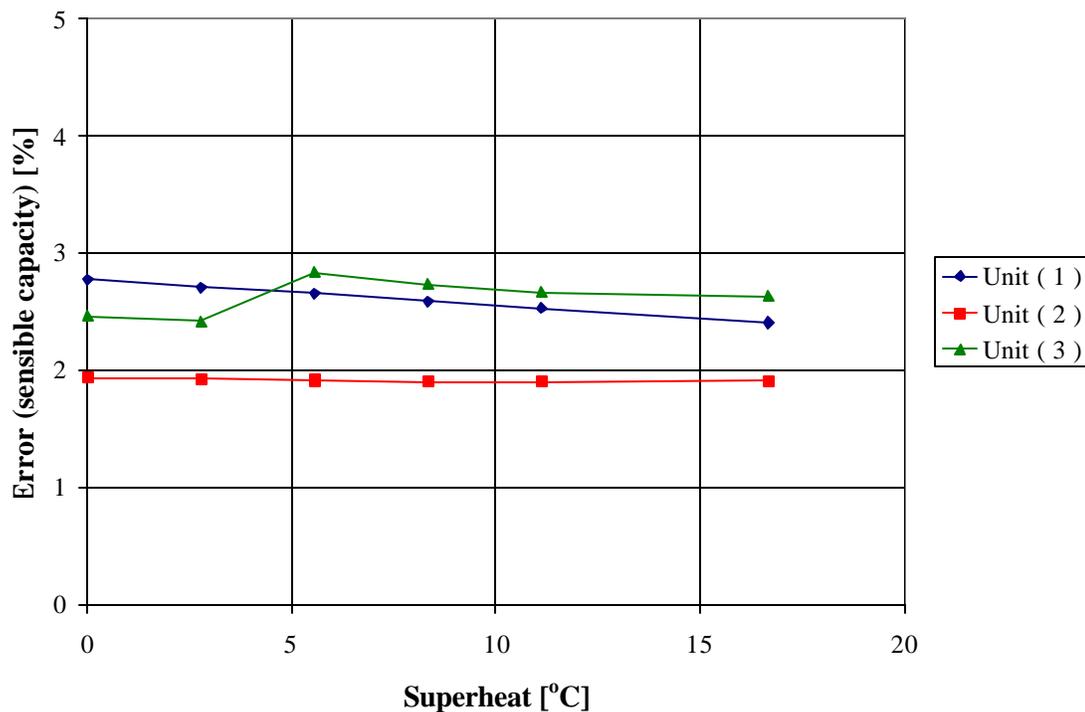


Figure 5.22: Effect of superheat on sensible capacity prediction

Figure 5.22 shows the effect of superheat on the sensible capacity prediction. The performance prediction for all three units is different. Performance prediction for unit (1) becomes better with increasing superheat, whereas the performance prediction for unit (2) is not affected by the amount of superheat. In contrast to unit (1) and (2), the performance prediction for unit (3) has a minimum at 2.8 °C (5 °F).

Generally speaking, components prediction can be improved by taking superheat into account. However, from the figures above it is not possible to make a general suggestion for the value of superheat that leads to best results in modeling rooftop packaged units. According to values found in the Trane Air Conditioning Manual (1979), the amount of superheat at the evaporator outlet is usually about 5 to 6 °C (10 °F), which would be a

realistic assumption for the vapor compression cycle. Assuming that there is no superheat is a valid assumption since the effect on performance prediction is very small as it is shown in this analysis.

5.4.4. Effect of Subcooling

In Section 5.4.1 the vapor compression cycle was modeled with the assumption that the refrigerant leaving the evaporator is saturated refrigerant. Section 5.4.3 takes superheating into account, which is more realistic assumption for the system. Both cases use the assumption that the refrigerant leaving the condenser is saturated liquid. However, in real systems the condensed liquid is often subcooled to increase the refrigeration capacity. The term subcooling describes the temperature difference between the condensing temperature and the temperature at which the refrigerant enters the thermostatic expansion device. The effect of subcooling at the condenser outlet on the performance prediction of the model was examined based on the assumption of 5.6 °C (10 °F) superheated refrigerant at the evaporator outlet. Thus, Figure 5.23 to Figure 5.25 were obtained.

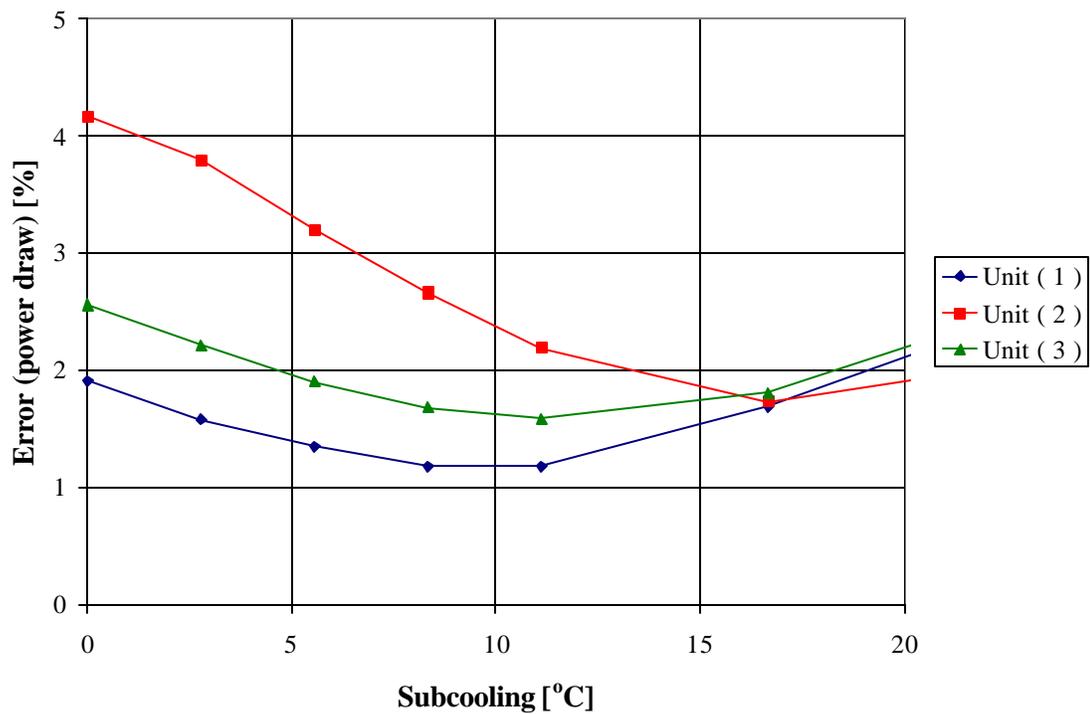


Figure 5.23: Effect of subcooling on power prediction

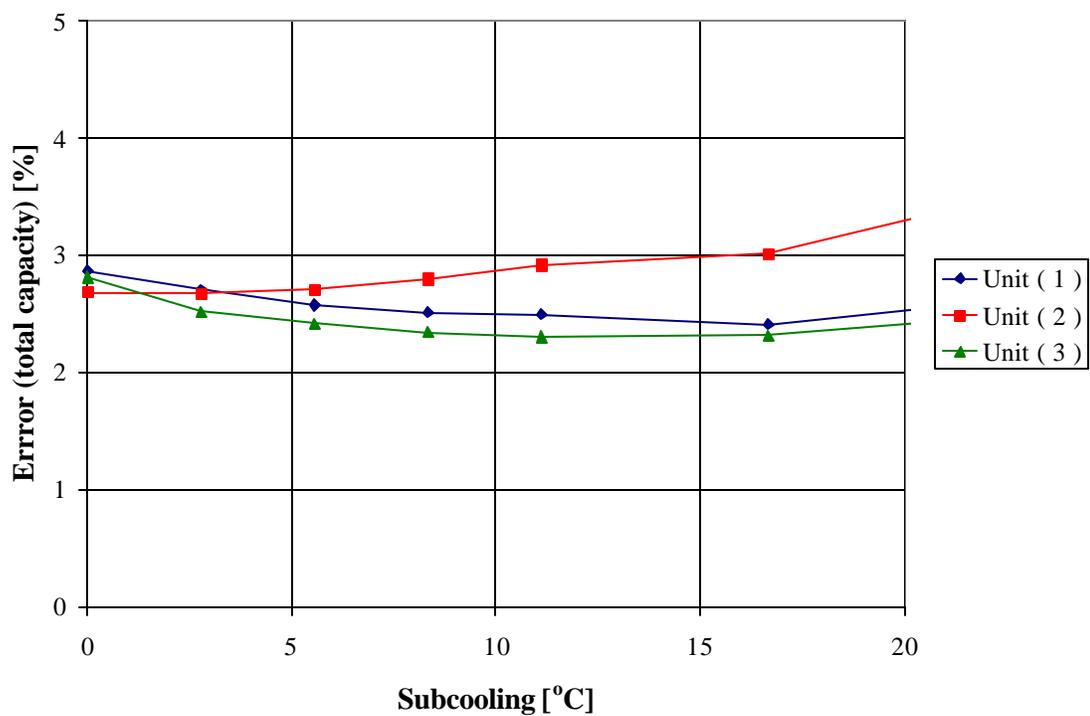


Figure 5.24: Effect of subcooling on total capacity prediction

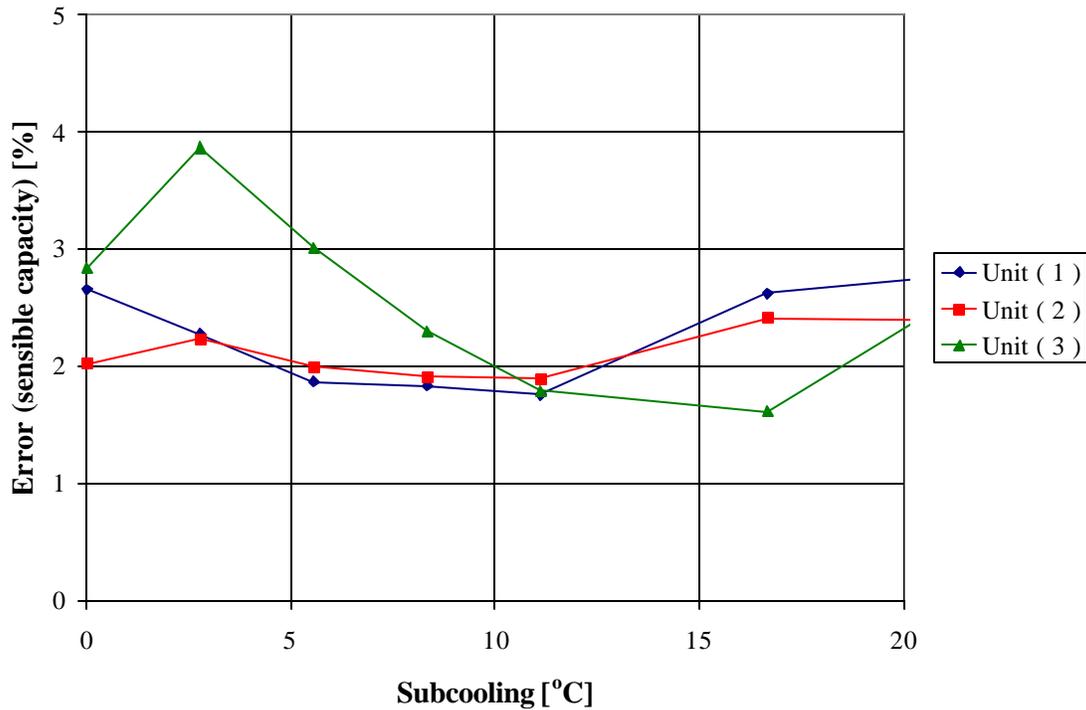


Figure 5.25: Effect of subcooling on sensible capacity prediction

For small values of subcooling, the performance prediction for the power draw, the total and sensible capacity is improved. The error in power draw prediction noticeably decreases for values up to about 11 °C (20 °F) subcooling, where the error for the performance prediction reaches a global minimum. The errors for total and sensible capacity prediction also tend to decrease as subcooling increases to about 11 °C (20 °F). This behavior, however, is by far not as distinctive as for the power prediction.

Generally speaking, the performance prediction of the model can be improved by introducing subcooling in comparison to the model, where the state of the refrigerant leaving the condenser is assumed to be saturated. An optimization for those values would be difficult due to the fact that the optimization problem is highly non-linear and multi dimensional. For

the model with superheat and subcooling at least six parameters (C_1 , C_3 , C_4 , C_6 , amount of subcooling and superheat) had to be determined simultaneously if the heat exchanger parameter C_2 and C_5 are set to a constant value. This problem can be avoided by using reasonable assumptions for the superheat and subcooling. Based on the results of the analysis shown above, for values of superheat of 5.6 °C (10 °F) and subcooling up to about 11 °C (20 °F) model performance prediction for the three rooftop packaged units was improved. A general suggestion for the amount of superheat and subcooling on the basis of this analysis, however, is difficult, since only three units were examined and all of them are from the same manufacturer. Thus, the assumption of 0°C superheat and subcooling was used for modeling the vapor compression cycle in rooftop packaged units.

5.4.5. Fans

In the previous section the performance of the vapor compression cycle in rooftop packaged air conditioning systems was modeled. The model allows the prediction of the sensible and total capacity as well as the compressor power draw. A significant amount of the total power of the unit, however, has to be supplied to the condenser and supply fans. The power draw of the condenser fans is available in the manufacturer's catalogs. The condensing fans operate at constant speed. Thus, the compressor power draw is for full load conditions a fixed number given in the catalog. In contrast to the power draw of the condenser fans, the power input to the supply fans changes according to the operating conditions. The power input to the supply fans is a function of the motor speed of the fan, the volumetric flow rate and the static pressure of the zone and is available as tabulated data in the catalog.

5.4.6. Heating

Often packaged units are not only used for space cooling but also for space heating. Generally, the heating section is equipped with an electric resistance heater or a gas fired furnace. The power draw of the electric resistance heater is given directly by the required heating demand. For the gas-fired furnace the manufacturers provide a steady-state furnace efficiency, so that the required gas input can be determined by dividing the required heating demand by the furnace efficiency.

5.4.7. Part load modeling

The model presented in the preceding sections of this chapter was evaluated for full load data only. However, air conditioning systems often operate at part load conditions. Thus, it is of interest to also have a part load model. Basically, using the model developed above allows controlling systematically the single components. During part load operation often a certain number of cylinders are unloaded or a complete compressor is turned off in larger equipment. Thus, part load performance can be modeled by introducing a function that describes the unloading behavior of the compressor. Part load performance data, however, is required to develop such a function and to validate the part load performance model. Part load modeling based on compressor unloading was not attempted in this project due to the lack of information on part load performance data available from the manufacturers.

5.4.8. Conclusions

A model for rooftop packaged air conditioning units was developed that is based on manufacturer's catalog data only. Thus, it can be easily used without any additional specific information about the system's components that need to be obtained from the manufacturer. The model uses a semi-mechanistic modeling technique. The system is modeled by using models for the single components, which are connected together. The models for these components are based on semi-empirical equations that describe the component's physical behavior. The model developed for rooftop packaged units can accurately predict the performance in the range of the fitted data and allows confident performance prediction to be made outside of the range of the fitted data. The errors in performance prediction for the three different rooftop packaged units modeled in this project were below 3.5 % compared with catalog data.

Chapter 6

Conclusions and Recommendations

6.1 Conclusions

In this study, a model for rooftop packaged air conditioning equipment was developed. For this type of equipment the model predicts the full load performance of this type of equipment such as power draw, total capacity, and sensible capacity for different operating conditions.

The rooftop packaged unit model is based on semi-mechanistic component models for each system's component, which are interconnected together to form the system model. Using manufacturer performance data for each specific component, component models were validated and extrapolation ability was checked.

The compressor is modeled by using a semi-empirical model for reciprocating and scroll compressors containing four parameters that are obtained in a fitting process to compressor performance data. The models were found to generally predict the compressor performance within an accuracy of 5 % to manufacturer's catalog data.

The heat exchanger models for the condenser and evaporator are based on the effectiveness-NTU method. Instead of assuming a constant UA-product, fundamental heat transfer relations are used to calculate the UA-product taking the different operating conditions into account. Each heat exchanger model contains three characteristic parameters that are obtained in the fitting process to performance data. Heat exchanger models can be used without specific knowledge of the heat exchanger geometry, since the geometric terms

are included in the characteristic parameters. The extrapolation ability of the models is very robust. Fitting the characteristic parameters with only a portion of the catalog data set, performance data could be predicted accurately over the entire catalog data range. The condenser model predicted the performance with an accuracy of 3.9 %, whereas the evaporator model predicted the performance with an accuracy of 1.8 % for the whole range of operating points given in the catalog.

The parameter fitting process for the heat exchanger models is difficult and depends strongly on the first guess for the characteristic parameters due to the highly non-linear optimization problem. To overcome this problem, heat exchanger parameters C_2 and C_5 can be set to a fixed value of 0.5.

The effect on system's performance prediction was examined using a more detailed condenser model in which the desuperheating region in the condenser is separately modeled. The performance prediction for the condenser could be improved up to about 4 %, which resulted in an improvement for the system's performance prediction of about 0.5 %. Since this is only a relatively small improvement, the simpler condenser model with only three heat exchanger parameters was used for system modeling.

The single components used for modeling the system are generally applicable to different refrigerants. Thus, the model should also allow the performance prediction for refrigerants other than R-22. This, however, needs to be validated by modeling rooftop packaged units operating with a different refrigerant type.

In order to use the system model for performance prediction, ten characteristic parameters have to be determined, which is a small number compared to detailed models and

consequently requires fewer data points in the fitting process. The need for knowledge of the system specific parameters such as heat exchanger geometry and compressor geometry was minimized to allow an easy use of the model based on manufacturer catalog information. For all three rooftop packaged units modeled in this project, the model performance prediction was within an accuracy of 3.3 % to catalog data. The error for the power prediction was in the range of 1.6 to 3.3 %. The performance prediction of the total capacity was in the range of 2.0 to 2.3 % for the three rooftop packaged units, whereas the error for the sensible capacity was in the range from 1.8 to 3.2 %. The extrapolation ability for the complete system is in good agreement with the catalog data and allows confident performance prediction outside of the range of the fitted data.

The selection of catalog data points used to fit the characteristic parameters for the system model is critical. These data points should be chosen as combinations of high and low cataloged values for each of the operating point parameters such as flow rate and temperatures. Thus, having four operating point parameters 16 data points should be used at minimum to allow confident extrapolation for all four operating parameters. Using more data points within the range of low and high values does not significantly improve model's performance prediction.

The model for the rooftop packaged air conditioning is based on an idealized vapor compression cycle. The effect of superheat and subcooling was examined. For the three rooftop packaged units used for model validation, the effect of superheat on performance turned out to be negligible. In contrast, the amount of subcooling at the condenser considerably affects the performance prediction. A general suggestion for a certain amount of

subcooling that generally leads to best performance prediction of the model, however, can not be made based on the analysis done in this project. Thus, for system modeling it was assumed that there is no superheat at the evaporator outlet and no subcooling at the condenser outlet.

The model was developed with the Engineering Equation Solver (EES). A user-friendly software package was developed that can be used to determine the model parameters based on manufacturer's catalog information for the rooftop packaged unit and the compressor performance (Appendix C to J). Having these parameters determined, the program calculates the rooftop packaged unit's performance for different operating conditions such as supply flow rates, ambient temperatures and air entering temperatures.

6.2 Future Work

The software program developed in this study was developed with the Engineering Equation Solver. The EES program can be called by TRNSYS. However, simulation speed is very slow. Writing a TRNSYS type in Fortran for the rooftop packaged model would considerably increase simulation speed.

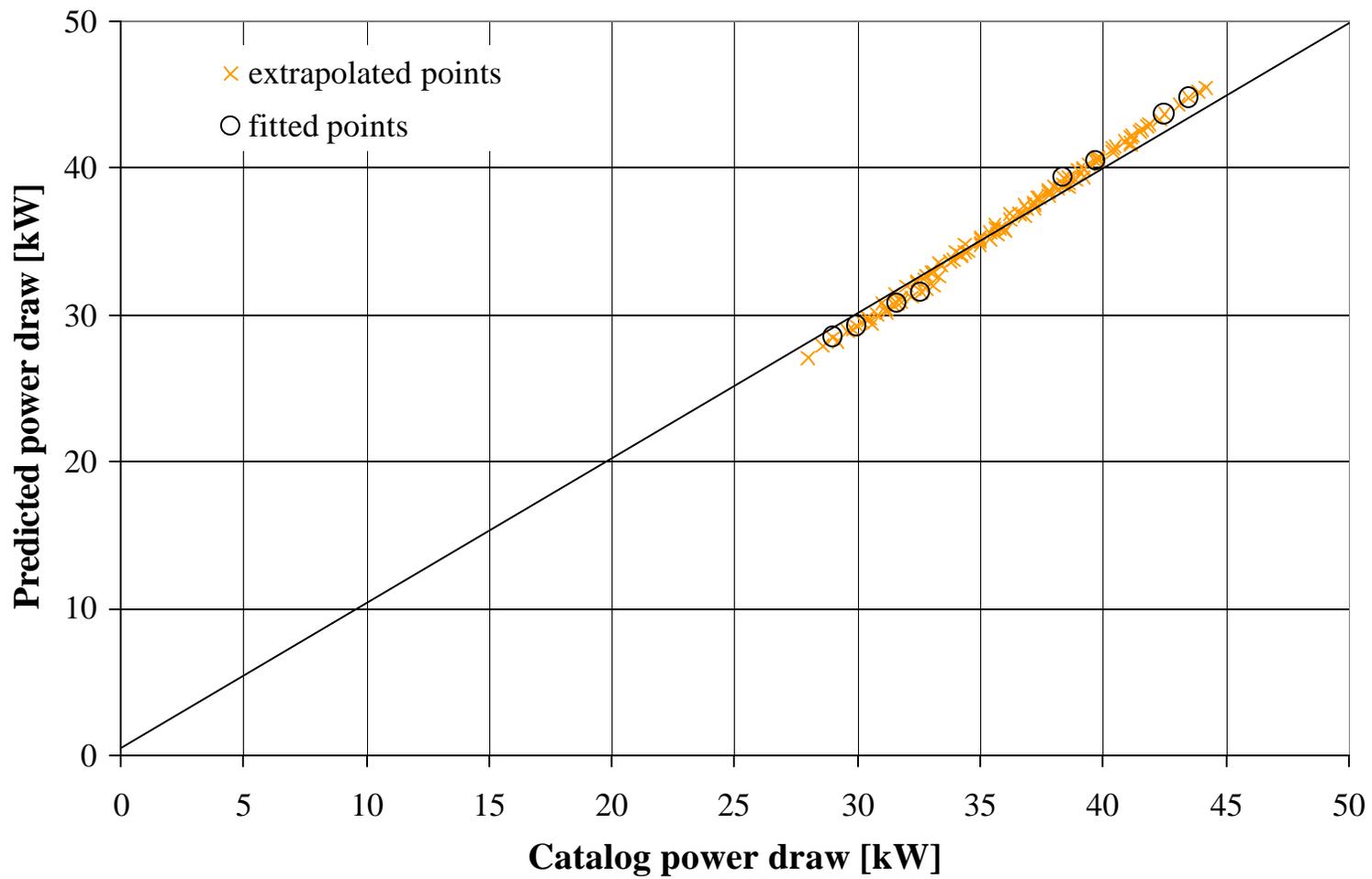
In this study the performance was modeled for full load operation of rooftop packaged units. However most of the time air conditioning systems operate under part load conditions. Thus, it is necessary to develop a part load model. The model presented in this work allows implementing control mechanisms for part load operation such as cylinder

unloading, since each component is modeled separately. For this purpose part load performance data and information on the control mechanisms have to be available from manufacturers, which is difficult to get. A simpler part load model could be developed in the form of a curve fit that uses the full load performance data and corrects those values for part load conditions.

