
A Generalized Simulation Model of Chillers and Heat Pumps to Be Calibrated on Published Manufacturer's Data

Vincent LEMORT^{1*} and Stéphane BERTAGNOLIO²

^{1,2} Thermodynamics Laboratory
University of Liège
Campus du Sart Tilman, B49
B-4000 Liège, Belgium

E-mail: vincent.lemort@ulg.ac.be

*Corresponding Author

ABSTRACT

This paper shows how semi-empirical models of chillers and heat pumps can be calibrated on the basis of manufacturer's published data, which is most of the time the only available source of information. A calibration methodology is proposed and illustrated for different technologies of vapor-compression machines, pointing out tips and tricks. Whatever the size and technology (e.g. scroll, screw or reciprocating compressors), the semi-empirical model is found to predict both full load and part load performance with a very good accuracy. This is partially due to the tailor-made formalism of the model, which allows describing the main features of the machine.

1. INTRODUCTION

Accurate simulation of chillers and heat pump systems is welcome at different stages of HVAC system life cycle, such as design, evaluation, commissioning and control. Different simulation models can be considered and are generally classified between empirical (Hydeman et al., 2002; Morisot and Marchio, 2002), semi-empirical (Bourdhoux, 1994) and deterministic models. Semi-empirical models, which are based on a physical description of the machine, allow extrapolating the performance for different operating conditions or modification of the design, the control, the refrigerant or the size of the machine. Moreover, they require a limited number of parameters that can be identified on the basis of information published by the manufacturers, when expensive and time-consuming experimental approach cannot be implemented. The objective of this paper is to show how this information can be used to calibrate a simulation model of a chiller/heat pump. A methodology will be defined and illustrated for different technologies of machines.

2. MODELING A VAPOR-COMPRESSION CHILLER/HEAT PUMP

2.1. General approach

The chiller/heat pump model associates the sub-models of a compressor, an evaporator and a condenser (Jin and Spitler, 2002a and 2002b). It can be conventionally described by the information flow diagram presented, in the special case of an air-cooled chiller, in Figure 1 and already described by the authors (Lemort et al., 2009).

2.2. Modeling the compressor

The compressor is one of the key components of the chiller/heat pump: its performance will strongly impact on that of the whole machine, which justifies modeling it with an appropriate degree of details. Different positive displacement technologies are used according to the capacity of the machine. This paper will focus on scroll, (double and single) screw and reciprocating compressors.

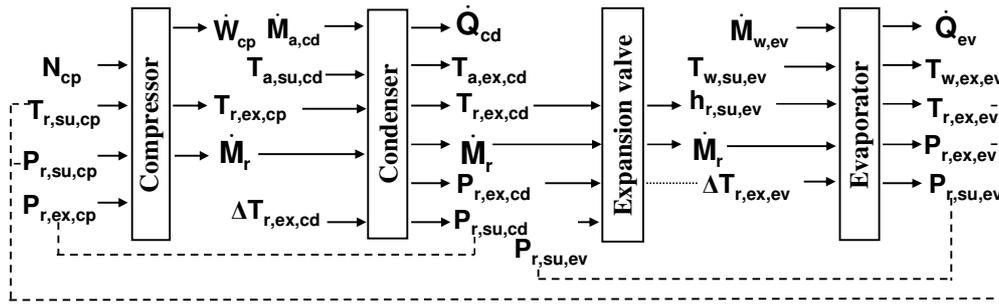


Figure 1: Schematic representation of an air-cooled chiller model

Scroll and screw compressors

Scroll and screw compressors are described in a very similar way, on the basis of the simulation model proposed by Winandy (2002). The modeling accounts for the built-in volume ratio (parameter $r_{v,in}$) and the internal leakages (lumped into one unique flow computed by introducing a fictitious leakage area A_{leak}). Suction and discharge heat transfers, as well as the heat transfer between the compressor and the ambient, are described on the basis of three heat transfer coefficients (AU_{su} , AU_{ex} and AU_{amb}). Electro-mechanical losses are split into constant losses ($\dot{W}_{loss,0}$) and losses proportional to the internal compression power (factor α). If not negligible, suction and discharge pressure losses can be introduced in the modeling, which requires two additional parameters (d_{su} and d_{ex}). The modeling assumes that the fluid undergoes the following consecutive steps: supply heating-up (su \rightarrow su,1); mixing with the internal leakage (su,1 \rightarrow su,2); adiabatic and reversible compression (su,2 \rightarrow in); adiabatic compression at a constant machine volume (in \rightarrow ex,1); exhaust cooling-down (ex,1 \rightarrow ex)