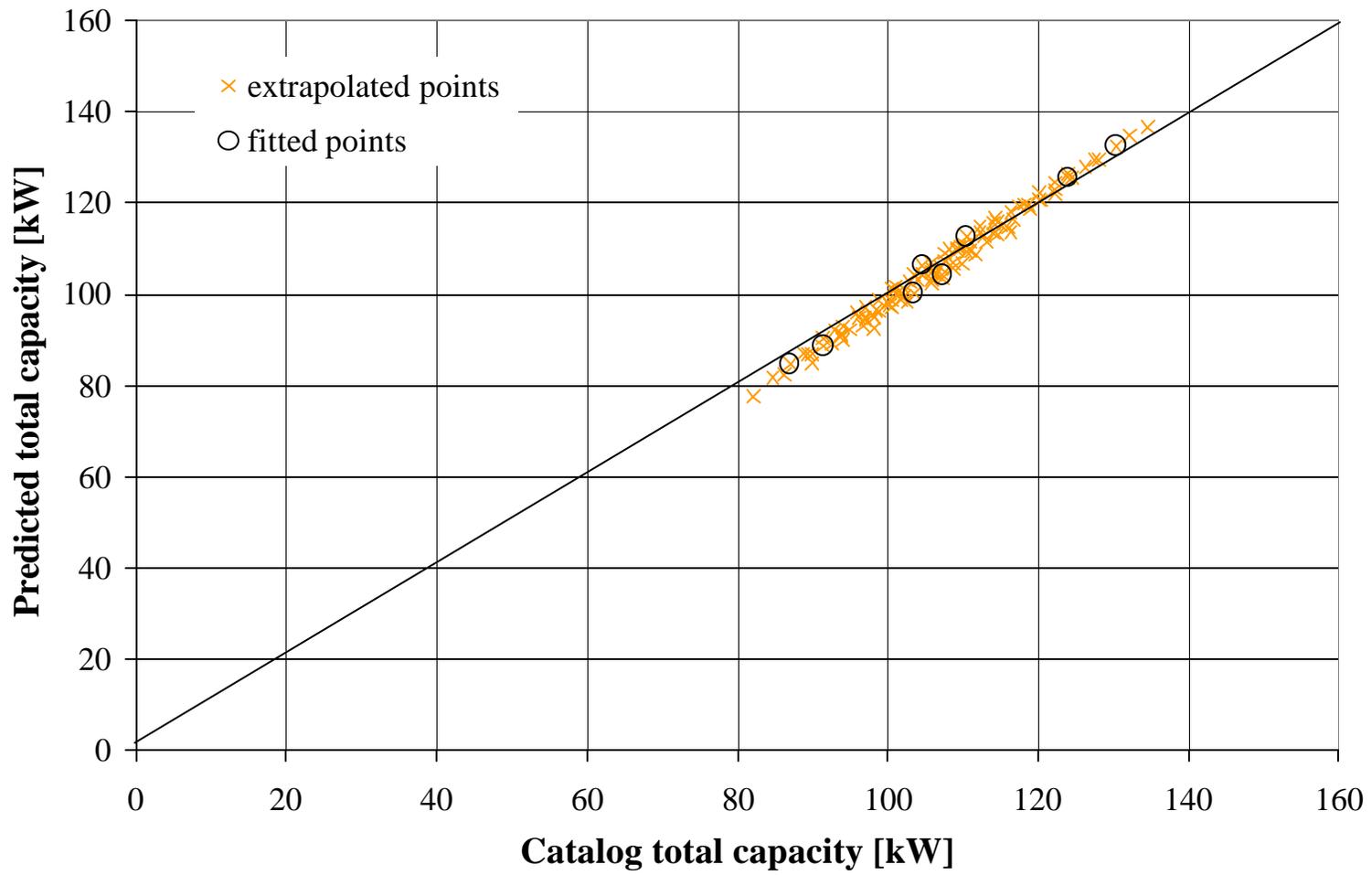
Appendix

Appendix A: Model Results for Rooftop Packaged Unit 50 EJ 034

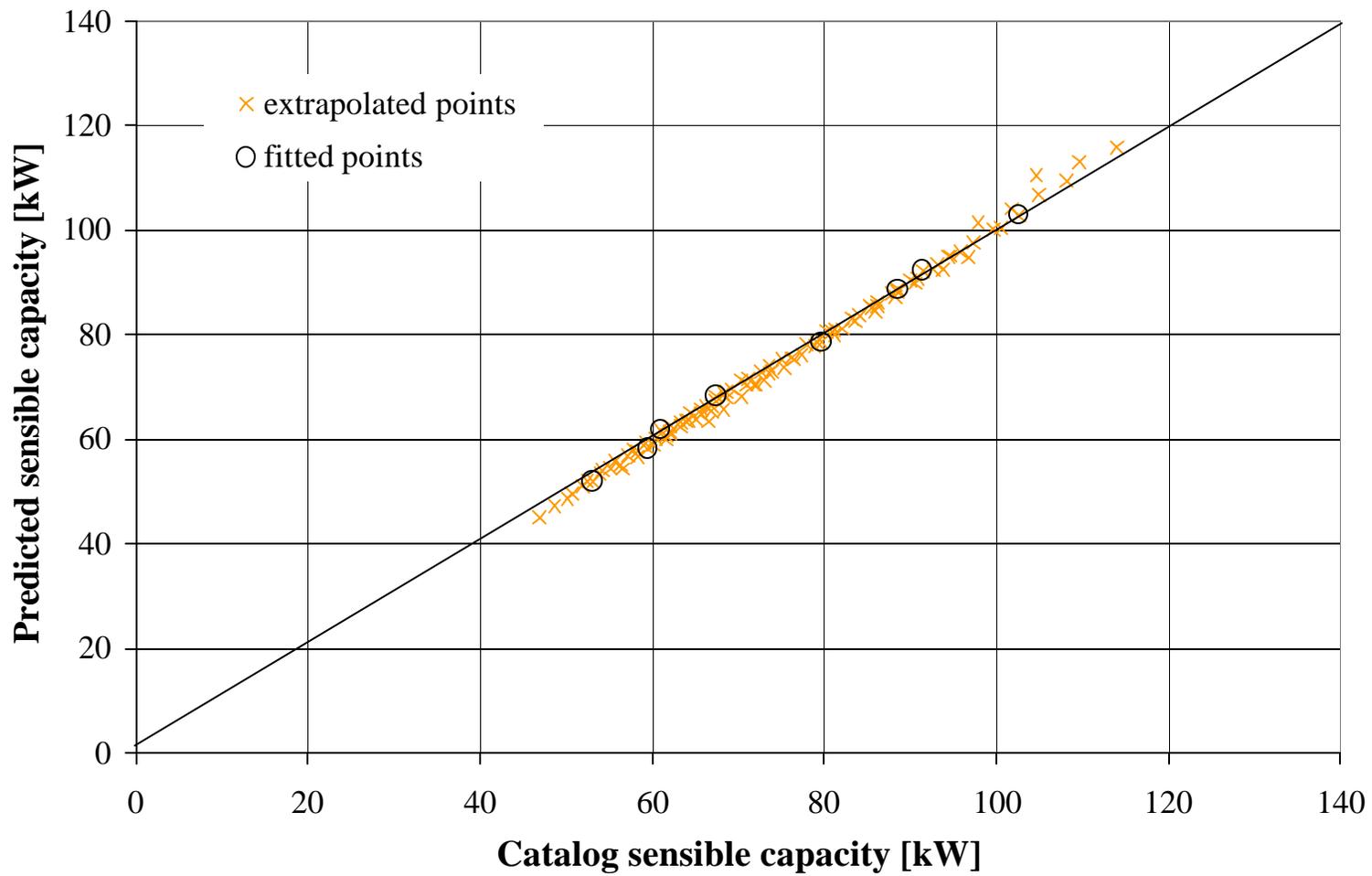
	Data set			
	(a)	(b)	(c)	(d)
C_1	1536	1695	1613	1650
C_2	0.5	0.5	0.5	0.5
C_3	3588	3246	3599	3607
Error (sensible) [%]	0.6	4.4	3.3	4.0
Error (total) [%]	1.5	5.2	5.8	5.8
C_4	5549	6203	5783	5581
C_5	0.5	0.5	0.5	0.5
C_6	31893	36315	33304	32051
Error [%]	9.6	4.8	7.4	9.5
Validation for 8 fitting points				
Error (power) [%]	2.6	1.5	2.0	2.5
Error (total capacity) [%]	2.3	2.1	2.6	3.0
Error (sensible capacity) [%]	1.2	3.2	1.4	2.5
Validation in catalog range				
Error (power) [%]	1.87	1.88	1.92	1.99
Error (total capacity) [%]	2.03	1.89	1.94	1.95
Error (sensible capacity) [%]	1.68	1.95	1.92	1.77



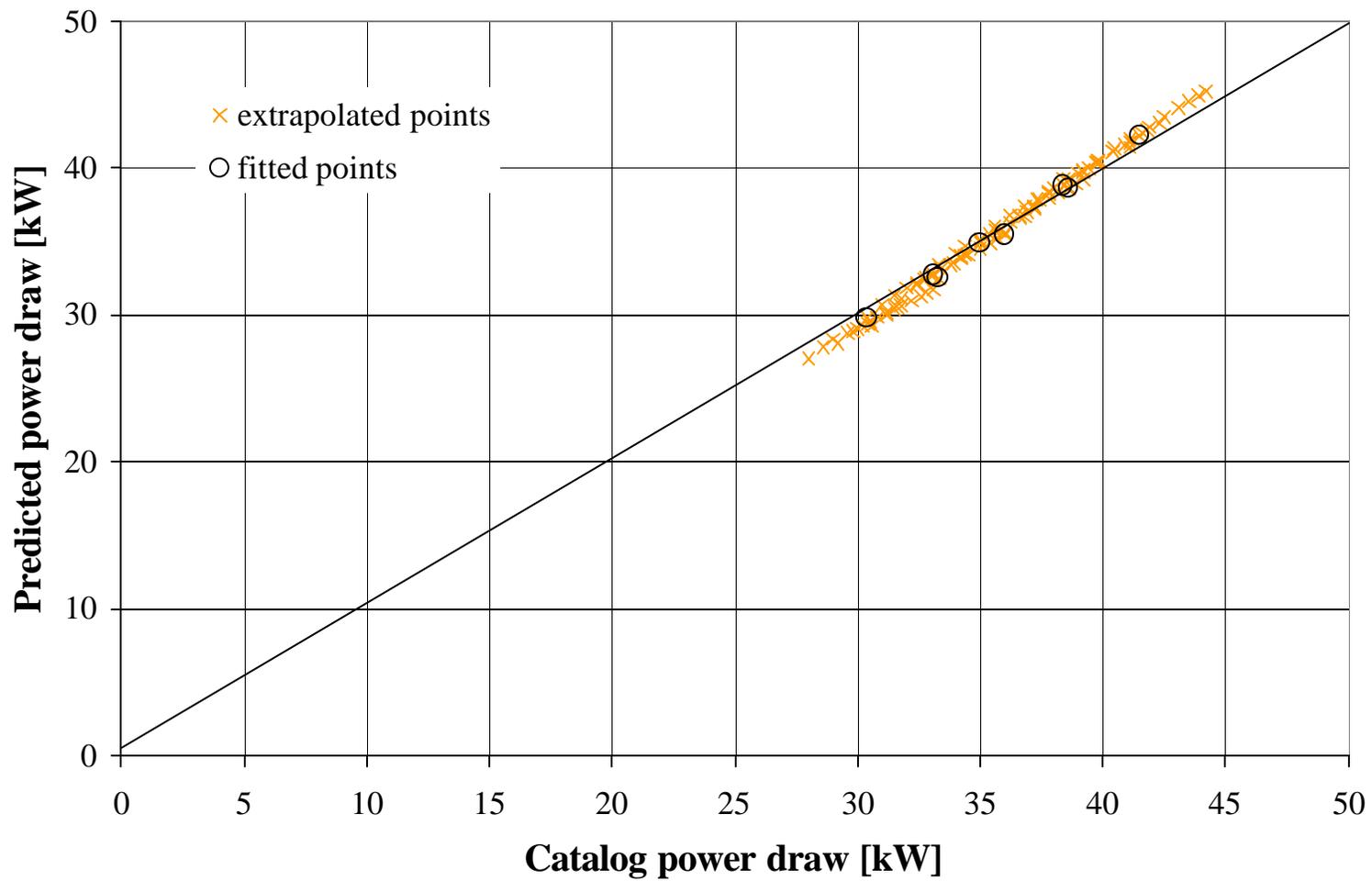
Model validation for data set (a)



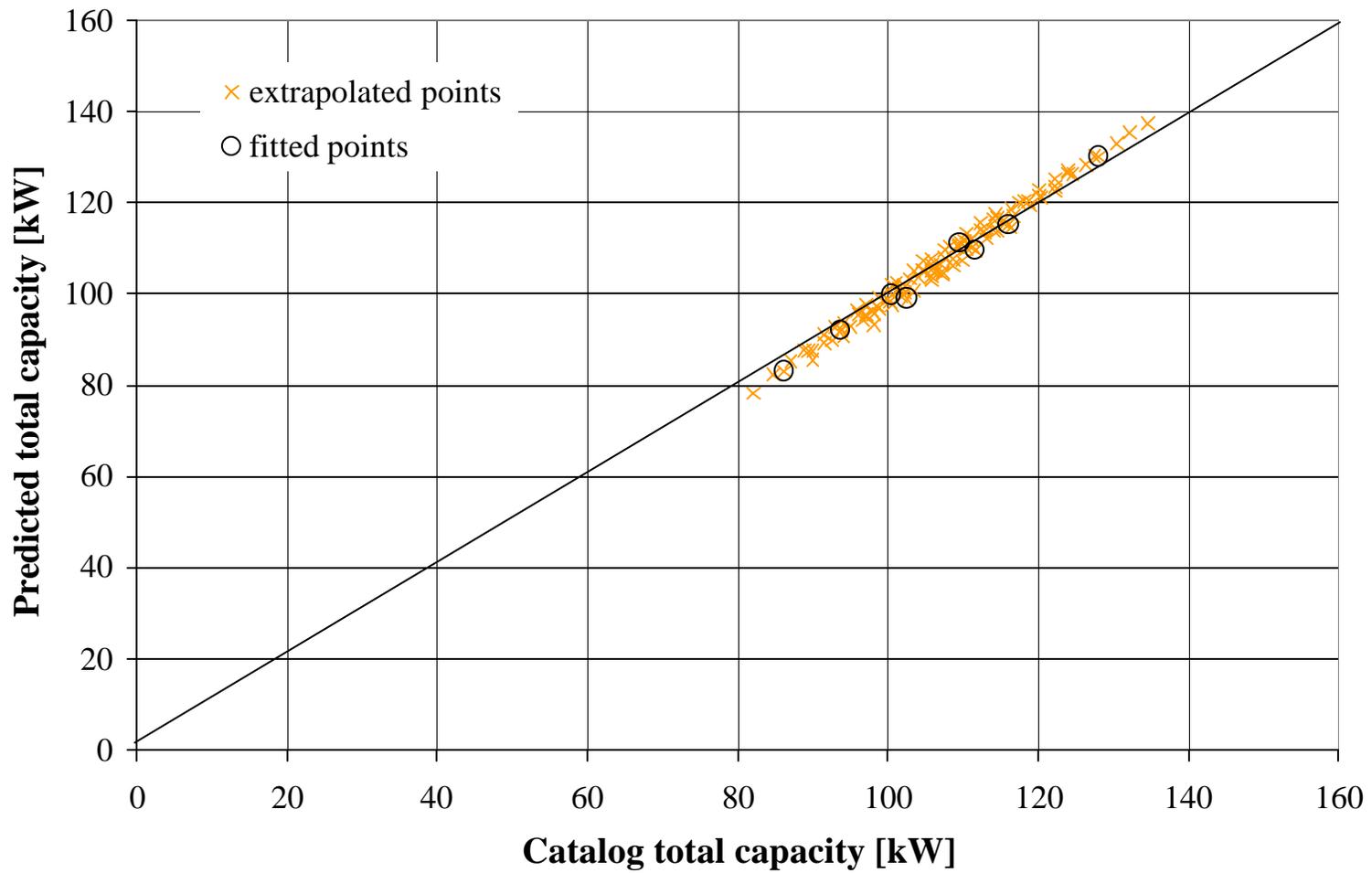
Model validation for data set (a)



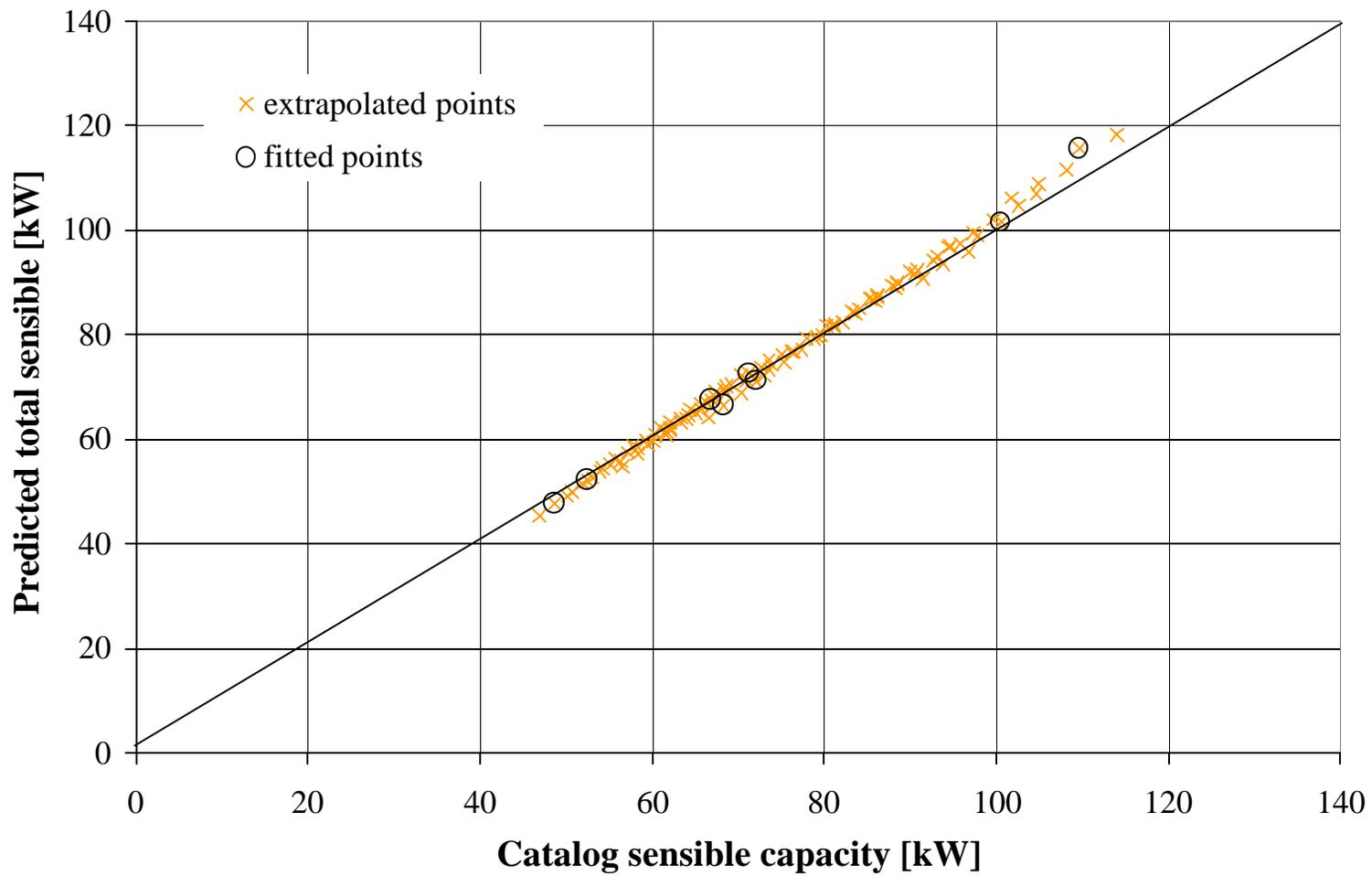
Model validation for data set (a)