Reciprocating compressors

Reciprocating compressors do not present a built-in volume ratio, so that the entire compression process (su,2 \rightarrow ex,1) can be considered as fully isentropic. Another difference with the two previous machines is the presence of a clearance volume, whose re-expansion will limit the refrigerant volume flow rate swept by the machine. The internal mass flow rate can be calculated by:

$$\dot{M}_{in,cp} \cdot v_{r,su3,cp} = \dot{V}_{s,cp} - C_{cp} \cdot \dot{V}_{s,cp} \cdot \left(\frac{v_{r,su3,cp}}{v_{r,ex2,cp}} - 1 \right) \quad (1)$$

2.3. Modeling the heat exchangers

The basic modeling of the evaporator and the condenser consists in assuming that the heat exchanger is semi-isothermal, with the constant temperature equal to the saturation temperature. The single-phase zones of the heat exchanger are neglected and the model reduces to a one-zone heat exchanger. This assumption is acceptable for the evaporator of a chiller, since it presents a large two-phase zone and a small single-phase (superheating) zone.

In order to be more accurate in the condenser modeling, an average condensing temperature \bar{T}_{cd} can be defined as the weighted average of the actual temperatures occurring in the three zones (single-phase desuperheating, two-phase condensation and single-phase undercooling) (Lemort et al., 2009).

For both heat exchangers, the overall heat transfer coefficient AU is computed by associating 2 convective heat transfer resistances in series: R_{sf} , and R_r . The evaporating and condensing powers are computed by using the ϵ -NTU method. For the condenser, it gives:

$$\dot{Q}_{cd} = \left(1 - \exp\left(\frac{-AU_{cd}}{\dot{C}_{sf,cd}}\right) \right) \dot{C}_{sf,cd} (\bar{T}_{cd} - T_{sf,su,cd}) \quad (2)$$

3. CALIBRATION OF THE MODEL BASED ON PUBLISHED MANUFACTURER DATA

3.1. Calibration methodology

Calibration of the chiller/heat pump simulation model is carried out in 2 steps (Figure 2). First, the parameters of the compressor model are identified based on the information provided by the compressor manufacturer. Then, the parameters of the other components are identified based on chiller/heat pump manufacturer data. The chiller/heat pump manufacturer only seldom gives precise information regarding the brand of compressor used in the machine. Hence, it is rather difficult to associate data published by both the compressor and the chiller manufacturers for a given chiller/heat pump. In that case, it is recommended to select a similar compressor “off-the-shelf”, of which performance is published, in order to estimate first guesses of the parameters of the compressor model. These parameters can be updated afterwards in order to best predict the performance of the entire machine.

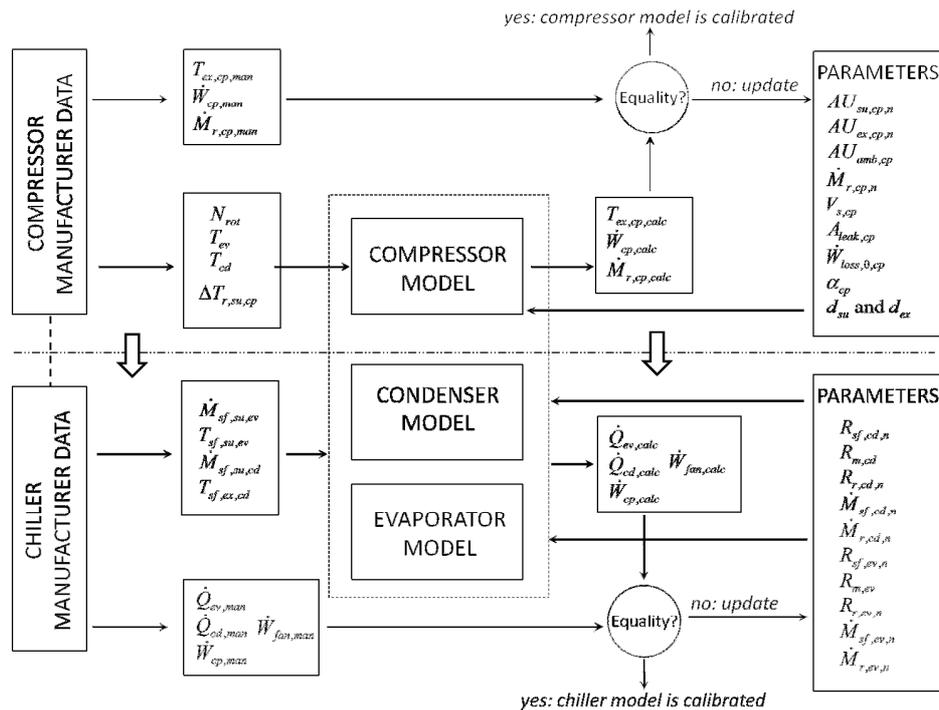


Figure 2: Chiller/heat pump calibration flow chart

The calibration methodology can be achieved in 4 steps:

- 1) *Analysis of manufacturer's submittal.* The main physical features of the machine are collected (type of condenser cooling, number and types of compressors,...). Performance points of the chiller/heat pump are retrieved. If it is possible, the brand and model of compressor are identified and performance points are retrieved. The major peculiarities of the machine are identified. The simulation model must be adapted in consequence.
- 2) *Generation of default guesses for the parameters.* The model is first “tested” with default guesses. The latter are determined on the basis of the modeler’s experience. They could also be easily assessed with rational assumptions (such as a 65% efficiency for a heat exchanger). Extrapolation for different sizes/capacities could also be considered.
- 3) *Manual calibration.* This calibration consists in carrying out a sensitivity analysis on each parameters, which quickly allows identifying the most influencing parameters and largely improve the model’s quality.
- 4) *Automatic calibration.* This step consists in implementing an algorithm optimizing an objective function. This step allows refining the parameters of the simulation model. The lower and upper bounds for the search of parameters are determined through the 3rd step, which guarantees realism in the choice of bounds.

3.2. Machines selected for calibration

Calibration based on manufacturer data will be illustrated hereunder for machines listed in Table 1. They differ one from the other by the type of compressor, the fluids used for cooling/heating the condenser/evaporator, the nominal cooling/heating capacities and the type of capacity modulation. The heat pump heating capacity is defined for brine entering temperature of 0°C and water leaving temperature of 35°C. Cooling capacity of

chillers is rated for a chilled water leaving temperature of 7°C and a cooling air/water entering temperature of 35/30°C.

Table 1: Series of vapor compression machines considered in this study

	Compressor	Condenser	Evaporator	Cooling/heating capacity	Regulation	Refrigerant
PAC-SCRO-BRI-WAT	Scroll	Water-cooled	Brine-heated	10.2	1 compressor ON/OFF	R407C
CH-SCRO-AIR	Scroll	Air-cooled micro-channel	Direct expansion Water-heated shell and tubes	346	4 to 6 compressors ON/OFF	R410a
CH-SCRO-WAT	Scroll	Water-cooled shell and tubes	Direct expansion Water-heated shell and tubes	286	4 to 6 compressors ON/OFF	R410a
CH-SCRE-AIR	Twin-screw	Air-cooled micro-channel	Flooded water heated shell and tube	293	2 compressors Slide valve	R134a
CH-SCRE-WAT	Single-screw	Water-cooled shell and tubes	Direct expansion water-heated shell and tubes	160-480	2 compressors Slide valve	R134a
CH-RECI-AIR	Reciprocating	Air-cooled Tubes and fins	Direct expansion water- heated shell and tubes	486	5 compressors ON/OFF	R22

Information provided by the chiller/heat pump's manufacturer consists of full load performance points (heating/cooling capacity, compressor and fan electrical consumption) as function of secondary fluid temperatures. Part load performance points are seldom provided. As far as possible, chiller/heat pump manufacturers were asked to provide information allowing identifying the compressor brand and model. Information published by the compressor manufacturer consists of the machine displacement, the displaced refrigerant mass flow rate (or the cooling capacity for a liquid subcooling at the condenser exhaust and vapor superheating at the evaporator exhaust) and the compressor electrical consumption as function of the evaporating and condensing temperatures. In some cases, the heat rejection at the condenser is provided, which allows estimating the compressor discharge temperature.