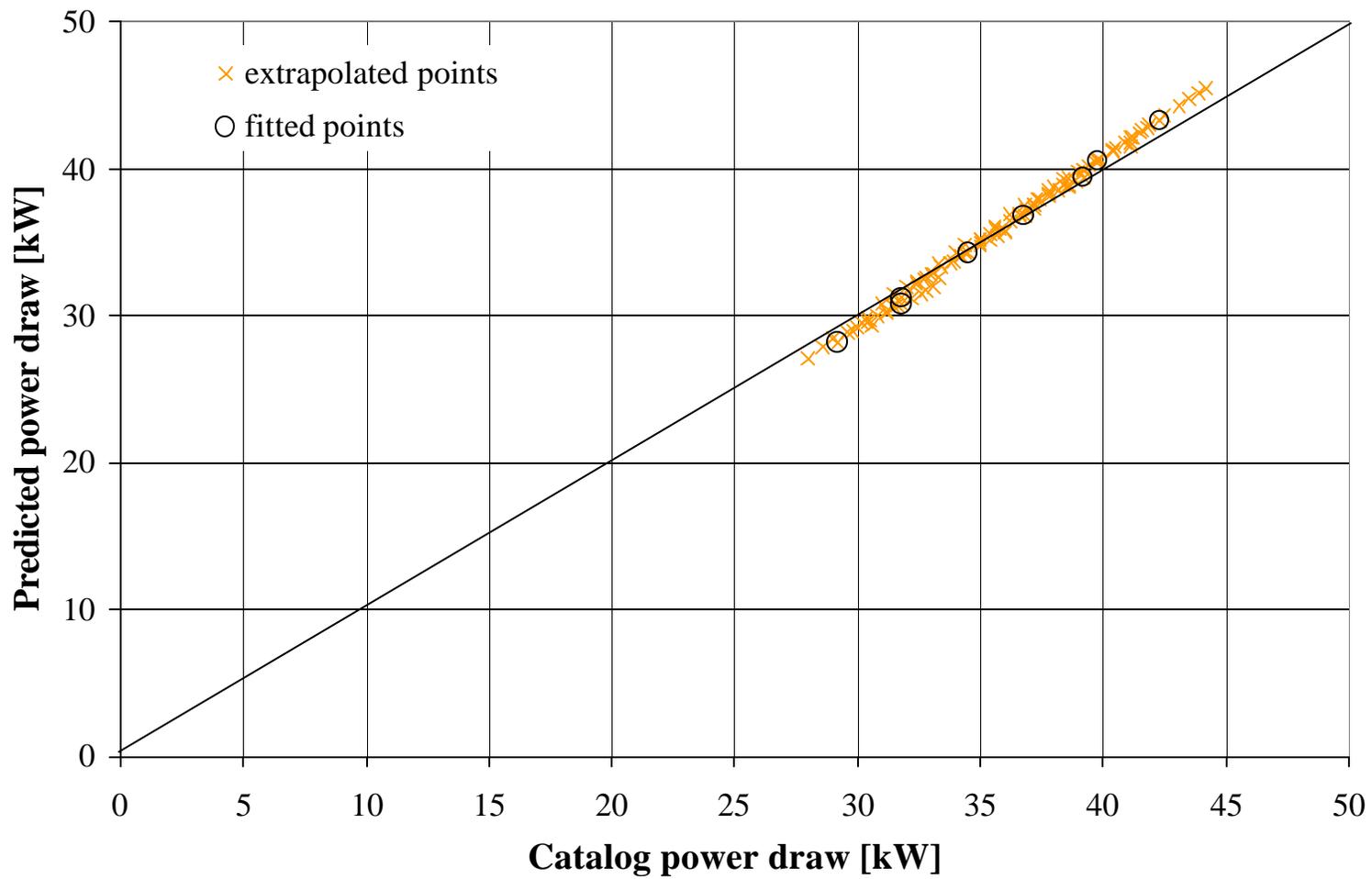
Model validation for data set (b)



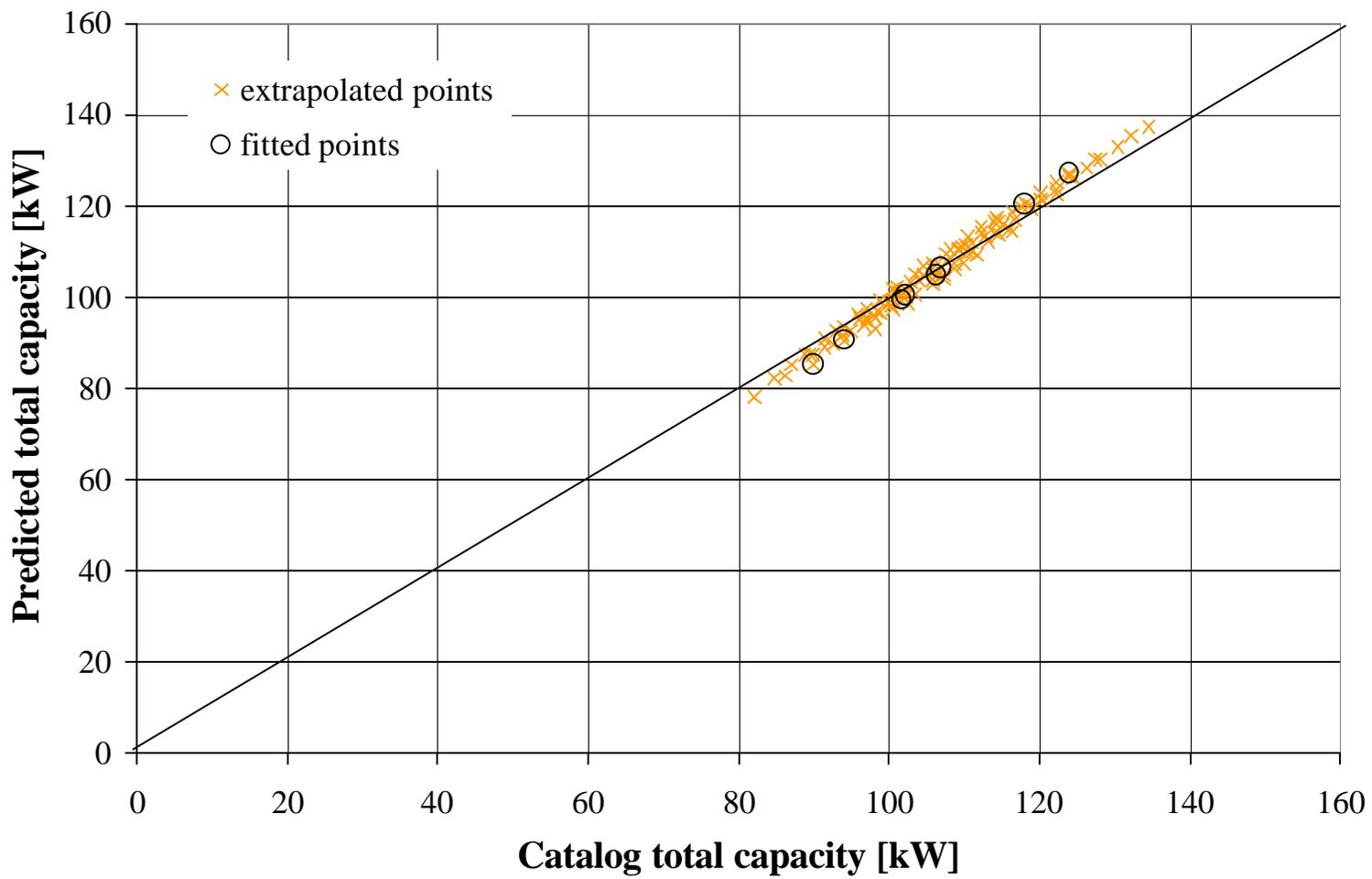
Model validation for data set (b)



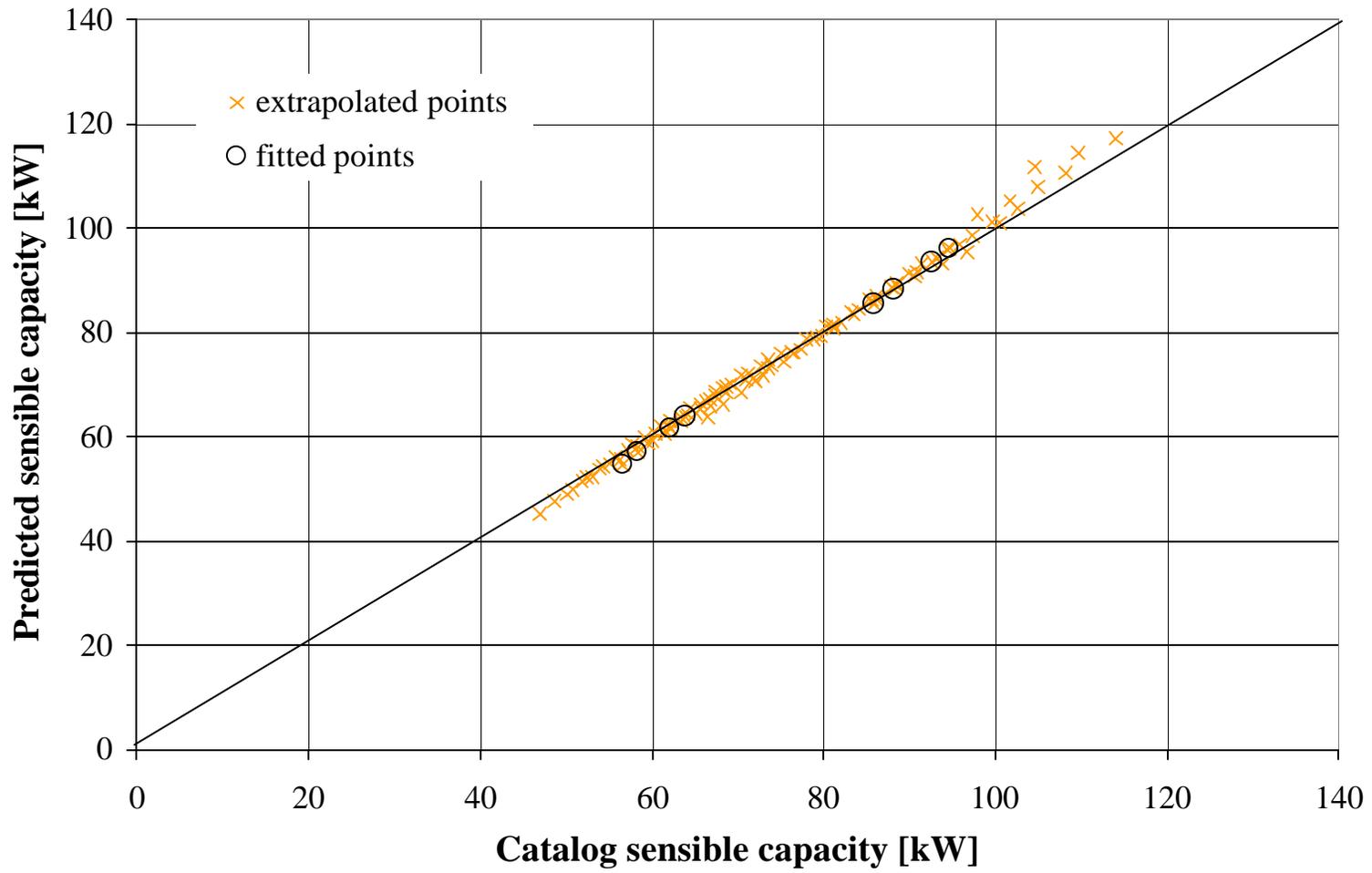
Model validation for data set (b)



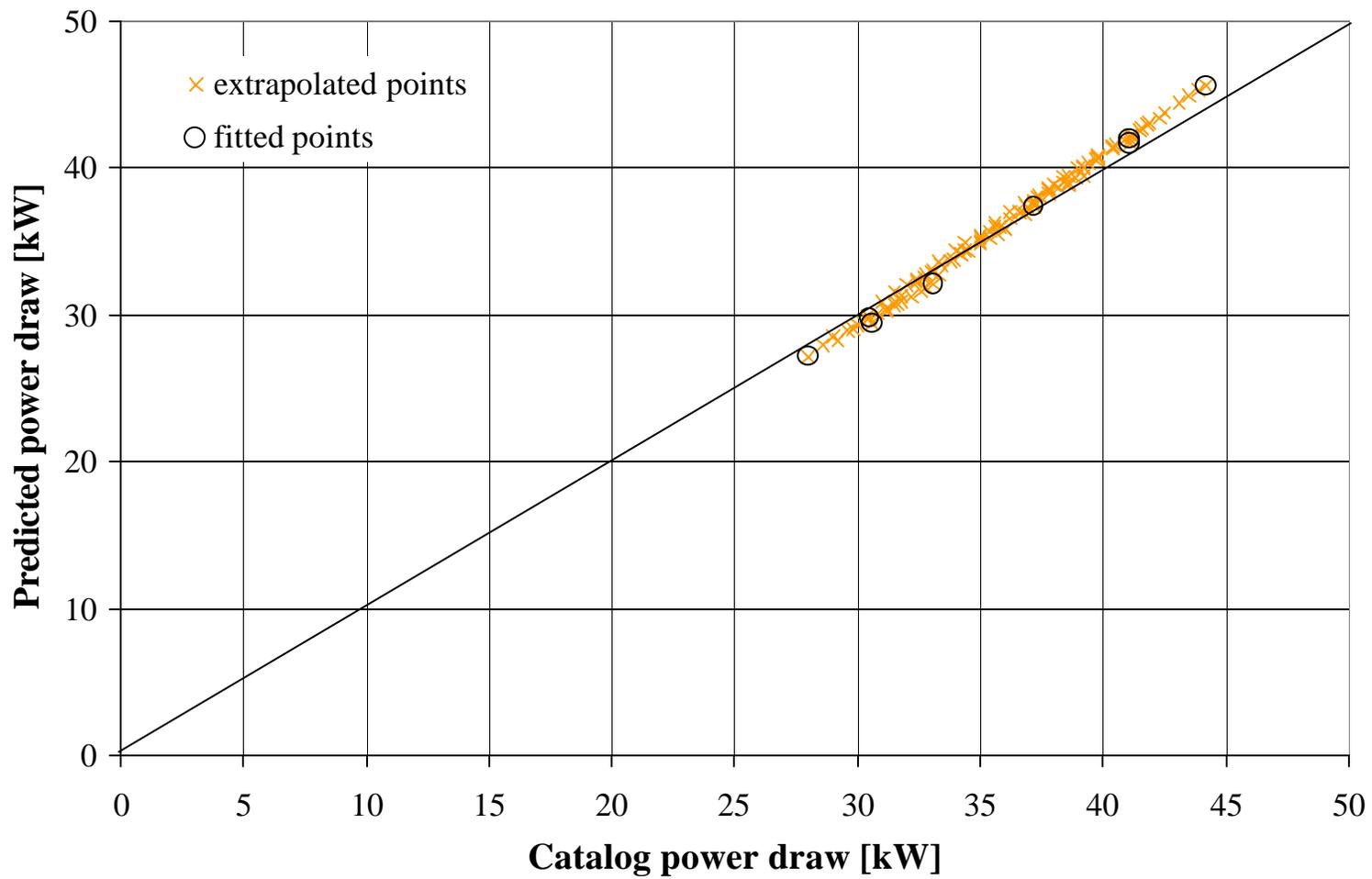
Model validation for data set (c)



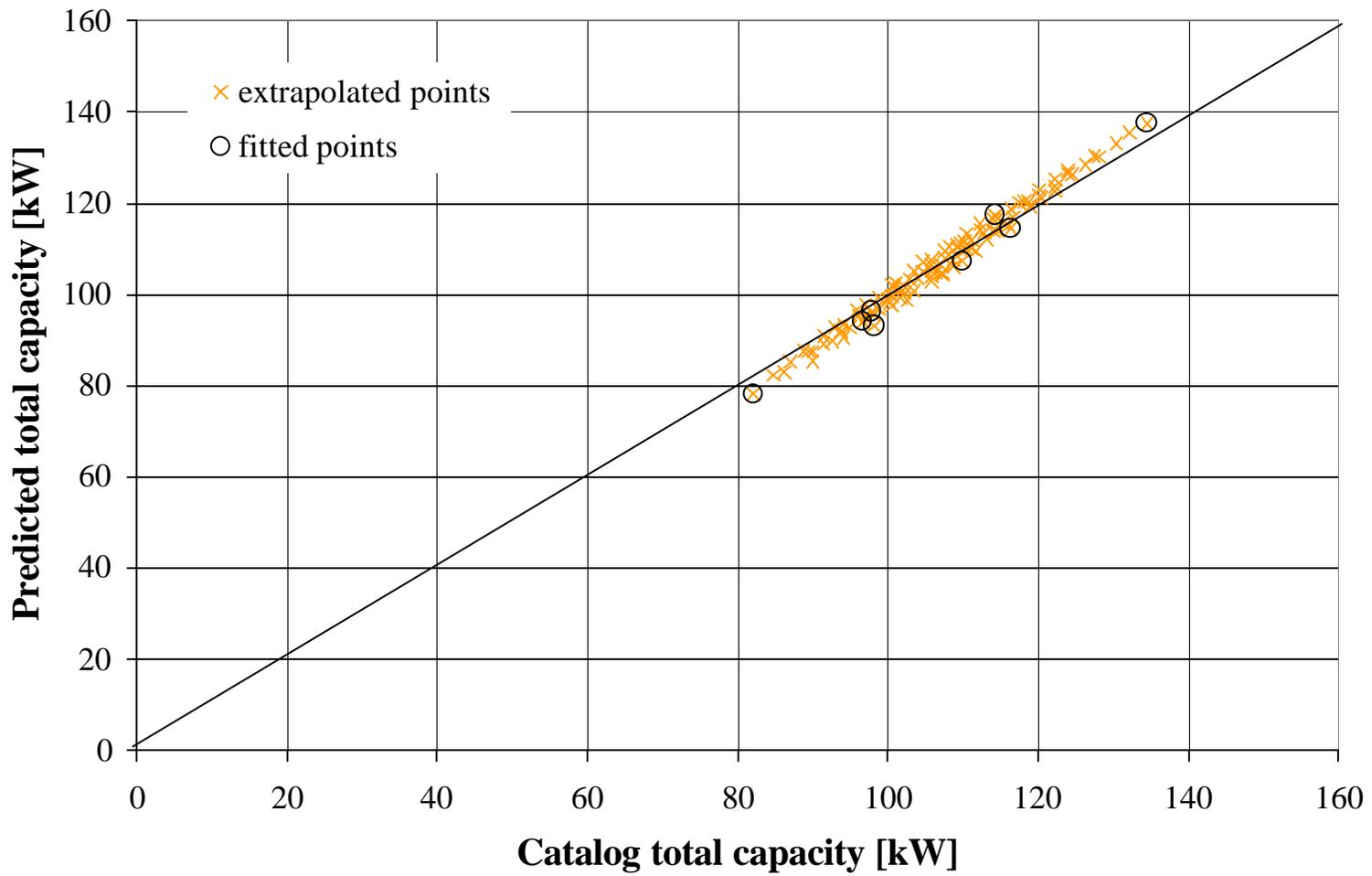
Model validation for data set (c)



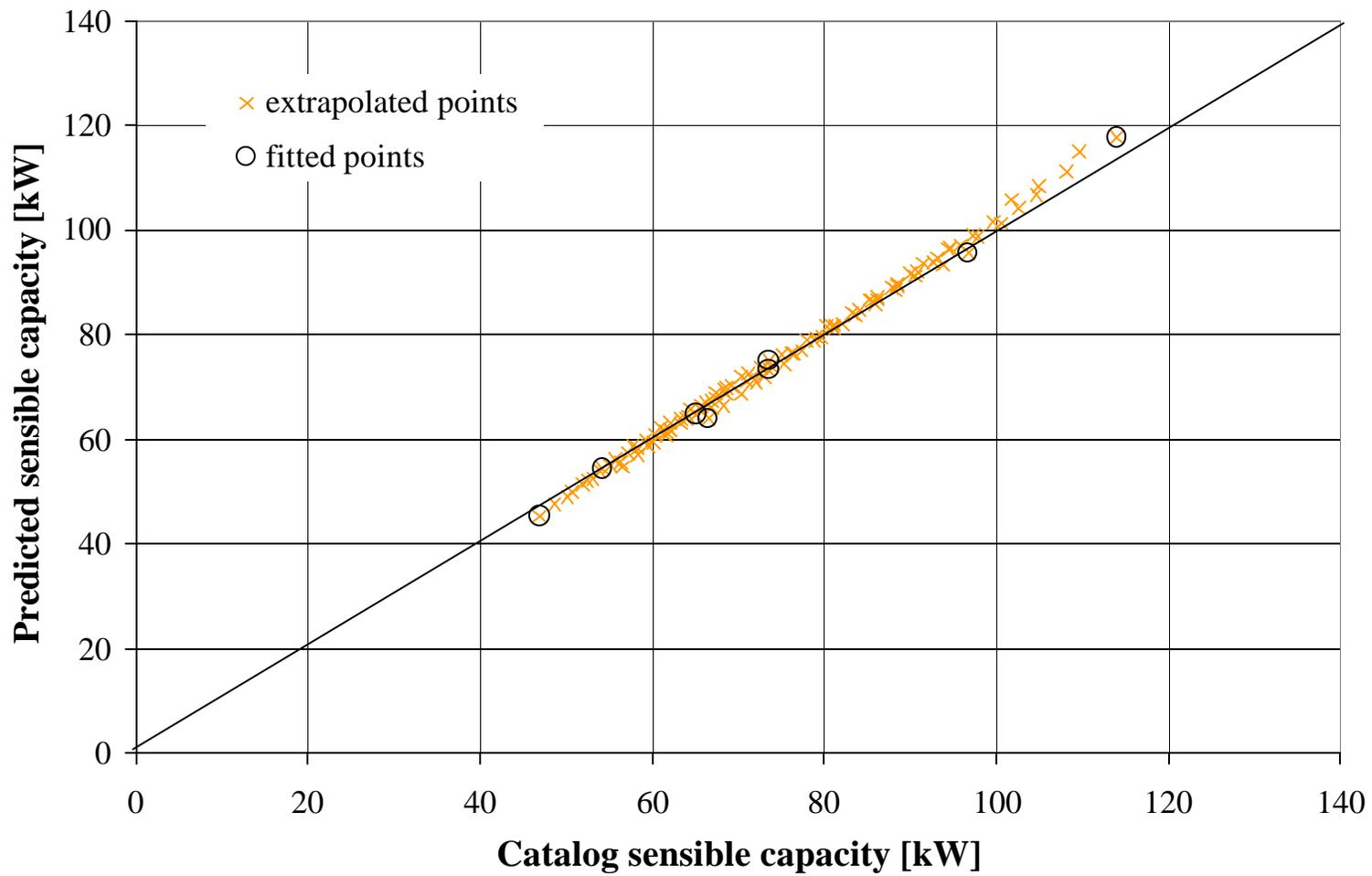
Model validation for data set (c)



Model validation for data set (d)



Model validation for data set (d)