3.3. Validation of the models

Brine-to-water heat pump with a scroll compressor

In the case of the heat pump, it was possible to retrieve the brand and the model of scroll compressor. The classical model of compressor was not able to predict correctly the performance of the compressor for high pressure ratios. It might be due to the presence of a discharge reed valve. The discharge valve is particularly important for systems working with pressure ratios of 6-8 or higher. In this case, the internal pressure ratio is significantly below the compressor operating pressure ratio, resulting in an excessive compression work due to gas back flow and recompression (Elson et al., 2008). The scroll compressor model proposed by Winandy et al. (2002) has been adapted to account for the presence of the valve by modifying the description of the internal compression process. In the case of under-compression (Figure 3 (b)), the internal compression is now described by an adiabatic and reversible compression ($su,2 \rightarrow in$), followed by an adiabatic compression at a constant machine volume ($in \rightarrow in^*$) and an adiabatic and reversible compression ($in^* \rightarrow ex,2$). The distinction between the two later compression processes is due to the presence of a residual clearance volume ($C.V_{s,cp}$) underneath the reed valve and in which compressed gas is discharged when the compression chambers open.

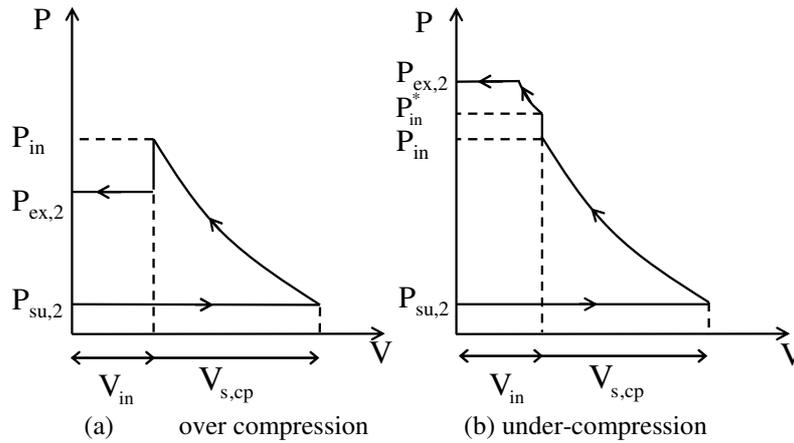


Figure 3: Representation of the entire compression process in the P-V diagram

Hence, in the case of under-compression, the total internal compression work is given by:

$$w_{in} = w_{in,1} + w_{in,2} = w_{in,1} + w_{in,2,v} + w_{in,2,s} \quad (3)$$

The work fictitiously associated to the back-flow process is given by:

$$w_{in,2,v} = v_{in} \cdot (P_{in}^* - P_{in}) \quad (4)$$

The corrected internal pressure P_{in}^* can finally be computed on the basis of the specific volume v_{in}^* and the internal energy u_{in}^* , respectively given by Equations (5) and (6), derived from mass and energy conservation with the merging of the compression chamber and the clearance volume.

$$v_{in}^* = \frac{V_{in}^*}{M_{in}^*} = \frac{1/r_{v,in} + C}{1/v_{su,2} + C/v_{ex,2}} \quad (5)$$

$$\frac{C}{v_{ex,2}} \cdot u_{ex,2} + \frac{1}{v_{su,2}} u_{in} = \left(\frac{C}{v_{ex,2}} + \frac{1}{v_{su,2}} \right) \cdot u_{in}^* \quad (6)$$

After that mixing process and as the discharge process continues, the volume of the region made up of the discharge chamber and the discharge region decreases and the corresponding pressure increases until the pressure $P_{ex,2}$ is reached. This evolution is assumed to be adiabatic and reversible. Consequently, the associated work may be expressed as:

$$w_{in,2,s} = h_{ex,2}(P_{ex,2}, s_{in}^*) - h_{in}^*(P_{in}^*, v_{in}^*) \quad (7)$$

Figure 4 compares the errors on the prediction of the electrical consumption of the chiller for the original and modified models. The original model tends to over-predict the power consumption for large pressure ratios. The validity of the calibrated compressor model can be estimated in Figure 5, comparing the volumetric and global isentropic efficiencies predicted by the model and announced by the manufacturer. Identified parameters are given in Table 2.

The parameters of both the brine-heated evaporator and water-cooled condenser were identified considering the published performance of the heat pump. Assumptions were introduced regarding the secondary fluid temperature differences over the heat exchangers. It was observed that the modified model accounting for the reed valve doesn't yield a better accuracy in comparison with the classical model. According to the heat pump simulation model, pressure ratios associated to the performance points never exceed 5.5. For such pressure ratios, the potential of improvement of the modified scroll compressor model (Figure 4) is limited and probably within the accuracy of the performance points provided by the heat pump manufacturer. Validity of the model is shown in Figure 6.

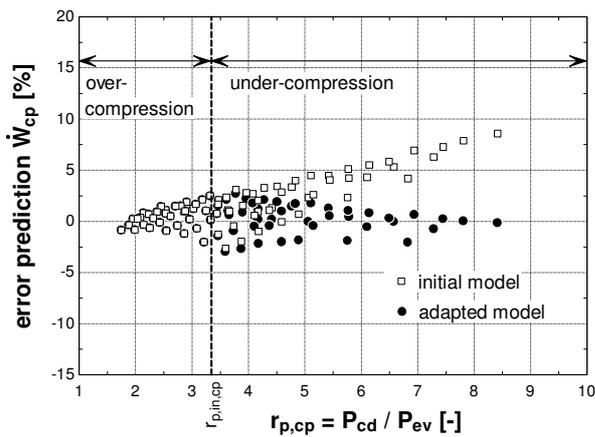


Figure 4: Error on the prediction of the compressor electrical consumption (SCRO-BRI-WAT)

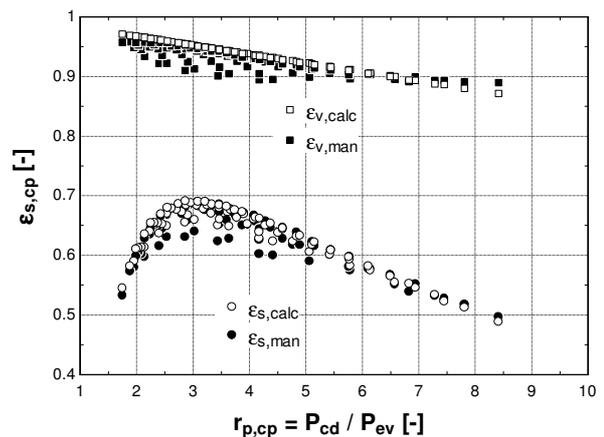


Figure 5: Prediction of the volumetric and global isentropic efficiencies (SCRO-BRI-WAT)

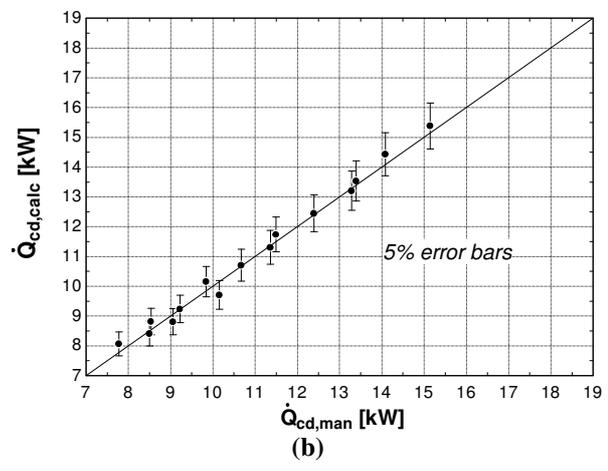
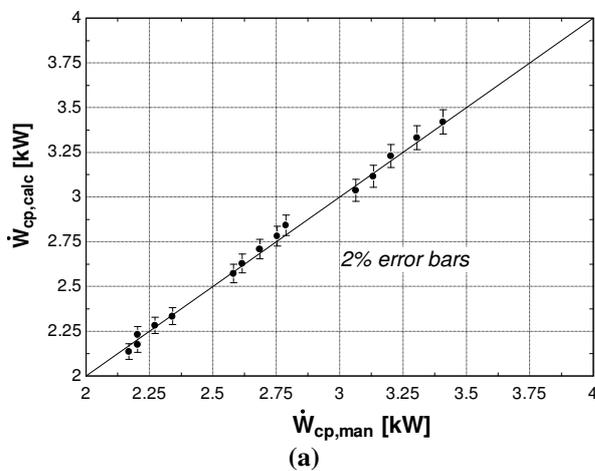