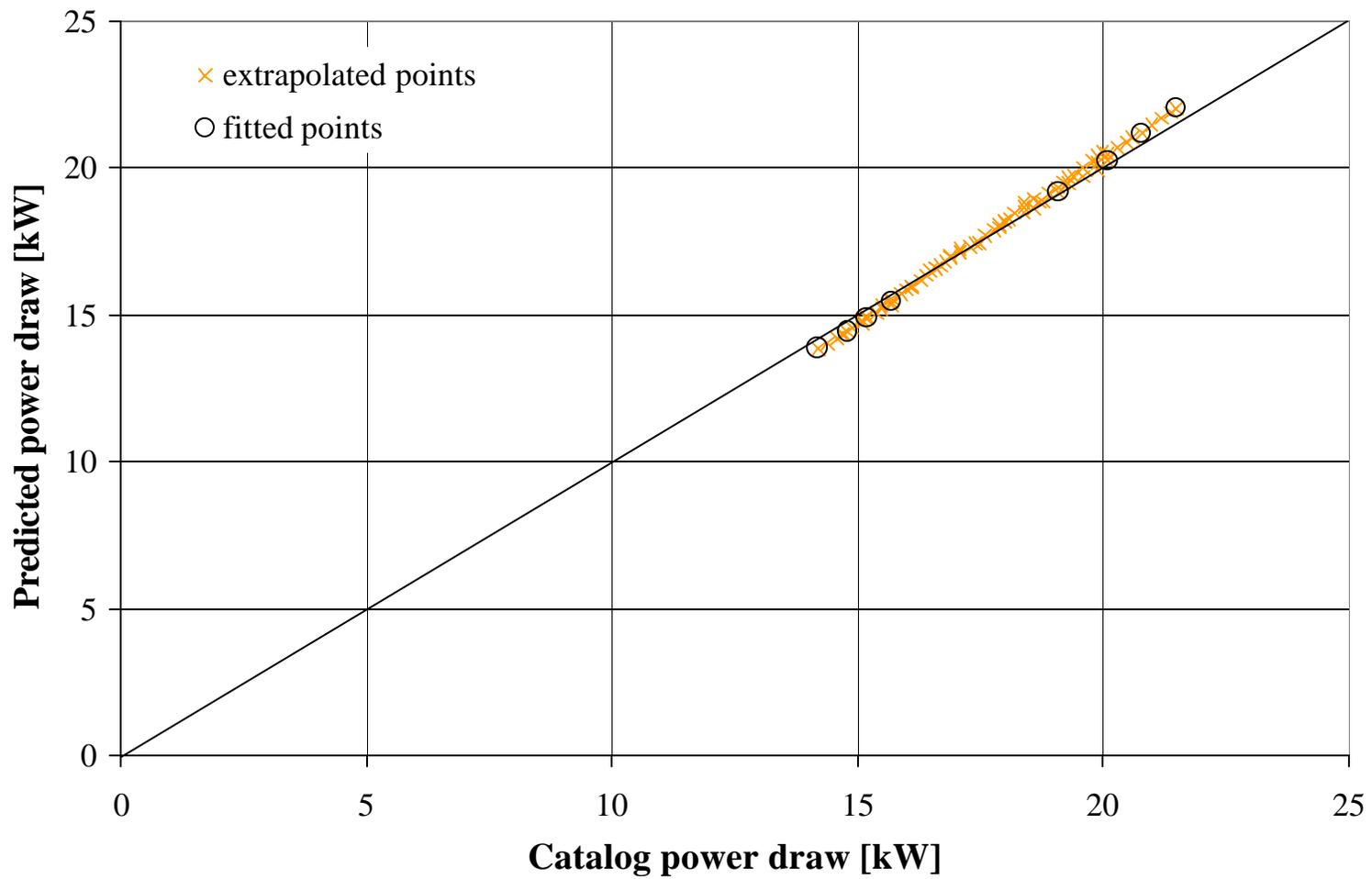


Model validation for data set (d)

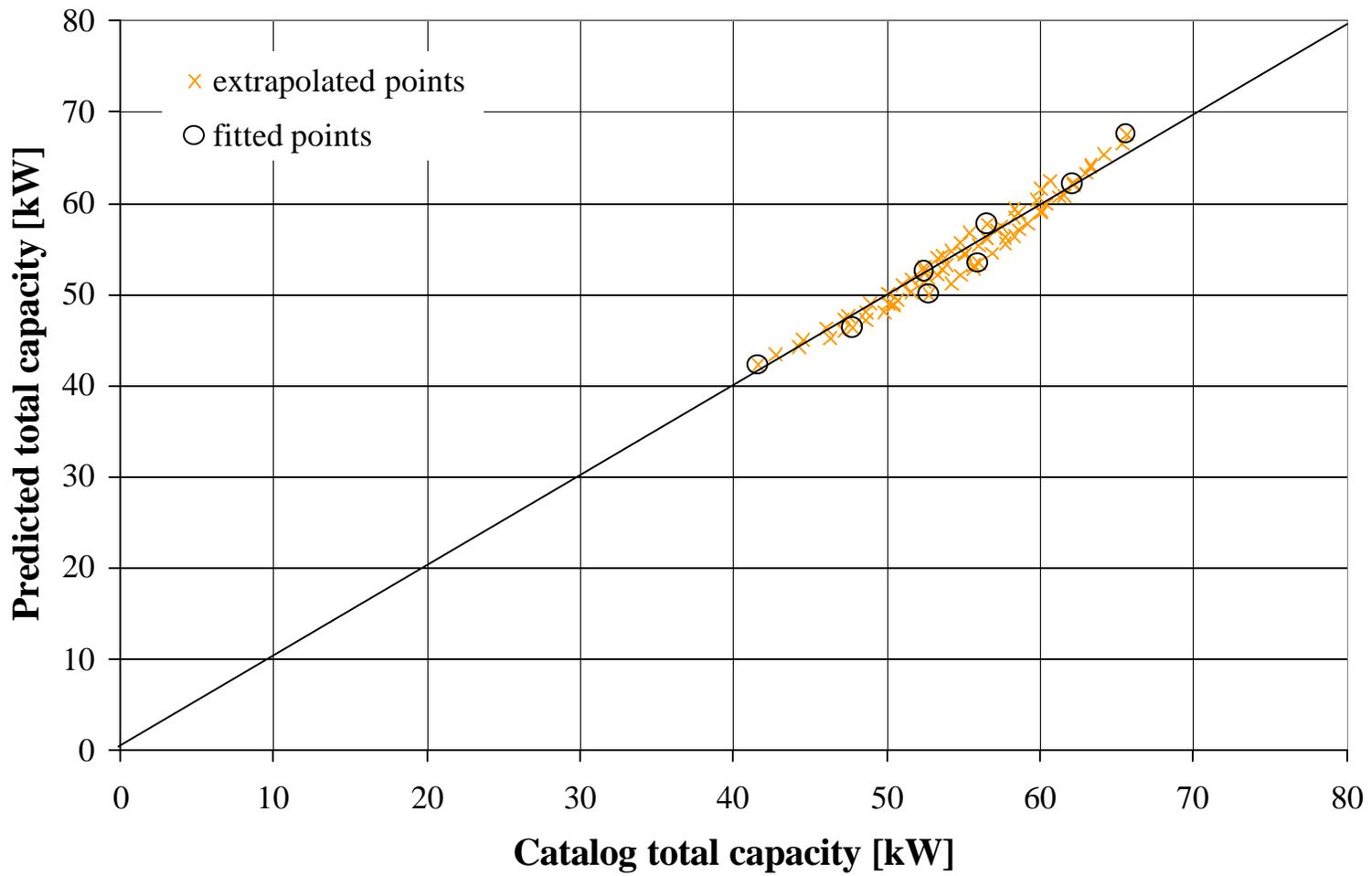
Appendix B: Model Results for Rooftop Packaged Unit 48 TJ 016 and 48 TJ 024

Data sets used for parameter fitting consisted of eight data points with extreme values for the volume flow rate, ambient temperature and air entering temperature.

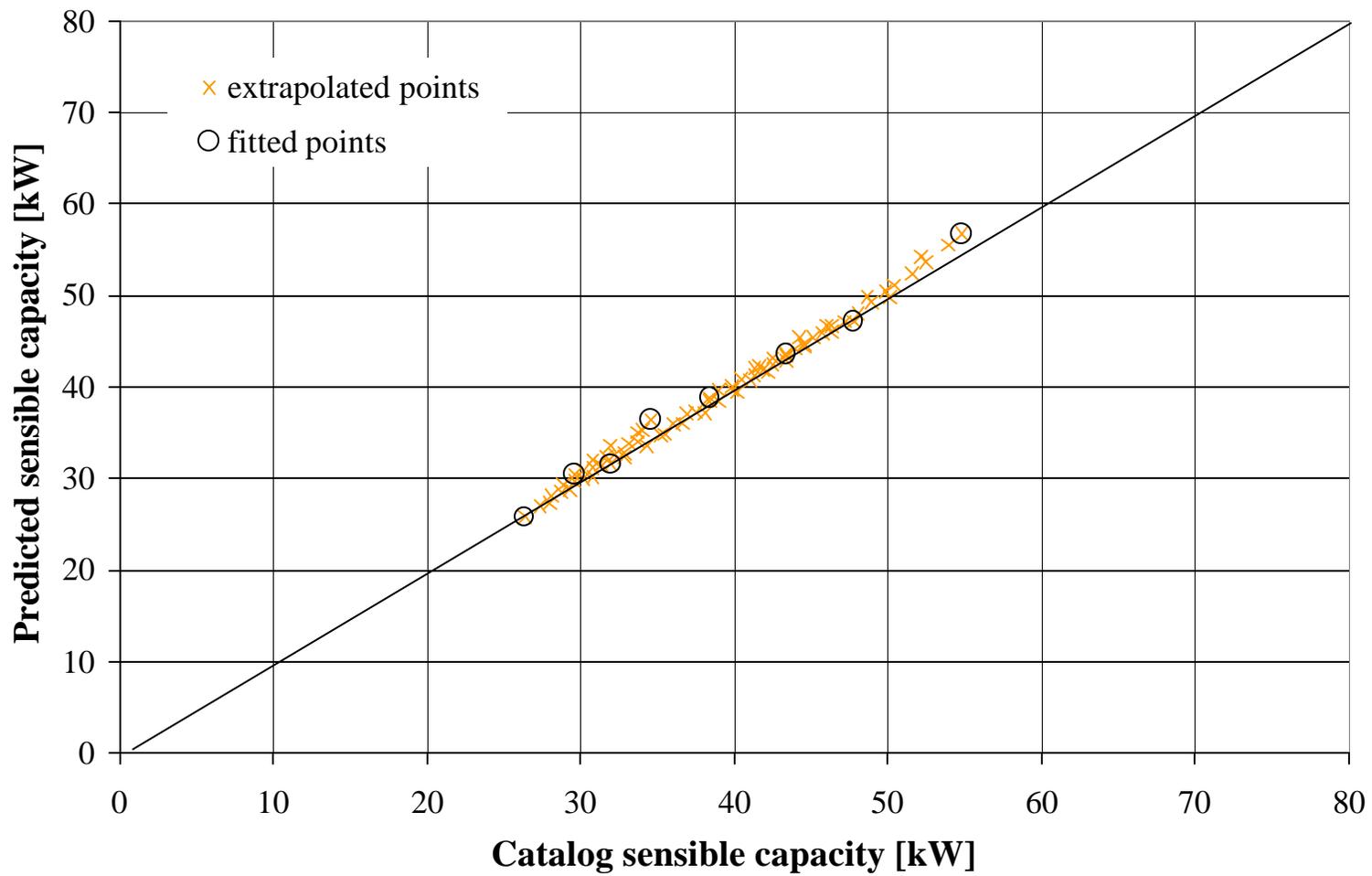
	Unit	
	(1)	(2)
	48 TJ 016	48 TJ 024
C ₁	1026	1025
C ₂	0.5	0.5
C ₃	1909	1343
Error (sensible) [%]	3.85	6.23
Error (total) [%]	4.78	2.72
C ₄	987.9	5904
C ₅	0.5	0.5
C ₆	31191	14482
Error [%]	8.85	0.164
Validation for 8 fitting points		
Error (power) [%]	1.87	4.22
Error (total capacity) [%]	3.00	2.82
Error (sensible capacity) [%]	2.78	1.94
Validation in catalog range		
Error (power) [%]	1.55	3.3
Error (total capacity) [%]	2.27	2.07
Error (sensible capacity) [%]	1.87	3.23



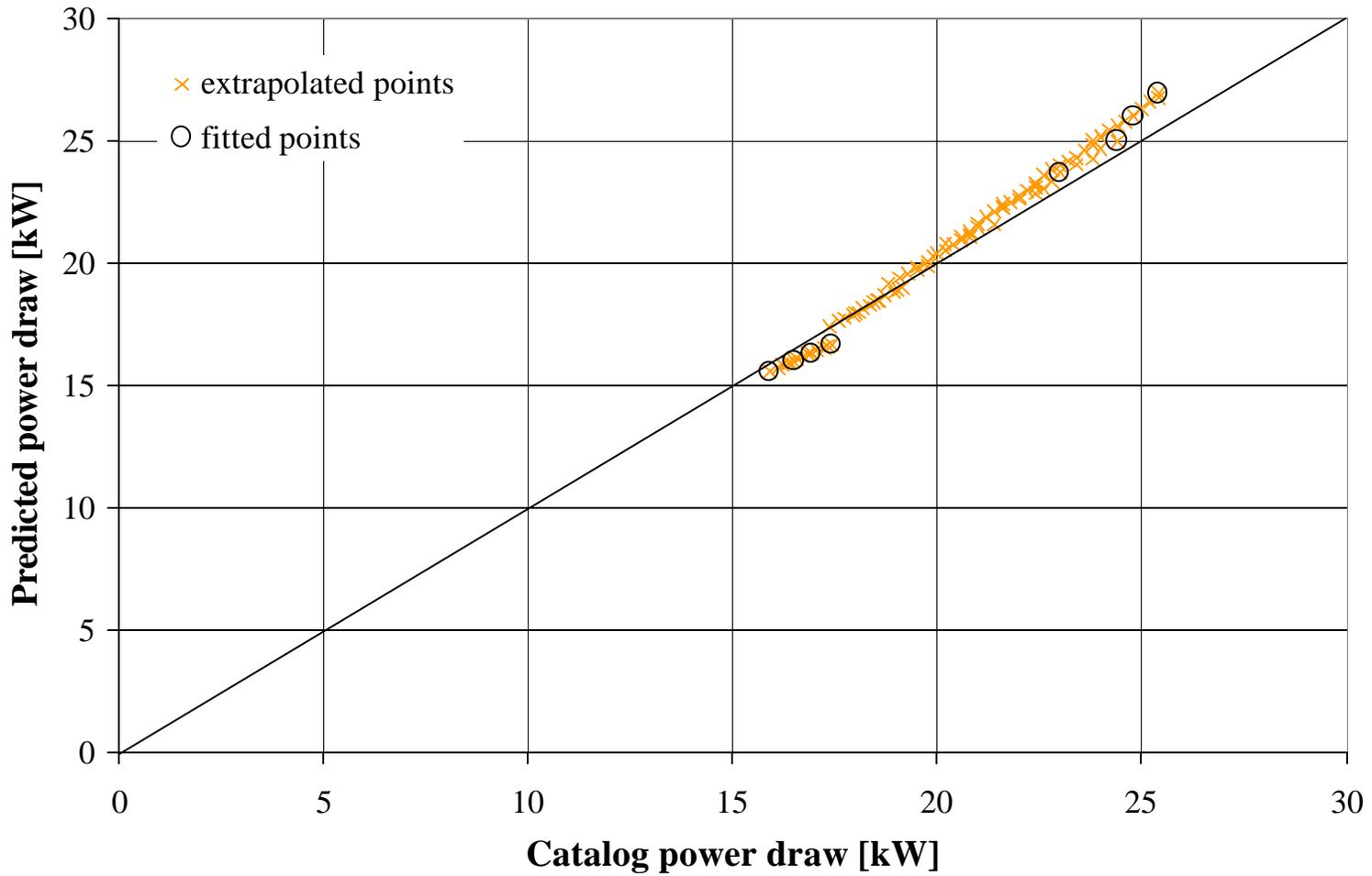
Model validation for rooftop packaged unit RT 48 TJ 016



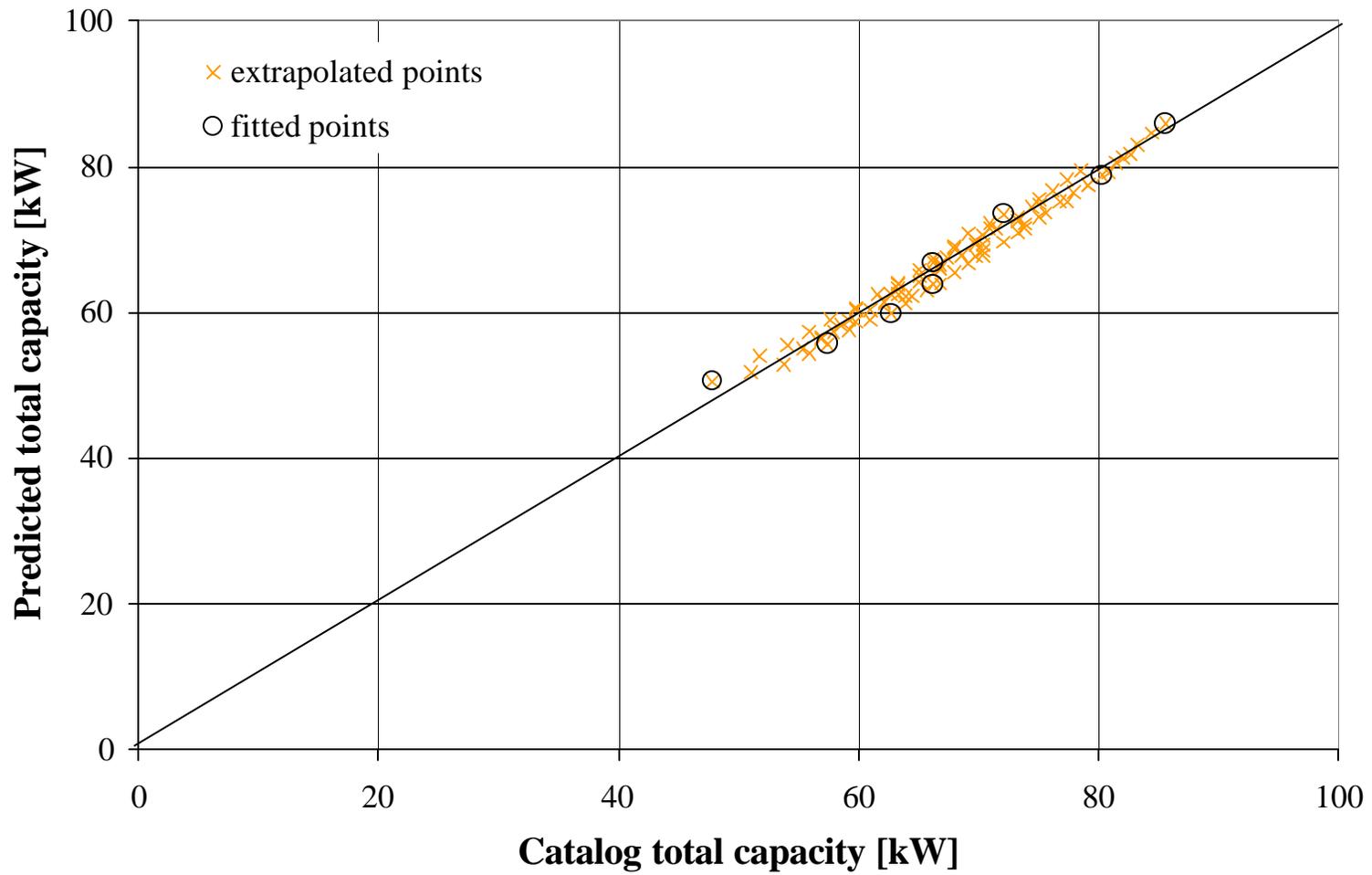
Model validation for rooftop packaged unit RT 48 TJ 016



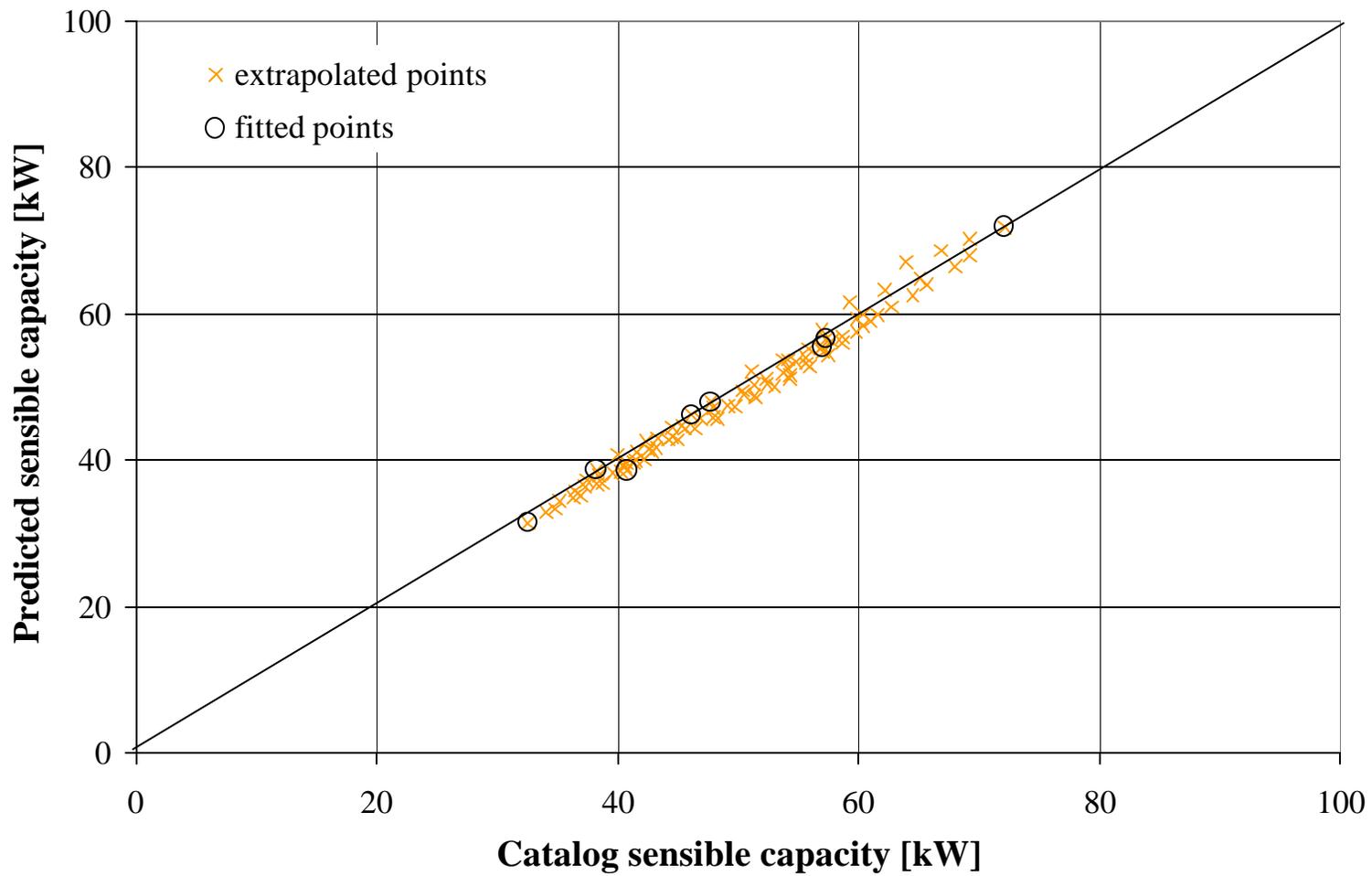
Model validation for rooftop packaged unit RT 48 TJ 016