Figure 6: Prediction of (a) the full load compressor consumption (b) the heating capacity (PAC-SCRO-BRI-WAT)

Air- and water-cooled chillers with scroll compressors

For these two types of machines, performance of the compressor was not available. Consequently, the compressor model has to be calibrated on the basis of the chiller's performance only. As initial guesses, values of parameters related to a compressor with the same technology, showing a similar capacity and already characterized were considered. The major modeling assumptions consisted in merging the two refrigerant circuits and the different condensers into one circuit and one condenser.

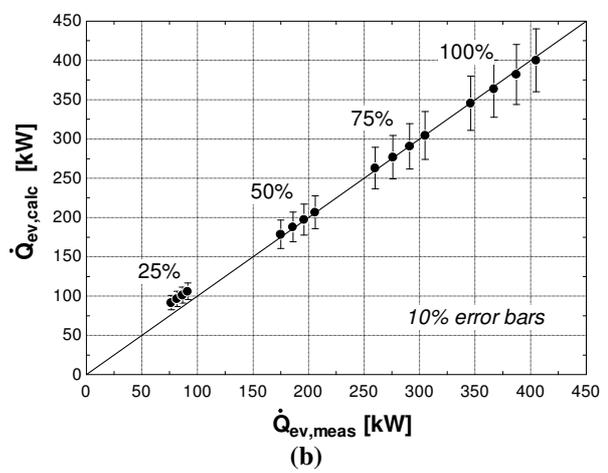
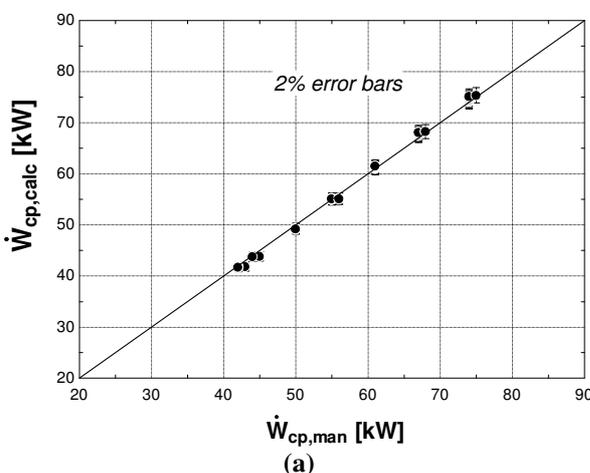


Figure 7: Prediction of (a) the full load compressor consumption (b) the part load cooling power (CH-SCRO-WAT) Figure 7(a) compares the valued predicted by the model and announced by the manufacturer for the compressor consumption. Figure 7(b) shows the capacity of the air-cooled scroll-compressor chiller model to predict the part load performance points. Except for the lowest capacity stage, the model appears to predict the performance with a very good accuracy. For the chiller electrical consumption, the best agreement was found when varying the fan power consumption in proportion with the number of compressors in use.