Model validation for rooftop packaged unit RT 48 TJ 024



Model validation for rooftop packaged unit RT 48 TJ 024



Model validation for rooftop packaged unit RT 48 TJ 024

Appendix C: EES-code for compressor model (mass flow rate)

```
"Compressor mass flow rate model"
```

```
"This program calculates the compressor parameter for the mass
flow rate model based on manufacturer's catalog data."
```

```
$OPENLOOKUP 'parameter.lkt'
```

```
$SAVELOOKUP 'parameter.lkt'
```

```
FUNCTION writeLOOKUP(C,DELTAP,N_dot_diagr,V_dot_disp_par,
N_cy,nc,unitflag$,capormassflag$,T_sub_diagr,T_sup_diagr,LKT$)
LOOKUP(1,'N_dot'):=N_dot_diagr
LOOKUP(1,'V_dot_disp'):=V_dot_disp_par
LOOKUP(1,'N_cy'):=N_cy
LOOKUP(1,'C'):=C
LOOKUP(1,'DELTAP'):=DELTAP
LOOKUP(1,'nc'):=nc
LOOKUP(1,'uc'):=unitflag$
LOOKUP(1,'morc'):=capormassflag$
LOOKUP(1,'T_sub_diagr'):=T_sub_diagr
LOOKUP(1,'T_sup_diagr'):=T_sup_diagr
LOOKUP(1,'file_comp'):=LKT$
writeLOOKUP:=1
END
```

```
PROCEDURE capormass(capormassflag$,unitflag$,LKT$,i,h_1r`,
h_4r:Cap)
IF capormassflag$='Capacity' THEN
IF unitflag$='SI' THEN "SI system"
Cap:=LOOKUP(LKT$,i,'Capacity')*convert(kW,Btu/hr)
ELSE "English system"
Cap:=LOOKUP(LKT$,i,'Capacity')
ENDIF
ELSE
IF unitflag$='SI' THEN "SI system"
massflowrate:=LOOKUP(LKT$,i,'m_r')*convert(kg/hr,lbm/hr)
ELSE "English system"
massflowrate:=LOOKUP(LKT$,i,'m_r')
ENDIF
Cap:=massflowrate*(h_1r`-h_4r)
ENDIF
END
```

```

PROCEDURE unitconv(unitflag$,V_dot_disp_diagr,T_sub_diagr,
T_sup_diagr:V_dot_disp,T_sub,T_sup)
IF unitflag$='SI' THEN          "SI system"
V_dot_disp      :=  V_dot_disp_diagr*convert(1/s,ft^3/hr)
T_sub           :=  T_sub_diagr*1.8
T_sup           :=  T_sup_diagr*1.8
ELSE
          "English system"
V_dot_disp      :=  V_dot_disp_diagr*convert(ft^3/min,ft^3/hr)
T_sub           :=  T_sub_diagr
T_sup           :=  T_sup_diagr
ENDIF
END

```

```

PROCEDURE unitconvarray(unitflag$,T_condenser,T_evaporator:
T_cond,T_evap)
IF unitflag$='SI' THEN          "SI system"
T_cond          :=  T_condenser*1.8+32
T_evap          :=  T_evaporator*1.8+32
ELSE
          "English system"
T_cond          :=  T_condenser
T_evap          :=  T_evaporator
ENDIF
END

```

"Capacity or mass flow rate:"

```

DUPLICATE i=1,nc
CALL capormass(capormassflag$,unitflag$,LKT$,i,h_1r`[i],
h_4r[i]:Cap[i])
END

```

"Unit conversion:"

```

CALL unitconv(unitflag$,V_dot_disp_diagr,T_sub_diagr,
T_sup_diagr:V_dot_disp,T_sub,T_sup)
DUPLICATE i=1,nc
CALL unitconvarray(unitflag$,T_condenser[i],T_evaporator[i]:
T_cond[i],T_evap[i])

```

"Compressor input data:"

```

T_condenser[i]=LOOKUP(LKT$,i,'T_condenser')
T_evaporator[i] =LOOKUP(LKT$,i,'T_evaporator')
END

```

"Error calculation:"

```

Error_mass      =SQRT(SUM(error_mass[i]^2,i=1,nc)/nc)
RMS_mass=SQRT(SUM((m_r[i]-m_r_calc[i])^2,i=1,nc)/nc)
DUPLICATE i=1,nc
Error_mass[i]=(m_r[i]-m_r_calc[i])/m_r_mean
END
m_r_mean=SUM(m_r[i],i=1,nc)/nc

```

```

"Data range:"
lower_m=MIN(m_r[1..nc])/m_r_mean
upper_m=MAX(m_r[1..nc])/m_r_mean
"Gas constant:"
R_m=1.986
R=R_m/M_R22
M_R22=MOLARMASS(R22)
"Calculated mass flow rate:"
DUPLICATE i=1,nc
m_r_calc[i]=(1+C-C*((P_r_cond[i]/P_r_suction[i]))^(1/k[i]))*
V_dot_disp/v_suction[i]
"Volumetric efficiencies:"
eta_V[i]=m_r_calc[i]*v_suction[i]/V_dot_disp
eta_V_measured[i]=m_r[i]*v_suction[i]/V_dot_disp
"Capacity and mass flow rate:"
Cap[i]=m_r[i]*(h_1r`[i]-h_4r[i])
"Polytropic exponent:"
k[i]=SPECHEAT(R22,T=T_r_suction[i],P=P_r_suction[i])/(SPECHEAT
(R22,T=T_r_suction[i],P=P_r_suction[i])-R)
"Enthalpies:"
h_1r[i]=ENTHALPY(R22,P=P_1r[i], x=1)
h_1r`[i]=ENTHALPY(R22,P=P_1r`[i], T=T_1r`[i])
h_3r[i]=ENTHALPY(R22,P=P_3r[i], x=0)
h_3r`[i]=ENTHALPY(R22,P=P_3r[i],T=T_3r`[i]-0.001)
h_4r[i]=h_3r`[i]
"Temperatures:"
T_1r[i]=T_evap[i]
T_1r`[i]=T_evap[i]+T_sup
T_r_suction[i]=T_evap[i]+T_sup
T_3r[i]=T_cond[i]
T_3r`[i]=T_cond[i]-T_sub
T_4r[i]=T_evap[i]
"Pressures:"
P_r_cond[i]=PRESSURE(R22,T=T_cond[i],x=0)
P_r_evap[i]=PRESSURE(R22,T=T_evap[i],x=1)
P_r_suction[i]=P_r_evap[i]*(1-DELTAP)
P_dis\P_suc[i]=P_r_cond[i]/P_r_evap[i]
P_1r[i]=P_r_evap[i]
P_1r`[i]=P_r_evap[i]
P_2r[i]=P_r_cond[i]
P_2rs[i]=P_r_cond[i]
P_3r[i]=P_r_cond[i]
P_3r`[i]=P_r_cond[i]
P_4r[i]=P_r_evap[i]
"Specific volumes:"
v_suction[i]=VOLUME(R22,T=T_1r`[i],P=P_r_suction[i])

```

```
"Writes data to LOOKUP table:"  
dummy[i]=writeLOOKUP(C,DELTA $\rho$ ,N_dot_diagr,V_dot_disp_par,N_cy,  
nc,unitflag$,capormassflag$,T_sub_diagr,T_sup_diagr,LKT$)  
END  
V_dot_disp_par=V_dot_disp*convert(ft^3/hr,ft^3/min)
```

Appendix D: EES-code for compressor model (power)

```

"Compressor power model"

$OPENLOOKUP 'parameter.lkt'
$SAVELOOKUP 'parameter.lkt'

FUNCTION writeLOOKUP(a,b)
LOOKUP(1,'a'):=a
LOOKUP(1,'b'):=b
writeLOOKUP:=1
END

PROCEDURE capormass(capormassflag$,unitflag$,LKT$,i,h_1r`,
h_4r:Cap)
IF capormassflag$='Capacity' THEN
IF unitflag$='SI' THEN          "SI system"
Cap:=LOOKUP(LKT$,i,'Capacity')*convert(kW,Btu/hr)
ELSE          "English system"
Cap:=LOOKUP(LKT$,i,'Capacity')
ENDIF
ELSE
IF unitflag$='SI' THEN          "SI system"
massflowrate:=LOOKUP(LKT$,i,'m_r')*convert(kg/hr,lbm/hr)
ELSE          "English system"
massflowrate:=LOOKUP(LKT$,i,'m_r')
ENDIF
Cap:=massflowrate*(h_1r`-h_4r)
ENDIF
END

PROCEDURE unitconv(unitflag$,V_dot_disp,T_sub_diagr,
T_sup_diagr:V_dot_disp_diagr,T_sub,T_sup)
IF unitflag$='SI' THEN          "SI system"
V_dot_disp_diagr:=V_dot_disp*convert(ft^3/hr,l/s)
T_sub:=T_sub_diagr*1.8
T_sup:=T_sup_diagr*1.8
ELSE          "English system"
V_dot_disp_diagr:=V_dot_disp*convert(ft^3/hr,ft^3/min)
T_sub:=T_sub_diagr
T_sup:=T_sup_diagr
ENDIF
END

```

```

PROCEDURE unitconvarray(unitflag$,T_condenser,T_evaporator:
T_cond,T_evap)
IF unitflag$='SI' THEN          "SI system"
T_cond:=T_condenser*1.8+32
T_evap:=T_evaporator*1.8+32
ELSE          "English system"
T_cond:=T_condenser
T_evap:=T_evaporator
ENDIF
END

"Reads information from the parameter table:"
LKT$=LOOKUP$(1,'file_comp')
nc=LOOKUP(1,'nc')
unitflag$=LOOKUP$(1,'uc')
capormassflag$=LOOKUP$(1,'morc')
"Capacity or mass flow rate:"
DUPLICATE i=1,nc
CALL capormass(capormassflag$,unitflag$,LKT$,i,h_1r`[i],
h_4r[i]:Cap[i])
END
"Unit conversion:"
CALL unitconv(unitflag$,V_dot_disp,T_sub_diagr,
T_sup_diagr:V_dot_disp_diagr,T_sub,T_sup)
DUPLICATE i=1,nc
CALL unitconvarray(unitflag$,T_condenser[i],T_evaporator[i]:
T_cond[i],T_evap[i])
"Compressor input data:"
T_condenser[i]=LOOKUP(LKT$,i,'T_condenser')
T_evaporator[i]=LOOKUP(LKT$,i,'T_evaporator')
Power[i]=LOOKUP(LKT$,i,'Power')
END
V_dot_disp
=LOOKUP(1,'V_dot_disp')*convert(ft^3/min,ft^3/hr)
N_dot_diagr=LOOKUP(1,'N_dot')
N_dot=N_dot_diagr*convert(1/min,1/hr)
N_cy=LOOKUP(LKT$,1,'N_cy')
C=LOOKUP(1,'C')
DELTAP=LOOKUP(1,'DELTAP')
"Rating conditions:"
T_sub_diagr=LOOKUP(1,'T_sub_diagr')      "F, Subcooling"
T_sup_diagr=LOOKUP(1,'T_sup_diagr')     "F, Return gas
temperature"
"Error calculation:"
Error_mass      =SQRT(SUM(Error_mass[i]^2,i=1,nc)/nc)
RMS_mass=SQRT(SUM((m_r[i]-m_r_calc[i])^2,i=1,nc)/nc)

```

```

Error_power=SQRT(SUM(Error_power[i]^2,i=1,nc)/nc)
RMS_power=SQRT(SUM((Power[i]-P_com_calc[i])^2,i=1,nc)/nc)
DUPLICATE i=1,nc
Error_mass[i]=(m_r[i]-m_r_calc[i])/m_r_mean
Error_power[i]=(Power[i]-P_com_calc[i])/P_com_mean
END
m_r_mean=SUM(m_r[i],i=1,nc)/nc
P_com_mean=SUM(Power[i],i=1,nc)/nc
"Data range:"
lower_m=MIN(m_r[1..nc])/m_r_mean
upper_m=MAX(m_r[1..nc])/m_r_mean
lower_P=MIN(Power[1..nc])/P_com_mean
upper_P=MAX(Power[1..nc])/P_com_mean
"Gas constant:"
R_m=1.986
R=R_m/M_R22
M_R22=MOLARMASS(R22)
"Calculated mass flow rate:"
DUPLICATE i=1,nc
m_r_calc[i]=(1+C-C*((P_r_cond[i]/P_r_suction[i]))^(1/k[i]))*
V_dot_disp/v_suction[i]
"Volumetric efficiency:"
eta_V[i]=m_r_calc[i]*v_suction[i]/V_dot_disp
eta_V_measured[i]=m_r[i]*v_suction[i]/V_dot_disp
"Capacity and mass flow rate:"
Cap[i]=m_r[i]*(h_1r`[i]-h_4r[i])
"Power:"
P_com[i]=m_r[i]*(h_2r[i]-h_1r`[i])
P_com[i]=P_com_calc[i]*eta_comb[i]*convert(kW,Btu/hr)
P_com_calc[i]*eta_comb[i]=convert(W,kW)*m_r_calc[i]*convert(lb
m/hr,kg/s)*(k[i]/(k[i]-
1))*P_r_suction[i]*convert(atm,Pa)*(v_suction[i]*convert(ft^3/
lbm,m^3/kg))*((P_2r[i]/P_r_suction[i])^((k[i]-1)/k[i])-1)
eta_comb[i]=a+b/P_r_evap[i]
"Polytropic exponent:"
k[i]=SPECHEAT(R22,T=T_r_suction[i],P=P_r_suction[i])/(SPECHEAT
(R22,T=T_r_suction[i],P=P_r_suction[i])-R)
"Isentropic efficiency:"
eta_S[i]=m_r[i]*(h_2rs[i]-h_1r`[i])/P_com[i]
"Entropies:"
s_1r`[i]=ENTROPY(R22,P=P_1r`[i], T=T_1r`[i])
s_2rs[i]=s_1r`[i]
"Enthalpies:"
h_1r[i]=ENTHALPY(R22,P=P_1r[i], x=1)
h_1r`[i]=ENTHALPY(R22,P=P_1r`[i], T=T_1r`[i])
h_2rs[i]=ENTHALPY(R22,P=P_2rs[i],s=s_2rs[i])

```

```

h_3r[i]=ENTHALPY(R22,P=P_3r[i], x=0)
h_3r`[i]=ENTHALPY(R22,P=P_3r`[i],T=T_3r`[i]-0.001)
h_4r[i]=h_3r`[i]
"Temperatures:"
T_1r[i]=T_evap[i]
T_1r`[i]=T_evap[i]+T_sup
T_r_suction[i]=T_evap[i]+T_sup
T_3r[i]=T_cond[i]
T_3r`[i]=T_cond[i]-T_sub
T_4r[i]=T_evap[i]
"Pressures:"
P_r_cond[i]=PRESSURE(R22,T=T_cond[i],x=0)
P_r_evap[i]=PRESSURE(R22,T=T_evap[i],x=1)
P_r_suction[i]=P_r_evap[i]*(1-DELTAP)
P_dis\P_suc[i]=P_r_cond[i]/P_r_evap[i]
P_1r[i]=P_r_evap[i]
P_1r`[i]=P_r_evap[i]
P_2r[i]=P_r_cond[i]
P_2rs[i]=P_r_cond[i]
P_3r[i]=P_r_cond[i]
P_3r`[i]=P_r_cond[i]
P_4r[i]=P_r_evap[i]
"Specific volumes:"
v_suction[i]=VOLUME(R22,T=T_1r`[i],P=P_r_suction[i])
"Writes data in LOOKUP table:"
dummy[i]=writeLOOKUP(a,b)
"Unit conversion for plot windows:"
m_r_SI[i]=m_r[i]*convert(lbm/hr,kg/hr)
m_r_calc_SI[i]=m_r_calc[i]*convert(lbm/hr,kg/hr)
END

```

Appendix E: EES-code for vapor compression cycle

"Calculates thermodynamic state points and rate equations for the assumed vapor compression cycle"

```
$OPENLOOKUP 'parameter.lkt'
$SAVELOOKUP 'parameter.lkt'
```

```
PROCEDURE unitconvarray1(unitflag$,Q_evap_cat,SHC_cat,
T_db_amb_cat,T_db_1_cat,T_wb_1_cat:Q_evap,SHC,T_db_amb,T_db_1,
T_wb_1)
IF unitflag$='SI' THEN          "SI system"
Q_evap:=Q_evap_cat*convert(kW,Btu/s)
SHC:=SHC_cat*convert(kW,Btu/s)
T_db_amb:=T_db_amb_cat*1.8+32
T_db_1:=T_db_1_cat*1.8+32
T_wb_1:=T_wb_1_cat*1.8+32
ELSE          "English system"
Q_evap:=Q_evap_cat*convert(MBtu/hr,Btu/s)
SHC:=SHC_cat*convert(MBtu/hr,Btu/s)
T_db_amb:=T_db_amb_cat
T_db_1:=T_db_1_cat
T_wb_1:=T_wb_1_cat
ENDIF
END
```

```
PROCEDURE unitconvarray2(unitflag$,V_circ_cat,V_cat_cat,
A_evap_cat,Powerdraw_comp_cat:V_circ,V_cat,A_evap,
Powerdraw_comp)
IF unitflag$='SI' THEN          "SI system"
V_circ:=V_circ_cat*convert(1/s,ft^3/min)
V_cat:=V_cat_cat*convert(1/s,ft^3/min)
A_evap:=A_evap_cat*convert(m^2,ft^2)
Powerdraw_comp:=Powerdraw_comp_cat
ELSE          "English system"
V_circ:=V_circ_cat
V_cat:=V_cat_cat
A_evap:=A_evap_cat
Powerdraw_comp:=Powerdraw_comp_cat
ENDIF
END
```

```
FUNCTION writeLOOKUP1(i,length,CT,ST,vel,EDB,EWB,LDB,
LWB,T_air_in,T_air_out,T_cond,m_r,m_a,Q_cond,N_comp,A_evap)
LOOKUP(1,'Length'):=length
```

```

LOOKUP(i, 'CT'):=CT
LOOKUP(i, 'ST'):=ST
LOOKUP(i, 'vel')      :=vel
LOOKUP(i, 'EDB')      :=EDB
LOOKUP(i, 'EWB')      :=EWB
LOOKUP(i, 'LDB')      :=LDB
LOOKUP(i, 'LWB')      :=LWB
LOOKUP(i, 'T_air_in') :=T_air_in
LOOKUP(i, 'T_air_out'):=T_air_out
LOOKUP(i, 'T_cond'):=T_cond
LOOKUP(i, 'm_r')      :=m_r
LOOKUP(i, 'm_a')      :=m_a
LOOKUP(i, 'Q_cond'):=Q_cond
LOOKUP(1, 'N_comp'):=N_comp
LOOKUP(i, 'A_evap'):=A_evap
writeLOOKUP1:=1
END

FUNCTION writeLOOKUP2(m,rooftop$,unitflag$)
LOOKUP(1, 'nu'):=m
LOOKUP(1, 'file_unit'):=rooftop$
LOOKUP(1, 'uu'):=unitflag$
writeLOOKUP2:=1
END

n=1
"Writes information to the parameter table:"
DUPLICATE i=n,m
dummy1[i]=writeLOOKUP1(i,length,T_3r[i],T_1r[i],FPM[i],
T_db_1[i],T_wb_1[i],T_db_2[i],T_wb_2[i],T_db_amb[i],T_db_co[i]
,T_3r[i],m_r[i],m_cond_a[i],Q_cond[i],N_comp,A_evap[i])
END
dummy2=writeLOOKUP2(m,rooftop$,unitflag$)
"Knowns from catalog:"
P_amb=14.7
DUPLICATE i=n,m
CALL unitconvarray1(unitflag$,Q_evap_cat[i],SHC_cat[i],
T_db_amb_cat[i],T_db_1_cat[i],T_wb_1_cat[i]:Q_evap[i],SHC[i],
T_db_amb[i],T_db_1[i],T_wb_1[i])
CALL unitconvarray2(unitflag$,V_circ_cat[i],V_cat_cat[i],
A_evap_cat[i],Powerdraw_comp_cat[i]:V_circ[i],V_cat[i],
A_evap[i],Powerdraw_comp[i])
END
DUPLICATE i=n,m
Q_evap_cat[i]=LOOKUP(Rooftop$,i,#Q_tot)
SHC_cat[i]    =LOOKUP(Rooftop$,i,#SHC)

```

```

T_db_amb_cat[i] =LOOKUP(Rooftop$,i,#TDB_amb)
T_db_1_cat[i]=LOOKUP(Rooftop$,i,#TDB_i)
T_wb_1_cat[i]=LOOKUP(Rooftop$,i,#TWB_i)
V_circ_cat[i]=LOOKUP(Rooftop$,i,#V_air)
V_cat_cat[i]=LOOKUP(1,#V_dot_cond_fan)
Powerdraw_comp_cat[i]=LOOKUP(Rooftop$,i,#Power)
SHR[i]=SHC[i]/Q_evap[i]
END
A_evap_cat =A_evap_cat[1]
DUPLICATE i=n,m-1
A_evap_cat[i]=A_evap_cat[i+1]
END
"Evaporator (air-side):"
DUPLICATE i=n,m
Q_evap[i]=m_a[i]*(h_1[i]-h_2[i])
m_a[i]=rho_1[i]*V_a[i]
rho_1[i]=DENSITY(AirH2O,T=T_db_1[i],B=T_wb_1[i],P=P_1[i])
w_1[i]=HUMRAT(AirH2O,T=T_db_1[i],P=P_amb,B=T_wb_1[i])
w_2[i]=HUMRAT(AirH2O,T=T_db_2[i],P=P_amb,B=T_wb_2[i])
h_1[i]=ENTHALPY(AirH2O,T=T_db_1[i],P=P_1[i],w=w_1[i])
h_2[i]=ENTHALPY(AirH2O,T=T_db_2[i],P=P_2[i],w=w_2[i])
V_a[i]=V_circ[i]*convert(ft^3/min,ft^3/s)
P_1[i]=P_amb
Q_evap_sen[i]=Q_evap[i]*SHR[i]
Q_evap_sen[i]=m_a[i]*c_p1[i]*dT[i]
c_p1[i]=SPECHEAT(AirH2O,T=T_mean_evap_a[i],
B=T_wb_mean_evap_a[i],P=P_1[i])
dT[i]=T_db_1[i]-T_db_2[i]
T_mean_evap_a[i]=(T_db_1[i]+T_db_2[i])/2
T_wb_mean_evap_a[i]=(T_wb_1[i]+T_wb_2[i])/2
P_2[i]=P_1[i]
FPM[i]=V_circ[i]/A_evap[i]
END
Length=SQRT(A_evap[n]*convert(ft^2,in^2))
"Condenser (air-side):"
DUPLICATE i=n,m
Q_cond[i]=m_cond_a[i]*c_P_cond[i]*(T_db_co[i]-T_db_amb[i])
c_P_cond[i]=SPECHEAT(Air,T=T_mean_cond_a[i])
m_cond_a[i]=V_cond_a[i]/vol_cond_a[i]
V_cond_a[i]=V_cat[i]*convert(ft^3/min,ft^3/s)
vol_cond_a[i]=VOLUME(Air,T=T_db_amb[i],P=P_amb)
T_mean_cond_a[i]=(T_db_amb[i]+T_db_co[i])/2
END
"Compressor data:"
N_dot=LOOKUP(1,'N_dot')*convert(1/min,1/hr)
V_dot_disp =LOOKUP(1,'V_dot_disp')*convert(1/min,1/hr)

```

```

a=LOOKUP(1,'a')
b=LOOKUP(1,'b')
C=LOOKUP(1,'C')
DELTAP=LOOKUP(1,'DELTAP')
V_disp=V_dot_disp/N_dot
DUPLICATE i=n,m
P_r_suction[i]=P_r_evap[i]*(1-DELTAP)
"Specific volumes:"
v_suction[i]=VOLUME(R22,T=T_lr[i],P=P_r_suction[i]-0.1)
"Polytropic exponent:"
k[i]=SPECHEAT(R22,T=T_lr[i],P=P_r_suction[i]-0.1)/(SPECHEAT
(R22,T=T_lr[i],P=P_r_suction[i]-0.1)-R)
END
"Calculation of the power:"
DUPLICATE i=n,m
P_comp[i]=N_comp*(1-k_loss)*P_com_calc[i]*eta_comb[i]
*convert(W,Btu/s)
P_com_calc[i]*eta_comb[i]=m_r_calc[i]*convert(lbm/hr,kg/s)
*(k[i]/(k[i]-1))*P_r_suction[i]*convert(psia,Pa)*(v_suction[i]
*convert(ft^3/lbm,m^3/kg))*((P_r_cond[i]/P_r_suction[i])
^((k[i]-1)/k[i])-1)
eta_comb[i]=a+b/(P_r_evap[i]*convert(psia,atm))
"Volumetric efficiency:"
eta_V[i]=m_r_calc[i]*v_suction[i]/V_dot_disp
"Calculated mass flow rate:"
m_r_calc[i]=(1+C-C*((P_r_cond[i]/P_r_suction[i]))^(1/k[i]))
*V_dot_disp/v_suction[i]
"Output of compressor model:"
Powerdraw_comp[i]=Power_r_comp[i]*convert(Btu/s,kW)
Power_r_comp[i]=N_comp*P_com_calc[i]*convert(W,Btu/s)
Power_r_comp_kW[i]=N_comp*P_com_calc[i]*convert(W,kW)
m_r[i]=N_comp*m_r_calc[i]*convert(lbm/hr,lbm/s)
END
"Ideal gas constant:"
R_m=1.986
R=R_m/M_R22
M_R22=MOLARMASS(R22)
"Refrigeration cycle:"
DUPLICATE i=n,m
"Energy balance:"
Q_evap[i]+P_comp[i]=Q_cond[i]
"Heat transfer at the evaporator:"
Q_evap[i]=m_r[i]*(h_lr[i]-h_4r[i])
"Power relationships for the compressor:"
P_comp[i]=m_r[i]*(h_2r[i]-h_lr[i])
"Enthalpies:"

```

```
h_1r[i]=ENTHALPY(R22,T=T_1r[i],x=1)
h_2r`[i]=ENTHALPY(R22,T=T_3r[i],x=1)
Q_cond_real[i]=m_r[i]*(h_2r`[i]-h_3r[i])
h_3r[i]=ENTHALPY(R22,T=T_3r[i],x=0)
h_4r[i]=h_3r[i]
"Temperatures:"
T_2r[i]=TEMPERATURE(R22,h=h_2r[i],P=P_r_cond[i])
T_2r`[i]=TEMPERATURE(R22,P=P_r_cond[i],x=1)
T_4r[i]=TEMPERATURE(R22,h=h_4r[i],P=P_r_evap[i])
"Pressures:"
P_r_cond[i]=PRESSURE(R22,T=T_3r[i],x=0)
P_r_evap[i]=PRESSURE(R22,T=T_1r[i],x=1)
END
```

Appendix F: EES-code for direct expansion coil model

"Calculates the heat exchanger parameters used for the direct expansion coil model"

```

$OPENLOOKUP 'parameter.lkt'
$SAVELOOKUP 'parameter.lkt'

PROCEDURE DryCoil (Length,m_a,m_ref,EDB,EWB,CT,ST,C_1,C_2,C_3:
Q_dry,LDB_dry,LWB_dry,hA_o_dry,hA_i_dry)
LDB_dry:=EDB "assumed leaving dry bulb temperature"
w_i:=HUMRAT(AirH2O,T=EDB,B=EWB,P=14.7) "entering
humidity ratio"
10: MDB_dry:=(EDB+LDB_dry)/2 "mean dry bulb temperature"
MST_dry:=ST+(MDB_dry-ST)/10 "mean surface temperature"
cp_b_a_dry:=SPECHEAT(Air,T=MDB_dry) "specific
heat"
mu_b_a_dry:=VISCOSITY(Air,T=MDB_dry) "mean bulk
dynamic viscosity of air"
k_b_a_dry:=CONDUCTIVITY(Air,T=MDB_dry) "mean bulk
thermal conductivity of air"
cp_s_a_dry:=SPECHEAT(Air,T=MST_dry) "specific
heat"
mu_s_a_dry:=VISCOSITY(Air,T=MST_dry) "dynamic
viscosity of air at mean surface temperature"
k_s_a_dry:=CONDUCTIVITY(Air,T=MST_dry) "thermal
conductivity of air at mean surface temperature"
P_cond:=PRESSURE(R22,T=CT,x=0) "condenser
pressure"
P_evap:=PRESSURE(R22,T=ST,x=0) "evaporator
pressure"
mu_ref_1:=VISCOSITY(R22,T=ST,P=P_evap+0.001) "dynamic
viscosity of liquid refrigerant"
k_ref_1:=CONDUCTIVITY(R22,T=ST,P=P_evap+0.001) "thermal
conductivity of liquid refrigerant"
h_in:=ENTHALPY(R22,T=CT,x=0) "specific enthalpy of inlet
refrigerant"
h_out:=ENTHALPY(R22,T=ST,x=1) "specific enthalpy of
outlet refrigerant"
x_in:=QUALITY(R22,h=h_in,T=ST) "inlet quality of
refrigerant"
h_fg:=ENTHALPY(R22,T=ST,x=1)-ENTHALPY(R22,T=ST,x=0) "latent
heat of fusion"
"Overall heat transfer coefficient-area products (dry-coil)"

```

```

hA_o_dry:=C_1*(m_a/mu_b_a_dry)^C_2*(mu_b_a_dry*cp_b_a_dry/k_b_
a_dry)^0.36*((mu_b_a_dry*cp_b_a_dry/k_b_a_dry)/(mu_s_a_dry*cp_
s_a_dry/k_s_a_dry))^0.25*k_b_a_dry"Btu/hr-ft"
hA_i_dry:=C_3*k_ref_1*((m_ref/mu_ref_1)*(1-x_in)*h_fg)^0.45
UA_tot_dry:=1/(1/hA_i_dry+1/hA_o_dry) "Btu/hr-F"
"Capacitance rates"
C_min:=m_a*cp_b_a_dry "minimum capacitance rate"
NTU_dry:=UA_tot_dry/C_min "totally dry coil NTU's"
epsilon_dry:=1-exp(-NTU_dry) "effectiveness of dry portion
coil"
Q_dry:=epsilon_dry*C_min*(EDB-ST) "energy transfer rate
for a totally dry coil"
LDB_new_dry:=EDB-Q_dry/(m_a*cp_b_a_dry) "air outlet
temperature"
IF (abs(LDB_dry-LDB_new_dry)>=0.1) THEN
LDB_dry:=LDB_new_dry
goto 10
ENDIF
LWB_dry:=WetBulb(AirH2O,T=LDB_dry,P=14.7,W=w_i) "wet bulb
temperature of air leaving dry portion of coil"
END
PROCEDURE WetCoil (Length,m_a,m_ref,EDB,EWB,CT,ST,FPM,C_1,C_2,
C_3:Q_wet,LDB_wet,LWB_wet,hA_o_wet,hA_i_wet)
DP_i=DewPoint(AirH2O,T=EDB,B=EWB,P=14.7) "entering
dewpoint"
LDB_wet:=EWB-20 "assumed leaving dry bulb
temperature"
w_i:=HUMRAT(AirH2O,T=EDB,B=EWB,P=14.7) "entering
humidity ratio"
20: MDB_wet:=(EDB+LDB_wet)/2 "mean dry bulb temperature for a
totally wet coil"
MST_wet:=ST+(MDB_wet-ST)/10 "mean surface temperature for a
totally wet coil"
cp_sat:=(ENTHALPY(AirH2O,T=DP_i,R=1,P=14.7)-
ENTHALPY(AirH2O,T=ST,R=1,P=14.7))/(DP_i-ST) "saturation
specific heat"
h_a_i:=ENTHALPY(AirH2O,T=EDB,B=EWB,P=14.7) "specIFic
enthalpy of entering air"
h_ref_sat_i:=ENTHALPY(AirH2O,T=ST,R=1,P=14.7) "saturation
specific enthalpy at suction temperature"
cp_b_a_wet:=SPECHEAT(Air,T=MDB_wet) "specific heat"
mu_b_a_wet:=VISCOSITY(Air,T=MDB_wet) "mean bulk dynamic
viscosity of air"
k_b_a_wet:=CONDUCTIVITY(Air,T=MDB_wet) "mean bulk thermal
conductivity of air"
cp_s_a_wet:=SPECHEAT(Air,T=MST_wet) "specific heat"

```

```

mu_s_a_wet:=VISCOSITY(Air,T=MST_wet)    "dynamic viscosity of
air at mean surface temperature"
k_s_a_wet:=CONDUCTIVITY(Air,T=MST_wet) "thermal conductivity
of air at mean surface temperature"
P_cond:=PRESSURE(R22,T=CT,x=0)    "condenser pressure"
P_evap:=PRESSURE(R22,T=ST,x=0)    "evaporator pressure"
mu_ref_1:=VISCOSITY(R22,T=ST,P=P_evap+0.001)"dynamic
viscosity of liquid refrigerant"
k_ref_1:=CONDUCTIVITY(R22,T=ST,P=P_evap+0.001) "thermal
conductivity of liquid refrigerant"
h_in:=ENTHALPY(R22,T=CT,x=0) " specific enthalpy of inlet
refrigerant"
h_out:=ENTHALPY(R22,T=ST,x=1)      "specific enthalpy of
outlet refrigerant"
x_in:=QUALITY(R22,h=h_in,T=ST)      "inlet quality of
refrigerant"
h_fg:=ENTHALPY(R22,T=ST,x=1)-ENTHALPY(R22,T=ST,x=0) "latent
heat of fusion"
"Overall heat transfer coefficient-area products (wet-coil)"
C_f:=0.626*FPM^0.101                "air side convection coefficient
correction factor for a totally wet coil"
hA_o_wet:=C_f*C_1*(m_a/mu_b_a_wet)^C_2*(mu_b_a_wet*cp_b_a_wet
/k_b_a_wet)^0.36*((mu_b_a_wet*cp_b_a_wet/k_b_a_wet)
/(mu_s_a_wet*cp_s_a_wet/k_s_a_wet))^0.25*k_b_a_wet
    "Btu/hr-ft"
hA_i_wet:=C_3*k_ref_1*((m_ref/mu_ref_1)*(1-x_in)*h_fg)^0.45
UA_h:=1/(cp_sat/hA_i_wet+cp_b_a_wet/hA_o_wet)    "Btu/hr-ft"
capAirWet:=m_a
NTU_wet:=UA_h/m_a
epsilon_wet:=1-exp(-NTU_wet)    "effectiveness of wet coil"
Qmax_wet:=m_a*abs(h_a_i-h_ref_sat_i)
h_a_o:=h_a_i-epsilon_wet*Qmax_wet/capAirWet
h_ref_sat_o:=h_ref_sat_i
R_i:=1/hA_i_wet
R_o:=1/hA_o_wet
Rratio:=R_i/(R_i+cp_b_a_wet/cp_sat*R_o)
h_s_sat_i:=h_ref_sat_o+Rratio*(h_a_i-h_ref_sat_o)
h_s_sat_o:=h_ref_sat_i+Rratio*(h_a_o-h_ref_sat_i)
T_s_i:=TEMPERATURE(AirH2O,P=14.7,R=1,h=h_s_sat_i)
Q_wet:=m_a*(h_a_i-h_a_o)
capAir:=m_a*(SpecHeat(Air,T=MDB_wet)+w_i*SpecHeat(H2O,
T=MDB_wet))
capAir_2:=m_a*SPECHEAT(AirH2O,T=MDB_wet,P=14.7,w=w_i)
NTU:=hA_o_wet/capAir
epsilon:=1-exp(-NTU)
h_sat_cond:=h_a_i-(h_a_i-h_a_o)/epsilon

```

```

T_cond:=TEMPERATURE(AirH2O,P=14.7,R=1,H=h_sat_cond)
LDB_new_wet:=EDB-(EDB-T_cond)*epsilon
IF (abs(LDB_wet-LDB_new_wet)>=0.1) THEN
LDB_wet:=LDB_new_wet
goto 20
ENDIF
w_o:=(h_a_o-(ENTHALPY(Air,T=LDB_wet)-
ENTHALPY(Air,T=0)))/ENTHALPY(Water,T=LDB_wet,x=1)
LWB_wet:=WETBULB(AirH2O,T=LDB_wet,P=14.7,w=w_o) "F; wet bulb
temperature of air leaving wet portion of coil"
END
PROCEDURE Compare(EDB, EWB, ST, Q_dry, LDB_dry, LWB_dry,
hA_o_dry, hA_i_dry, Q_wet, LDB_wet, LWB_wet, hA_o_wet,
hA_i_wet: Q_calc, LDB_calc, LWB_calc, hA_o_calc, hA_i_calc)
DP_i=DEWPOINT(AirH2O,T=EDB,B=EWB,P=14.7) "F; entering dew
point"
IF (DP_i<=ST) THEN
Q_calc:=Q_dry
LDB_calc:=LDB_dry
LWB_calc:=LWB_dry
hA_o_calc:=hA_o_dry
hA_i_calc:=hA_i_dry
ELSE
Ts_i=(EDB+ST)/2
IF (DP_i<=Ts_i) THEN
IF (Q_dry>Q_wet) THEN
Q_calc:=Q_dry
LDB_calc:=LDB_dry
LWB_calc:=LWB_dry
hA_o_calc:=hA_o_dry
hA_i_calc:=hA_i_dry
ELSE
Q_calc:=Q_wet
LDB_calc:=LDB_wet
LWB_calc:=LWB_wet
hA_o_calc:=hA_o_wet
hA_i_calc:=hA_i_wet
ENDIF
ELSE
Q_calc:=Q_wet
LDB_calc:=LDB_wet
LWB_calc:=LWB_wet
hA_o_calc:=hA_o_wet
hA_i_calc:=hA_i_wet
ENDIF
ENDIF

```

END

```

FUNCTION writeLOOKUP(C_1,C_2,C_3)
LOOKUP(1,'C_1'):=C_1
LOOKUP(1,'C_2'):=C_2
LOOKUP(1,'C_3'):=C_3
writeLOOKUP:=1
END

n=LOOKUP(1,'nu')
DUPLICATE i=1,n
EDB[i]=LOOKUP(i,#EDB)    "entering air dry bulb temperature"
EWB[i]=LOOKUP(i,#EWB)    "entering air wet bulb temperature"
CT[i]=LOOKUP(i,#CT)     "condensing temperature"
ST[i]=LOOKUP(i,#ST)     "suction temperature"
FPM[i]=LOOKUP(i,#vel)   "air velocity"
LDB[i]=LOOKUP(i,#LDB)   "leaving air dry bulb temperature"
LWB[i]=LOOKUP(i,#LWB)   "leaving air wet bulb temperature"
"Catalog performance"
h_i[i]=ENTHALPY(AirH2O,T=EDB[i],B=EWB[i],P=14.7)
h_o[i]=ENTHALPY(AirH2O,T=LDB[i],B=LWB[i],P=14.7)
m_a[i]=rho_std[i]*Area*FPM[i]*60
Q_act[i]=m_a[i]*(h_i[i]-h_o[i])
h_ref_i[i]=ENTHALPY(R22,T=CT[i],X=0)
h_ref_o[i]=ENTHALPY(R22,T=ST[i],X=1)
Q_act[i]=m_ref[i]*(h_ref_o[i]-h_ref_i[i])
"Coil models"
Call DryCoil (Length,m_a[i],m_ref[i],EDB[i],EWB[i],CT[i],ST[i],
,C_1,C_2,C_3:Q_dry[i],LDB_dry[i],LWB_dry[i],hA_o_dry[i],
hA_i_dry[i])
Call WetCoil (Length,m_a[i],m_ref[i],EDB[i],EWB[i],CT[i],
ST[i],FPM[i],C_1,C_2,C_3:Q_wet[i],LDB_wet[i],LWB_wet[i],
hA_o_wet[i],hA_i_wet[i])
Call Compare
(EDB[i],EWB[i],ST[i],Q_dry[i],LDB_dry[i],LWB_dry[i],
hA_o_dry[i],hA_i_dry[i],Q_wet[i],LDB_wet[i],LWB_wet[i],
hA_o_wet[i],hA_i_wet[i]:Q_calc[i],LDB_calc[i],LWB_calc[i],
hA_o_calc[i],hA_i_calc[i])
END
Length=LOOKUP(1,#length)
Area=length^2*convert(in^2,ft^2)
DUPLICATE i=1,n
rho_std[i]=DENSITY(Air,T=EDB[i],P=14.7)
Q_evap[i]=Q_act[i]*convert(Btu/hr,Btu/s)
Q_evap_calc[i]=Q_calc[i]*convert(Btu/hr,Btu/s)
Error_evap_mean[i]=(Q_evap[i]-Q_evap_calc[i])/Q_evap_mean

```

```

END
Q_evap_mean=SUM(Q_evap[i],i=1,n)/(n)
Error_evap=SQRT(SUM((Error_evap_mean[i])^2,i=1,n)/(n))
RMS_evap=SQRT(SUM((Q_evap[i]-Q_evap_calc[i])^2,i=1,n)/n)
lower_Q=MIN(Q_evap[1..n])/Q_evap_mean
upper_Q=MAX(Q_evap[1..n])/Q_evap_mean
DUPLICATE i=1,n
"Calculation of plot variables:"
Q_evaporator[i]=Q_evap[i]*convert(Btu/s,MBtu/hr)
Q_evaporator_calc[i]
    =Q_evap_calc[i]*convert(Btu/s,MBtu/hr)
Q_evaporator_SI[i]=Q_evap[i]*convert(Btu/s,kW)
Q_evaporator_calc_SI[i]=Q_evap_calc[i]*convert(Btu/s,kW)
END
dummy=writeLOOKUP(C_1,C_2,C_3)
DUPLICATE i=1,n
T_db_mean[i]=(EDB[i]+LDB[i])/2
T_wb_mean[i]=(EWB[i]+LWB[i])/2
c_p_mean[i]=SPECHEAT(AirH2O,T=T_db_mean[i],B=T_wb_mean[i],
p=14.7)
SHC[i]=m_a[i]*c_p_mean[i]*(EDB[i]-LDB[i])
*convert(Btu/hr,Btu/s)
T_db_mean_calc[i]=(EDB[i]+LDB_calc[i])/2
T_wb_mean_calc[i]=(EWB[i]+LWB_calc[i])/2
c_p_mean_calc[i]=SPECHEAT(AirH2O,T=T_db_mean_calc[i],
B=T_wb_mean_calc[i],p=14.7)
SHC_calc[i]=m_a[i]*c_p_mean_calc[i]*(EDB[i]-LDB_calc[i])
*convert(Btu/hr,Btu/s)
Error_SHC_mean[i]=(SHC[i]-SHC_calc[i])/SHC_mean
END
SHC_mean=SUM(SHC[i],i=1,n)/(n)
Error_SHC=SQRT(SUM((Error_SHC_mean[i])^2,i=1,n)/(n))
Error_total=(Error_SHC+Error_evap)/2

```

Appendix G: EES-code for 3-parameter condensing coil model

"Calculates the heat exchanger parameters used for the 3-parameter condensing coil model"

```

$OPENLOOKUP 'parameter.lkt'
$SAVELOOKUP 'parameter.lkt'

PROCEDURE CondCoil(T_air_in,T_air_out,T_cond,m_r,m_a,C_4,C_5,
C_6: hA_i,hA_o,Q_calc,epsilon)
m_r:=m_r*3600
m_a:=m_a*3600
P_cond:=PRESSURE(R22,T=T_cond,x=1)
c_P:=SPECHEAT(R22,T=T_cond,P=P_cond+0.001)
k_r:=CONDUCTIVITY(R22,T=T_cond,x=0)
mu_r:=VISCOSITY(R22,T=T_cond,x=0)
rho_v_r:=DENSITY(R22,T=T_cond,x=1)
rho_l_r:=DENSITY(R22,T=T_cond,x=0)
Pr:=c_P*mu_r/k_r
h_fg:=ENTHALPY(R22,T=T_cond,x=1)-ENTHALPY(R22,T=T_cond,x=0)
MDB:=(T_air_in+T_air_out)/2      "mean dry bulb temperature"
MST:=T_cond-(T_cond-MDB)/10    "mean surface temperature"
hA_i:=C_6*(rho_l_r*(rho_l_r-rho_v_r)*k_r^3*h_fg/(mu_r*(T_cond
-MST)))^0.25
c_P_b_a:=SPECHEAT(Air,T=MDB)    "specific heat"
mu_b_a:=VISCOSITY(Air,T=MDB)    "mean bulk dynamic
viscosity of air"
k_b_a:=CONDUCTIVITY(Air,T=MDB)  "mean bulk thermal
conductivity of air"
c_P_s_a:=SPECHEAT(Air,T=MST)    "specific heat"
mu_s_a:=VISCOSITY(Air,T=MST)    "dynamic viscosity of air
at mean surface temperature"
k_s_a:=CONDUCTIVITY(Air,T=MST)  "thermal conductivity
of air at mean surface temperature"
hA_o:=C_4*(m_a/mu_b_a)^C_5*(mu_b_a*c_P_b_a/k_b_a)^0.36*((mu_b_
a*c_P_b_a/k_b_a)/(mu_s_a*c_P_s_a/k_s_a))^0.25*k_b_a
UA_tot:=1/(1/hA_i+1/hA_o)
C_min:=m_a*c_P_b_a
NTU:=UA_tot/C_min
epsilon:=1-exp(-NTU)
Q_calc:=epsilon*C_min*(T_cond-T_air_in)
END

FUNCTION writeLOOKUP(C_4,C_5,C_6)
LOOKUP(1,'C_4')      :=C_4

```

```

LOOKUP(1, 'C_5')      :=C_5
LOOKUP(1, 'C_6')      :=C_6
writeLOOKUP:=1
END

dummy=writeLOOKUP(C_4,C_5,C_6)
n=1
m=LOOKUP(1, 'nu')
DUPLICATE i=n,m
Call
CondCoil(T_air_in[i],T_air_out[i],T_cond[i],m_r[i],m_a[i],C_4,
C_5,C_6:hA_i[i],hA_o[i],Q_co_calc_hr[i],epsilon[i])
Q_cond_calc[i]=Q_co_calc_hr[i]*convert(Btu/hr,Btu/s)
END
DUPLICATE i=n,m
T_air_in[i]=LOOKUP(i, 'T_air_in')
T_air_out[i]=LOOKUP(i, 'T_air_out')
T_cond[i]=LOOKUP(i, 'T_cond')
m_r[i]=LOOKUP(i, 'm_r')
m_a[i]=LOOKUP(i, 'm_a')
Q_cond[i]=      LOOKUP(i, 'Q_cond')
Error_cond_mean[i]=(Q_cond[i]-Q_cond_calc[i])/Q_cond_mean
END
Q_cond_mean=SUM(Q_cond[i],i=n,m)/(m-n+1)
Error_cond      =SQRT(SUM((Error_cond_mean[i])^2,I=n,m)/(m-
n+1))
RMS_cond=SQRT(SUM((Q_cond[i]-Q_cond_calc[i])^2,i=n,m)/(m-n+1))
lower_Q=MIN(Q_cond[1..m])/Q_cond_mean
upper_Q=MAX(Q_cond[1..m])/Q_cond_mean
DUPLICATE i=n,m
"Calculation of plot variables:"
Q_condenser[i]=Q_cond[i]*convert(Btu/s,MBtu/hr) "English
units"
Q_condenser_calc[i]=Q_cond_calc[i]*convert(Btu/s,MBtu/hr)
Q_condenser_SI[i]=Q_cond[i]*convert(Btu/s,kW)
Q_condenser_calc_SI[i]=Q_cond_calc[i]*convert(Btu/s,kW)
END

```

Appendix H: EES-code for rooftop packaged model

"Program calculates predicted performance for different operating points using the parameters for the condenser, evaporator and compressor model"

```

$OPENLOOKUP 'parameter.lkt'
$SAVELOOKUP 'parameter.lkt'

PROCEDURE CondCoil(T_air_in,T_air_out,T_cond,m_r,m_a,C_4,C_5,
C_6:hA_i,hA_o,Q_calc,epsilon)
m_r:=m_r*3600
m_a:=m_a*3600
P_cond:=PRESSURE(R22,T=T_cond,x=1)
c_P:=SPECHEAT(R22,T=T_cond,P=P_cond+0.001)
k_r:=CONDUCTIVITY(R22,T=min(152,T_cond),x=0)
mu_r:=VISCOSITY(R22,T=T_cond,x=0)
rho_v_r:=DENSITY(R22,T=T_cond,x=1)
rho_l_r:=DENSITY(R22,T=T_cond,x=0)
Pr:=c_P*mu_r/k_r
h_fg:=ENTHALPY(R22,T=T_cond,x=1)-ENTHALPY(R22,T=T_cond,x=0)
MDB:=(T_air_in+T_air_out)/2      "mean dry bulb temperature"
MST:=T_cond-(T_cond-MDB)/10    "mean surface temperature"
hA_i:=C_6*(rho_l_r*(rho_l_r-rho_v_r)*k_r^3*h_fg/(mu_r*(T_cond
-MST)))^0.25
c_P_b_a:=SPECHEAT(Air,T=MDB)    "specific heat"
mu_b_a:=VISCOSITY(Air,T=MDB)    "mean bulk dynamic
viscosity of air"
k_b_a:=CONDUCTIVITY(Air,T=MDB)  "mean bulk thermal
conductivity of air"
c_P_s_a:=SPECHEAT(Air,T=MST)    "specific heat"
mu_s_a:=VISCOSITY(Air,T=MST)    "dynamic viscosity of air
at mean surface temperature"
k_s_a:=CONDUCTIVITY(Air,T=MST)  "thermal conductivity of
air at mean surface temperature"
hA_o:=C_4*(m_a/mu_b_a)^C_5*(mu_b_a*c_P_b_a/k_b_a)^0.36
*((mu_b_a*c_P_b_a/k_b_a)/(mu_s_a*c_P_s_a/k_s_a))^0.25*k_b_a
UA_tot:=1/(1/hA_i+1/hA_o)
C_min:=m_a*c_P_b_a
NTU:=UA_tot/C_min
epsilon:=1-exp(-NTU)
Q_calc:=epsilon*C_min*(T_cond-T_air_in)
END

```

```

PROCEDURE DryCoil (Length,m_a,m_ref,EDB,EWB,CT,ST,C_1,C_2,C_3:
Q_dry,LDB_dry,LWB_dry,hA_o_dry,hA_i_dry)
m_ref:=m_ref*3600
m_a:=m_a*3600
LDB_dry:=EDB          "assumed leaving dry bulb temperature"
w_i:=HUMRAT(AirH2O,T=EDB,B=EWB,P=14.7)      "entering
humidity ratio"
10: MDB_dry:=(EDB+LDB_dry)/2          "mean dry bulb temperature"
MST_dry:=ST+(MDB_dry-ST)/10          "mean surface temperature"
cp_b_a_dry:=SPECHEAT(Air,T=MDB_dry)    "specific heat"
mu_b_a_dry:=VISCOSITY(Air,T=MDB_dry)   "mean bulk dynamic
viscosity of air"
k_b_a_dry:=CONDUCTIVITY(Air,T=MDB_dry) "mean bulk thermal
conductivity of air"
cp_s_a_dry      :=SPECHEAT(Air,T=MST_dry)          "specific
heat"
mu_s_a_dry      :=VISCOSITY(Air,T=MST_dry)         "dynamic
viscosity of air at mean surface temperature"
k_s_a_dry:=CONDUCTIVITY(Air,T=MST_dry) "thermal conductivity
of air at mean surface temperature"
P_cond:=PRESSURE(R22,T=CT,x=0)          "condensing
pressure"
P_evap:=PRESSURE(R22,T=ST,x=0)         "evaporating
pressure"
mu_ref_1:=VISCOSITY(R22,T=ST,P=P_evap+0.001)    "dynamic
viscosity of liquid refrigerant"
k_ref_1:=CONDUCTIVITY(R22,T=ST,P=P_evap+0.001) "thermal
conductivity of liquid refrigerant"
h_in:=ENTHALPY(R22,T=CT,x=0)           "specific enthalpy of inlet
refrigerant"
h_out:=ENTHALPY(R22,T=ST,x=1)         "specific enthalpy of
outlet refrigerant"
x_in:=QUALITY(R22,h=h_in,T=ST)        "inlet quality of
refrigerant"
h_fg:=ENTHALPY(R22,T=ST,x=1)-ENTHALPY(R22,T=ST,x=0) "latent
heat of fusion"
"Overall heat transfer coefficient-area products (dry-coil)"
hA_o_dry:=C_1*(m_a/mu_b_a_dry)^C_2*(mu_b_a_dry*cp_b_a_dry
/k_b_a_dry)^0.36*((mu_b_a_dry*cp_b_a_dry/k_b_a_dry)/
(mu_s_a_dry*cp_s_a_dry/k_s_a_dry))^0.25*k_b_a_dry
hA_i_dry:=C_3*k_ref_1*((m_ref/mu_ref_1)*(1-x_in)*h_fg)^0.45
UA_tot_dry:=1/(1/hA_i_dry+1/hA_o_dry)
"Capacitance rates"
C_min:=m_a*cp_b_a_dry                "minimum capacitance rate"
NTU_dry:=UA_tot_dry/C_min             "totally dry coil NTU's"

```

```

epsilon_dry:=1-exp(-NTU_dry) "effectiveness of dry portion
coil"
Q_dry:=epsilon_dry*C_min*(EDB-ST) "energy transfer rate for a
totally dry coil"
LDB_new_dry:=EDB-Q_dry/(m_a*cp_b_a_dry)"air outlet
temperature"
IF (abs(LDB_dry-LDB_new_dry)>=0.1) THEN
LDB_dry:=LDB_new_dry
goto 10
ENDIF
LWB_dry:=WetBulb(AirH2O,T=LDB_dry,P=14.7,W=w_i) "wet bulb
temperature of air leaving dry portion of coil"
END

```

```

PROCEDURE WetCoil(Length,m_a,m_ref,EDB,EWB,CT,ST,FPM,C_1,C_2,
C_3:Q_wet,LDB_wet,LWB_wet,hA_o_wet,hA_i_wet)
m_ref:=m_ref*3600
m_a:=m_a*3600
DP_i:=DewPoint(AirH2O,T=EDB,B=EWB,P=14.7) "entering
dewpoint"
LDB_wet:=EWB-20 "assumed leaving dry bulb
temperature"
w_i:=HUMRAT(AirH2O,T=EDB,B=EWB,P=14.7) "entering humidity
ratio"
20: MDB_wet:=(EDB+LDB_wet)/2 "mean dry bulb temperature
for a totally wet coil"
MST_wet:=ST+(MDB_wet-ST)/10 "mean surface temperature
for a totally wet coil"
cp_sat:=(ENTHALPY(AirH2O,T=DP_i,R=1,P=14.7)-ENTHALPY(AirH2O,
T=ST,R=1,P=14.7))/(DP_i-ST) "saturation specific heat"
h_a_i:=ENTHALPY(AirH2O,T=EDB,B=EWB,P=14.7) "specific
enthalpy of entering air"
h_ref_sat_I:=ENTHALPY(AirH2O,T=ST,R=1,P=14.7)
"saturation specific enthalpy at suction temperature"
cp_b_a_wet:=SPECHEAT(Air,T=MDB_wet) "specific
heat"
mu_b_a_wet:=VISCOSITY(Air,T=MDB_wet) "mean bulk
dynamic viscosity of air"
k_b_a_wet:=CONDUCTIVITY(Air,T=MDB_wet) "mean bulk
thermal conductivity of air"
cp_s_a_wet:=SPECHEAT(Air,T=MST_wet) "specific
heat"
mu_s_a_wet:=VISCOSITY(Air,T=MST_wet) "dynamic
viscosity of air at mean surface temperature"
k_s_a_wet:=CONDUCTIVITY(Air,T=MST_wet) "thermal
conductivity of air at mean surface temperature"

```

```

P_cond:=PRESSURE(R22,T=CT,x=0)           "condensing
pressure"
P_evap:=PRESSURE(R22,T=ST,x=0)          "evaporator
pressure"
mu_ref_1:=VISCOSITY(R22,T=ST,P=P_evap+0.001)  "dynamic
viscosity of liquid refrigerant"
k_ref_1:=CONDUCTIVITY(R22,T=ST,P=P_evap+0.001) "thermal
conductivity of liquid refrigerant"
h_in:=ENTHALPY(R22,T=CT,x=0)            "specific enthalpy of inlet
refrigerant"
h_out:=ENTHALPY(R22,T=ST,x=1)           "specific enthalpy of
outlet refrigerant"
x_in:=QUALITY(R22,h=h_in,T=ST)          "inlet quality of
refrigerant"
h_fg:= ENTHALPY(R22,T=ST,x=1)-ENTHALPY(R22,T=ST,x=0) "latent
heat of fusion"
"Overall heat transfer coefficient-area products (wet-coil)"
C_f:=0.626*FPM^0.101                    "air side convection
coefficient correction factor for a totally wet coil"
hA_o_wet:=C_f*C_1*(m_a/mu_b_a_wet)^C_2*(mu_b_a_wet*cp_b_a_wet
/k_b_a_wet)^0.36*((mu_b_a_wet*cp_b_a_wet/k_b_a_wet)
/(mu_s_a_wet*cp_s_a_wet/k_s_a_wet))^0.25*k_b_a_wet
"Btu/hr-ft"
hA_i_wet:=C_3*k_ref_1*((m_ref/mu_ref_1)*(1-x_in)*h_fg)^0.45
UA_h:=1/(cp_sat/hA_i_wet+cp_b_a_wet/hA_o_wet) "Btu/hr-ft"
capAirWet:=m_a
NTU_wet:=UA_h/m_a
epsilon_wet:=1-exp(-NTU_wet)            "effectiveness of wet coil"
Qmax_wet:=m_a*abs(h_a_i-h_ref_sat_i)
h_a_o:=h_a_i-epsilon_wet*Qmax_wet/capAirWet
h_ref_sat_o:=h_ref_sat_i
Q_wet:=m_a*(h_a_i-h_a_o)
capAir:=m_a*(SpecHeat(Air,T=MDB_wet)+w_i*SpecHeat(H2O,
T=MDB_wet))
NTU:=hA_o_wet/capAir
epsilon:=1-exp(-NTU)
h_sat_cond:=h_a_i-(h_a_i-h_a_o)/epsilon
T_cond:=TEMPERATURE(AirH2O,P=14.7,R=1,H=h_sat_cond)
LDB_new_wet:=EDB-(EDB-T_cond)*epsilon
IF (abs(LDB_wet-LDB_new_wet)>=0.1) THEN
LDB_wet:=LDB_new_wet
goto 20
ENDIF
w_o:=(h_a_o-(ENTHALPY(Air,T=LDB_wet)-ENTHALPY(Air,T=0)))
/ENTHALPY(Water,T=LDB_wet,x=1)

```

```
LWB_wet:=WETBULB(AirH2O,T=LDB_wet,P=14.7,w=w_o) "wet bulb
temperature of air leaving wet portion of coil"
END
```

```
PROCEDURE Compare(EDB,EWB,ST,Q_dry,LDB_dry,LWB_dry,hA_o_dry,
hA_i_dry,Q_wet,LDB_wet,LWB_wet,hA_o_wet,hA_i_wet:Q_calc,LDB_ca
lc,LWB_calc,hA_o_calc,hA_i_calc)
DP_i=DEWPOINT(AirH2O,T=EDB,B=EWB,P=14.7) "entering dew
point"
IF (DP_i<=ST) THEN
Q_calc:=Q_dry
LDB_calc:=LDB_dry
LWB_calc:=LWB_dry
hA_o_calc:=hA_o_dry
hA_i_calc:=hA_i_dry
ELSE
Ts_i=(EDB+ST)/2
IF (DP_i<=Ts_i) THEN
IF (Q_dry>Q_wet) THEN
Q_calc:=Q_dry
LDB_calc:=LDB_dry
LWB_calc:=LWB_dry
hA_o_calc:=hA_o_dry
hA_i_calc:=hA_i_dry
ELSE
Q_calc:=Q_wet
LDB_calc:=LDB_wet
LWB_calc:=LWB_wet
hA_o_calc:=hA_o_wet
hA_i_calc:=hA_i_wet
ENDIF
ELSE
Q_calc:=Q_wet
LDB_calc:=LDB_wet
LWB_calc:=LWB_wet
hA_o_calc:=hA_o_wet
hA_i_calc:=hA_i_wet
ENDIF
ENDIF
ENDIF
END
```

```
PROCEDURE unitconvarray1(unitflag$,Q_evap_cat,SHC_cat,
T_db_amb_cat,T_db_1_cat,T_wb_1_cat:Q_evap,SHC,T_db_amb,T_db_1,
T_wb_1)
IF unitflag$='SI' THEN "SI system"
Q_evap:=Q_evap_cat*convert(kW,Btu/s)
```

```

SHC:=SHC_cat*convert(kW,Btu/s)
T_db_amb:=T_db_amb_cat*1.8+32
T_db_1:=T_db_1_cat*1.8+32
T_wb_1:=T_wb_1_cat*1.8+32
ELSE      "English system"
Q_evap:=Q_evap_cat*convert(MBtu/hr,Btu/s)
SHC:=SHC_cat*convert(MBtu/hr,Btu/s)
T_db_amb:=T_db_amb_cat
T_db_1:=T_db_1_cat
T_wb_1:=T_wb_1_cat
ENDIF
END

```

```

PROCEDURE unitconvarray2(unitflag$,V_circ_cat,V_cat_cat,
A_evap_cat,Powerdraw_comp_cat:V_circ,V_cat,A_evap,
Powerdraw_comp)
IF unitflag$='SI' THEN      "SI system"
V_circ:=V_circ_cat*convert(1/s,ft^3/min)
V_cat:=V_cat_cat*convert(1/s,ft^3/min)
A_evap:=A_evap_cat*convert(m^2,ft^2)
Powerdraw_comp:=Powerdraw_comp_cat
ELSE      "English system"
V_circ:=V_circ_cat
V_cat:=V_cat_cat
A_evap:=A_evap_cat
Powerdraw_comp:=Powerdraw_comp_cat
ENDIF
END

```

"Reads information from parameter table:"

```

Rooftop$=LOOKUP$(1,'file_unit')
m=LOOKUP(1,'nu')
unitflag$=LOOKUP$(1,'uu')
n=1
"Calls coil models"
DUPLICATE i=n,m
Call DryCoil (Length,m_a[i],m_r[i],T_db_1[i],T_wb_1[i],
T_3r[i],T_1r[i],C_1,C_2,C_3:Q_dry[i],LDB_dry[i],LWB_dry[i],
hA_o_dry[i],hA_i_dry[i])
Call WetCoil(Length,m_a[i],m_r[i],T_db_1[i],T_wb_1[i],T_3r[i],
T_1r[i],FPM[i],C_1,C_2,C_3:Q_wet[i],LDB_wet[i],LWB_wet[i],
hA_o_wet[i],hA_i_wet[i])
Call Compare(T_db_1[i],T_wb_1[i],T_1r[i],Q_dry[i],LDB_dry[i],
LWB_dry[i],hA_o_dry[i],hA_i_dry[i],Q_wet[i],LDB_wet[i],
LWB_wet[i],hA_o_wet[i],hA_i_wet[i]:Q_ev_ca_hr[i],LDB_calc[i],
LWB_calc[i],hA_o_calc[i],hA_i_calc[i])

```

```

Q_evap_calc[i]=Q_ev_ca_hr[i]*convert(Btu/hr,Btu/s)
END
DUPLICATE i=n,m
Error_evaporator_mean[i]=(Q_evap_catalog[i]
-Q_evap_calc[i])/Q_evaporator_mean
Error_SHC_mean[i]=(SHC_catalog[i]-SHC_calc[i])/SHC_mean
END
SHC_mean=SUM(SHC_catalog[i],i=n,m)/(m-n+1)
Error_SHC=SQRT(SUM((Error_SHC_mean[i])^2,i=n,m)/(m-n+1))
Q_evaporator_mean=SUM(Q_evap_catalog[i],i=n,m)/(m-n+1)
Error_evaporator=SQRT(SUM((Error_evaporator_mean[i])^2,i=n,m)
/(m-n+1))
DUPLICATE i=n,m
Call
CondCoil(T_db_amb[i],T_db_co[i],T_3r[i],m_r[i],m_cond_a[i],
C_4,C_5,C_6:hA_cond_i[i],hA_cond_o[i],Q_co_ca_hr[i],
epsilon_cond[i])
Q_cond_calc[i]=Q_co_ca_hr[i]*convert(Btu/hr,Btu/s)
END
DUPLICATE i=n,m
Error_compressor_mean[i]=(Powerdraw_comp_catalog[i]
-Power_r_comp_kW[i])/Power_catalog_mean
END
Power_catalog_mean=SUM(Powerdraw_comp_catalog[i],i=n,m)
/(m-n+1)
Error_compressor=SQRT(SUM((Error_compressor_mean[i])^2,i=n,m)
/(m-n+1))
"Heat exchanger parameter as a result of optimization:"
C_1=LOOKUP(1,'C_1')      "evaporator outer heat transfer"
C_2=LOOKUP(1,'C_2')      "evaporator outer heat transfer"
C_3=LOOKUP(1,'C_3')      "evaporator inner heat transfer"
C_4=LOOKUP(1,'C_4')      "condenser, outer heat transfer"
C_5=LOOKUP(1,'C_5')      "condenser, outer heat transfer"
C_6=LOOKUP(1,'C_6')      "condenser, inner heat transfer"
"ROOFTOP-MODEL:"
P_amb = 14.7             "ambient pressure"
DUPLICATE i=n,m
CALL unitconvarray1(unitflag$,Q_evap_cat[i],SHC_cat[i],
T_db_amb_cat[i],T_db_1_cat[i],T_wb_1_cat[i]:Q_evap_catalog[i],
SHC_catalog[i],T_db_amb[i],T_db_1[i],T_wb_1[i])
CALL unitconvarray2(unitflag$,V_circ_cat[i],V_cat_cat[i],
A_evap_cat[i],Powerdraw_comp_cat[i]:V_circ[i],V_cat[i],
A_evap[i],Powerdraw_comp_catalog[i])
END
DUPLICATE i=n,m
Q_evap_calc[i]=Q_evap[i]

```

```

Q_cond_calc[i]=Q_cond[i]
Q_evap_cat[i]=LOOKUP(Rooftop$,i,#Q_tot)           "total heat
transfer rate for comparison"
SHC_cat[i]    =LOOKUP(Rooftop$,i,#SHC)
    "sensible heat capacity for comparison"
SHR_catalog[i]=SHC_catalog[i]/Q_evap_catalog[i] "sensible
heat ratio for comparison"
Powerdraw_comp_cat[i]=LOOKUP(Rooftop$,i,#Power) "powerdraw
of all compressors for comparison"
T_db_amb_cat[i]    =LOOKUP(Rooftop$,i,#TDB_amb)
    "ambient drybulb temperature"
T_db_1_cat[i]=LOOKUP(Rooftop$,i,#TDB_i)           "entering
drybulb temperature"
T_wb_1_cat[i]=LOOKUP(Rooftop$,i,#TWB_i)           "entering
wetbulb temperature"
V_circ_cat[i]=LOOKUP(Rooftop$,i,#V_air)           "circulation
air flow rate"
V_cat_cat[i]=LOOKUP(1,'V_dot_cond_fan')           "volume flow
rate of condenser fan"
A_evap_cat[i]=LOOKUP(1,'A_evap')                 "evaporator face area"
END
N_comp=LOOKUP(1,'N_comp')                         "number of
compressors"
"Evaporator (air-side):"
DUPLICATE i=n,m
m_a[i]=rho_1[i]*V_a[i]
rho_1[i]=DENSITY(AirH2O,T=T_db_1[i],B=T_wb_1[i],P=P_1[i])
V_a[i]=V_circ[i]*convert(ft^3/min,ft^3/s)
P_1[i]=P_amb
FPM[i]=V_circ[i]/A_evap[i]                        "air velocity"
END
Length=SQRT(A_evap[n]*convert(ft^2,in^2))
"Condenser (air-side):"
DUPLICATE i=n,m
Q_cond[i]=m_cond_a[i]*c_P_cond[i]*(T_db_co[i]-T_db_amb[i])
c_P_cond[i]=SPECHEAT(Air,T=T_mean_cond_a[i])
m_cond_a[i]=V_cond_a[i]/vol_cond_a[i]
V_cond_a[i]=V_cat[i]*convert(ft^3/min,ft^3/s)
vol_cond_a[i]=VOLUME(Air,T=T_db_amb[i],P=P_amb)
T_mean_cond_a[i]=(T_db_amb[i]+T_db_co[i])/2
END
"Compressor data:"
N_dot=LOOKUP(1,'N_dot')*convert(1/min,1/hr)
V_dot_disp    =LOOKUP(1,'V_dot_disp')*convert(1/min,1/hr)
a=LOOKUP(1,'a')
b=LOOKUP(1,'b')

```

```

C=LOOKUP(1, 'C')
DELTAP=LOOKUP(1, 'DELTAP')
V_disp=V_dot_disp/N_dot
DUPLICATE i=n,m
P_r_suction[i] = P_r_evap[i]*(1-DELTAP)
v_suction[i]=VOLUME(R22,T=T_1r[i],P=P_r_suction[i]-0.1)
"Polytropic exponent:"
k[i]=SPECHEAT(R22,T=T_1r[i],P=P_r_suction[i]-0.1)
/(SPECHEAT(R22,T=T_1r[i],P=P_r_suction[i]-0.1)-R)
"Calculation of the power:"
P_comp[i]=N_comp*(1-k_loss)*P_com_calc[i]
*eta_comb[i]*convert(W,Btu/s)
P_com_calc[i]*eta_comb[i]=m_r_calc[i]*convert(lbm/hr,kg/s)
*(k[i]/(k[i]-1))*P_r_suction[i]*convert(psia,Pa)*(v_suction[i]
*convert(ft^3/lbm,m^3/kg))*((P_r_cond[i]/P_r_suction[i])
^((k[i]-1)/k[i])-1)
eta_comb[i]=a+b/(P_r_evap[i]*convert(psia, atm))
"Volumetric efficiency:"
eta_V[i]=m_r_calc[i]*v_suction[i]/V_dot_disp
"Calculated mass flow rate:"
m_r_calc[i]=(1+C-C*((P_r_cond[i]/P_r_suction[i]))
^(1/k[i]))*V_dot_disp/v_suction[i]
"Output of compressor model:"
Powerdraw_comp[i]=Power_r_comp[i]*convert(Btu/s,kW)
Power_r_comp[i]=N_comp*P_com_calc[i]*convert(W,Btu/s)
Power_r_comp_kW[i]=N_comp*P_com_calc[i]*convert(W,kW)
m_r[i]=N_comp*m_r_calc[i]*convert(lbm/hr,lbm/s)
END
"Ideal gas constant:"
R_m=1.986
R=R_m/M_R22
M_R22=MOLARMASS(R22)
"Refrigeration cycle:"
DUPLICATE i=n,m
"Energy balance:"
Q_evap[i]+P_comp[i]=Q_cond[i]
"Heat transfer at the evaporator:"
Q_evap[i]=m_r[i]*(h_1r[i]-h_4r[i])
"Power relationships for the compressor:"
P_comp[i]=m_r[i]*(h_2r[i]-h_1r[i])
"Enthalpies:"
h_1r[i]=ENTHALPY(R22,T=T_1r[i],x=1)
h_2r`[i]=ENTHALPY(R22,T=T_3r[i],x=1)
h_3r[i]=ENTHALPY(R22,T=T_3r[i], x=0)
h_4r[i]=h_3r[i]
"Temperatures:"

```

```

T_2r[i]=TEMPERATURE(R22,h=h_2r[i],P=P_r_cond[i])
T_2r`[i]=TEMPERATURE(R22,P=P_r_cond[i],x=1)
T_4r[i]=TEMPERATURE(R22,h=h_4r[i],P=P_r_evap[i])
"Pressures:"
P_r_cond[i]=PRESSURE(R22,T=T_3r[i],x=0)"condensing pressure"
P_r_evap[i]=PRESSURE(R22,T=T_1r[i],x=1)"evaporating pressure"
"Calculates sensible capacity:"
T_db_mean[i]=(T_db_1[i]+LDB_calc[i])/2
T_wb_mean[i]=(T_wb_1[i]+LWB_calc[i])/2
c_p_air[i]=SPECHEAT(AirH2o,T=T_db_mean[i],B=T_wb_mean[i],
P=P_amb)
SHC_calc[i]=m_a[i]*c_p_air[i]*(T_db_1[i]-LDB_calc[i])
"Plot variables:"
Q_evaporator_SI[i]=Q_evap[i]*convert(Btu/s,kW)
Q_evaporator_catalog_SI[i]=Q_evap_catalog[i]*convert(Btu/s,kW)
P_compressor_SI[i]=Power_r_comp_kW[i]
P_compressor_catalog_SI[i]=Powerdraw_comp_catalog[i]
SHC_SI[i]=SHC_calc[i]*convert(Btu/s,kW)
SHC_catalog_SI[i]=SHC_catalog[i]*convert(Btu/s,kW)
Q_evaporator_Engl[i]=Q_evap[i]*convert(Btu/s,MBtu/hr)
Q_evaporator_catalog_Engl[i]=Q_evap_catalog[i]*convert(Btu/s,
MBtu/hr)
SHC_Engl[i]=SHC_calc[i]*convert(Btu/s,MBtu/hr)
SHC_catalog_Engl[i]=SHC_catalog[i]*convert(Btu/s,MBtu/hr)
END

```

Appendix I:EES-code for 6-parameter condensing coil model

"Calculates the heat exchanger parameters used for the 6-parameter condensing coil model"

```

$OPENLOOKUP 'parameter.lkt'
$SAVELOOKUP 'parameter.lkt'

PROCEDURE CondCoil(T_db_int,T_2r,T_air_in,T_air_out,T_cond,
m_r,m_a,C_4,C_5,C_6,C_7,C_8,C_9: hA_i,hA_o,Q_cond,epsilon)
m_r:=m_r*3600
m_a:=m_a*3600
P_cond:=PRESSURE(R22,T=T_cond,x=1)
"condensing part:"
c_P:=SPECHEAT(R22,T=T_cond,P=P_cond+0.001)
k_r:=CONDUCTIVITY(R22,T=T_cond,x=0)
mu_r:=VISCOSITY(R22,T=T_cond,x=0)
rho_v_r:=DENSITY(R22,T=T_cond,x=1)
rho_l_r:=DENSITY(R22,T=T_cond,x=0)
Pr:=c_P*mu_r/k_r
h_fg:=ENTHALPY(R22,T=T_cond,x=1)-ENTHALPY(R22,T=T_cond,x=0)
MDB:=(T_air_in+T_db_int)/2 "mean dry bulb temperature"
MST:=T_cond-(T_cond-MDB)/10 "mean surface temperature"
hA_i:=C_6*(rho_l_r*(rho_l_r-rho_v_r)*k_r^3*h_fg/(mu_r*(T_cond-
MST)))^0.25
c_P_b_a:=SPECHEAT(Air,T=MDB) "specific heat"
mu_b_a:=VISCOSITY(Air,T=MDB) "mean bulk dynamic
viscosity of air"
k_b_a:=CONDUCTIVITY(Air,T=MDB) "mean bulk thermal
conductivity of air"
c_P_s_a:=SPECHEAT(Air,T=MST) "specific heat"
mu_s_a:=VISCOSITY(Air,T=MST) "dynamic viscosity of air
at mean surface temperature"
k_s_a:=CONDUCTIVITY(Air,T=MST) "thermal conductivity
of air at mean surface temperature"
hA_o:=C_4*(m_a/mu_b_a)^C_5*(mu_b_a*c_P_b_a/k_b_a)^0.36
*((mu_b_a*c_P_b_a/k_b_a)/(mu_s_a*c_P_s_a/k_s_a))^0.25*k_b_a
UA_tot:=1/(1/hA_i+1/hA_o)
C_min:=m_a*c_P_b_a
NTU:=UA_tot/C_min
epsilon:=1-exp(-NTU)
Q_cond:=epsilon*C_min*(T_cond-T_air_in)
END

```

```

PROCEDURE Super(T_db_int,T_2r,T_air_in,T_air_out,T_cond,m_r,
m_a,C_4,C_5,C_6,C_7,C_8,C_9:hA_i_sup,hA_o_sup,Q_sup,
epsilon_sup)
m_r:=m_r*3600
m_a:=m_a*3600
P_cond:=PRESSURE(R22,T=T_cond,x=1)
"desuperheating"
T_mean_ref_sup:=(T_2r+T_cond)/2
c_P_ref_sup:=SPECHEAT(R22,T=T_mean_ref_sup,P=P_cond)
k_ref_sup:=CONDUCTIVITY(R22,T=T_mean_ref_sup,P=P_cond)
mu_ref_sup:=VISCOSITY(R22,T=T_mean_ref_sup,P=P_cond)
MDB_sup:=(T_db_int+T_air_in)/2
MST_sup:=T_mean_ref_sup*0.95      "mean surface temperature"
c_P_b_a_sup:=SPECHEAT(Air,T=MDB_sup)      "specific heat"
mu_b_a_sup:=VISCOSITY(Air,T=MDB_sup)      "mean bulk
dynamic viscosity of air"
k_b_a_sup:=CONDUCTIVITY(Air,T=MDB_sup)      "mean bulk
thermal conductivity of air"
c_P_s_a_sup:=SPECHEAT(Air,T=MST_sup)      "specific heat"
mu_s_a_sup:=VISCOSITY(Air,T=MST_sup)      "dynamic viscosity of
air at mean surface temperature"
k_s_a_sup:=CONDUCTIVITY(Air,T=MST_sup) "thermal conductivity
of air at mean surface temperature"
hA_o_sup:=C_7*(m_a/mu_b_a_sup)^C_8*(mu_b_a_sup*c_P_b_a_sup
/k_b_a_sup)^0.36*((mu_b_a_sup*c_P_b_a_sup/k_b_a_sup)
/(mu_s_a_sup*c_P_s_a_sup/k_s_a_sup))^0.25*k_b_a_sup
hA_i_sup:=C_9*K_ref_sup*(m_r/mu_ref_sup)^0.8*(mu_ref_sup
*c_P_ref_sup/k_ref_sup)^0.4
UA_sup:=1/(1/hA_i_sup+1/hA_o_sup)
C_air:=m_a*c_P_b_a_sup
C_ref:=m_r*c_p_ref_sup
C_min_sup:=min(C_air,C_ref)
C_max_sup:=max(C_air,C_ref)
C_r_sup:=C_min_sup/C_max_sup
NTU_sup:=UA_sup/C_min_sup
IF C_air>C_ref THEN
epsilon_sup:=(1/C_r_sup)*(1-exp(-C_r_sup*(1-exp(-NTU_sup))))
ELSE
epsilon_sup:=1-exp(-C_r_sup^(-1)*(1-exp(-C_r_sup*(NTU_sup))))
ENDIF
Q_sup:=epsilon_sup*C_min_sup*(T_2r-T_db_int)
END

FUNCTION writelookup(C_4,C_5,C_6,C_7,C_8,C_9)
lookup(1,'C_4')      := C_4
lookup(1,'C_5')      := C_5

```

```

lookup(1, 'C_6')           := C_6
lookup(1, 'C_7')           := C_7
lookup(1, 'C_8')           := C_8
lookup(1, 'C_9')           := C_9
writelookup                :=1
END

dummy=writelookup(C_4,C_5,C_6,C_7,C_8,C_9)
n=1
m=lookup(1, 'nu')
DUPLICATE i=n,m
Call CondCoil (T_db_int[i],T_2r[i],T_air_in[i], T_air_out[i],
T_cond[i], m_r[i], m_a[i],
C_4,C_5,C_6,C_7,C_8,C_9:hA_i[i],hA_o[i],Q_co_calc_hr[i],epsilon
n[i])
Call Super (T_db_int[i],T_2r[i],T_air_in[i], T_air_out[i],
T_cond[i], m_r[i], m_a[i],
C_4,C_5,C_6,C_7,C_8,C_9:hA_i_sup[i],hA_o_sup[i],Q_sup_calc_hr[
i],epsilon_sup[i])
Q_sup_calc[i]=Q_sup_calc_hr[i]*convert(Btu/hr,Btu/s)
Q_cond_calc[i]=Q_co_calc_hr[i]*convert(Btu/hr,Btu/s)
Q_condenser_calc[i]=Q_sup_calc[i]+Q_cond_calc[i]
END
DUPLICATE i=n,m
T_air_in[i]=LOOKUP(i, 'T_air_in')
T_air_out[i]=LOOKUP(i, 'T_air_out')
T_cond[i]=LOOKUP(i, 'T_cond')
m_r[i]=LOOKUP(i, 'm_r')
m_a[i]=LOOKUP(i, 'm_a')
Q_condenser[i]=LOOKUP(i, 'Q_cond')
P_cond[i]=PRESSURE(R22, T=T_cond[i], x=1)
h_2r[i]=ENTHALPY(R22, T=T_2r[i], P=P_cond[i])
h_2r`[i]=ENTHALPY(R22, T=T_cond[i], x=1)
Q_sup[i]=m_r[i]*(h_2r[i]-h_2r`[i])
Q_cond[i]=Q_condenser[i]-Q_sup[i]
Error_sup_mean[i]=(Q_sup[i]-Q_sup_calc[i])/Q_sup_mean
Error_cond_mean[i]=(Q_cond[i]-Q_cond_calc[i])/Q_cond_mean
Error_condenser_mean[i]=(Q_condenser[i]-
Q_condenser_calc[i])/Q_condenser_mean
T_2r[i]=LOOKUP(i, 'T_2r')
T_db_int[i]=LOOKUP(i, 'T_db_int')
END
Error_sup=SQRT(SUM((Error_sup_mean[i])^2, i=n,m)/(m-n+1))
Error_cond     =SQRT(SUM((Error_cond_mean[i])^2, i=n,m)/(m-
n+1))

```

```
Error_condenser=SQRT(SUM((Error_condenser_mean[i])^2,i=n,m)
/(m-n+1))
RMS_condenser=SQRT(SUM((Q_cond[i]-Q_cond_calc[i])^2,i=n,m)/(m
-n+1))
lower_Q=MIN(Q_condenser[1..m])/Q_condenser_mean
upper_Q=MAX(Q_condenser[1..m])/Q_condenser_mean
Q_sup_mean      =SUM(Q_sup[i],i=n,m)/(m-n+1)
Q_cond_mean=SUM(Q_cond[i],i=n,m)/(m-n+1)
Q_condenser_mean=SUM(Q_condenser[i],i=n,m)/(m-n+1)
DUPLICATE i=n,m
"Calculation of plot variables:"
Q_condenser_plot[i]=Q_condenser[i]*convert(Btu/s,MBtu/hr)
Q_condenser_calc_plot[i]=Q_condenser_calc[i]*convert(Btu/s,
MBtu/hr)
Q_condenser_SI_plot[i]=Q_condenser[i]*convert(Btu/s,kW)
Q_condenser_calc_SI_plot[i]=Q_condenser_calc[i]*convert(Btu/s,
kW)
END
```

Appendix J: Use of the EES-program RTPU

The RTPU (**R**ooftop **P**ackaged **U**nits) program calculates the performance prediction of rooftop packaged air conditioning units based on manufacturer's catalog data. Manufacturer's catalog data have to be available for the rooftop packaged unit and for the compressor used in the unit. The performance data for the equipment are stored in a lookup table, which is accessed from the EES-program. To start RTPU, run the file "RTPU.EES".

Having executed the file "RTPU.EES" a calculation can be started by pushing the start-button. Before starting the calculations, however, two lookup-tables have to be created in which the rooftop packaged air conditioning performance data and the compressor data are stored for the fitting process. The compressor lookup table contains the following information:

Input	Column name	English units	SI-units
Evaporating temperature	T_evaporator	°F	°C
Condensing temperature	T_condenser	°F	°C
Mass flow rate	m_r	lbm/hr	kg/hr
Capacity	Capacity	Btu/hr	kW
Compressor power draw	Power	kW	kW

The required input is the compressor power draw either the capacity or the mass flow rate and the referring evaporating and condensing temperatures. These data can be input either in English units or in SI-units.

The lookup table for the rooftop packaged unit contains the following information:

Input	Column name	English units	SI-units
Entering dry bulb temperature	TDB_i	°F	°C
Entering wet bulb temperature	TWB_i	°F	°C
Ambient temperature	TDB_amb	°F	°C
Volume flow rate	V_air	ft ³ /min	l/s
Capacity	Q_tot	MBtu/hr	kW
Sensible capacity	SHC	MBtu/hr	kW
Power draw	Power	kW	kW

Having created both lookup tables, the program can be started by pushing the start button. First the compressor mass flow rate model is loaded. The number of data points given in the lookup table, the motor speed, the displacement rate, and the rating conditions are required input in the diagram window for calculating the compressor mass flow rate parameters. Also the file path and name of the lookup table where the compressor performance data are stored has to be provided. The unit system, in which the compressor performance data are given, has to be selected and the user can select whether the compressor performance is given as cataloged mass flow rate values or capacity data. Having provided the required information the optimization for the parameters C and Δp can be started by hitting the optimize button. The results of the optimization will be displayed in the output box. A screen shot of the EES-diagram window for the mass flow rate compressor model is shown on the next page.

COMPRESSOR MODEL

Input:

Number of data points:

Motor speed: [1/min]

Total displacement rate: [l/s, ft³/min]

Rating conditions:

Subcooling: [C, F]

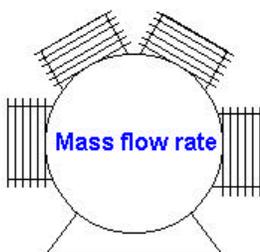
Superheat: [C, F]

Compressor data:

Unit system:

Type of compressor data:

English: m, [lbm/hr], capacity [Btu/hr], power [kW]
SI: m, [kg/hr], capacity [kW], power [kW]



File menu:

$$\eta_v = \frac{\dot{m}_{act}}{\dot{m}_{th}} = \left(1 + C - C \cdot \left(\frac{p_{dis}}{p_{suc}} \right)^{1/n} \right) \cdot \frac{v_{suction}}{v_a} \cdot \frac{RPM}{RPM}$$

$$p_{suc} = p_{evap} \cdot (1 - \Delta p)$$

→

→

Output:

C=0.0706

Δp=0.09419

Error_{mass}=0.01955

Data range:

min/mean = 0.3585

max/mean = 1.742

EES-diagram window for the compressor mass flow rate model

By hitting the “Next” button the program is called, that is used for determining the coefficients for the compressor power draw prediction. There is no further information required and the calculation can be started by hitting the “Optimize” button. Having finished the calculations, the parameters and the error are displayed in the output box. In addition, the fitted data points (mass flow rate and power) can be viewed by accessing the plot menu. The next program creates the data set for the heat exchanger parameter fitting procedures. Before starting the calculation, the unit system that is used in the lookup table with the rooftop packaged unit performance data has to be specified. Also, the number of compressors used in the packaged unit has to be provided. In addition, the face area of the evaporator coil and the

condenser volume flow rate are required input. The file path and the name of the lookup file have to be provided as well as the number of data points given in the lookup table. Next, the calculation can be started, and the data set used for determining the heat exchanger parameters is calculated. The following two programs are used to determine the heat exchanger parameters for the evaporator and the condenser. There is no additional input necessary to run these programs. Having determined the heat exchanger parameters, the following program calculates the performance for the data points used in the fitting process and compares them with the catalog data points.

by Mark Reichler
Solar Energy Lab
University of Wisconsin
1999




Plot Menu:

Capacity (total) [MBtu/hr]

Capacity (sensible) [MBtu/hr]

Capacity (total) [kW]

Capacity (sensible) [kW]

Compressor power draw [kW]

File Menu:

Next >>

<< Back

Main Menu

Model Validation:

Lookup file with unit data:
unit15_Tdb.lkt

Number of data points: 8

Unit system: English

Model Parameters:

Evaporator:	Condenser:	Compressor:
C ₁ =1026	C ₄ =987.9	C=0.0706
C ₂ =0.5	C ₅ =0.5	ΔP=0.09419
C ₃ =1909	C ₆ =31191	a=0.716
		b=-0.1905

Mean normalized RMS-errors:

Compressor power draw:	Capacity:
Error _{power} = 0.01869	Error _{total} = 0.02997
	Error _{sensible} = 0.02776

EES-diagram window showing the results of the fitting process

The window displays the calculated model parameters for the evaporator, for the condenser and the compressor. Also, the mean weighted RMS-errors are displayed as a result of performance prediction. The plot menu allows selecting plots for the sensible heat capacity, the total capacity and the compressor power draw prediction. By hitting the next button the program for calculating the performance under different operating conditions is started. For each operating point the supply flow rate, the air entering temperature (dry and wet bulb), and the ambient air has to be specified. Then, the program calculates the sensible capacity, total capacity, and the compressor power draw for this operating point.

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