Table 2: Identified parameters of the machines (values in brackets correspond to the calibration based on compressor performance points)

			PAC-SCRO- BRI-WAT	CH-SCRO- AIR-1	CH-SCRO- WAT-1	CH-SCRE- AIR-1	CH-SCRE- WAT-1	CH-RECI-AIR- 1
COMPRESSOR	N_{cp}		1	4	4	2	2	5
	rpm_{cp}	[rpm]	3000	3000	3000	3000	3000	1450
	$V_{s,cp}$	[cm ³]	60.04	342	242	(2548) 1566	1476	(1026) 1378
	C	[-]	(0.75) 2	-	-	-	-	0.045
	$r_{v,in,cp}$	[-]	(3.075) 2.8	2.55	2.55	4	2.5	-
	$r_{v,in1,cp}$	[-]	-	-	-	1.2	-	-
	$\dot{M}_{r,cp,n}$	[kg/s]	0.091	0.3572	0.3572	0.3572	0.3572	0.3572
	$AU_{su,cp,n}$	[W/K]	30	50	50	50	30	15
	$AU_{ex,cp,n}$	[W/K]	20	10	13.5	20	20	15
	$AU_{amb,cp}$	[W/K]	10	10	10	(50) 100	30	15
	$A_{leak,cp}$	[mm ²]	0.45	0.35	0.35	(22.5) 12	3.5	(3.33) 0
	α	[-]	(0.20) 0.23	0.3	0.3	(0.24) 0.26	0.51	(0.29) 0.32
	$\dot{W}_{loss,0,cp}$	[W]	120	900	800	3000	0	(1000) 1800
	d_{su}	[mm]	-	-	-	-	-	(38.3) 37
d_{ex}	[mm]	5.6	-	-	-	-	(19) 15	
CONDENSER	$\dot{M}_{r,cd,n}$	[kg/s]	2.143	2.143	2.143	3.011	2.143	2.143
	$\dot{M}_{sf,cd,n}$	[kg/s]	16.19	43.44	16.19	40.22	16.19	47.02
	$R_{r,cd,n}$	[K/W]	1.009 10^{-5}	1.085 10^{-5}	5.551 10^{-6}	8.491 10^{-6}	6.6616 10^{-6}	1.11 10^{-5}
	$R_{sf,cd,n}$	[K/W]	1.211 10^{-5}	1.302 10^{-5}	6.662 10^{-6}	8.491 10^{-6}	6.662 10^{-6}	1.665 10^{-5}
EVAPORATOR	$\dot{M}_{r,ev,n}$	[kg/s]	2.143	2.143	2.143	2.731	2.143	2.143
	$\dot{M}_{sf,ev,n}$	[kg/s]	16.44	16.44	16.44	18.58	16.44	16.44
	$R_{r,ev,n}$	[K/W]	1.237 10^{-5}	5.556 10^{-6}	4.451 10^{-6}	3.447 10^{-6}	7.048 10^{-6}	5.688 10^{-6}
	$R_{sf,ev,n}$	[K/W]	1.855 10^{-5}	6.677 10^{-6}	6.677 10^{-6}	3.447 10^{-6}	7.048 10^{-6}	8.532 10^{-6}
	AU_{econ}	[W/K]	-	-	-	3400	-	-

Air-cooled chiller with an economizer and double-screw compressors

The considered chiller has the peculiarity to have an economizer in the cycle, as shown in Figure 8. For this machine, the calibration exploited both the compressor and the chiller data. Compressor data were extracted from the selection software of a manufacturer and were particularly detailed, consisting in the compressor supply and exhaust temperatures, the evaporating and condensing temperatures, the saturation temperature in the economizer, the compressor electrical consumption, the mass flow rate through the condenser and through the evaporator as well as the thermal powers exchanged at the evaporator, the condenser and the economizer.

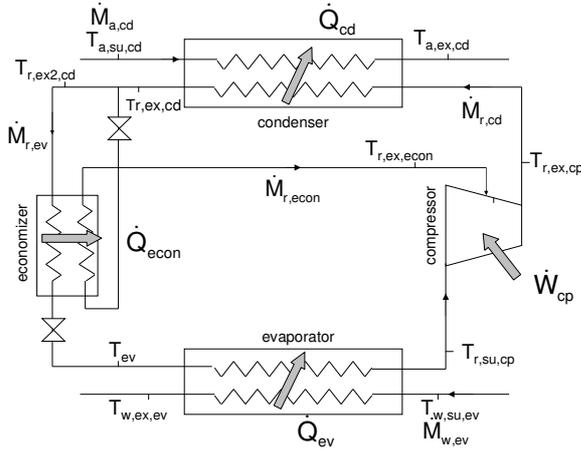


Figure 8: Schematic representation of a vapor compression cycle with an economizer

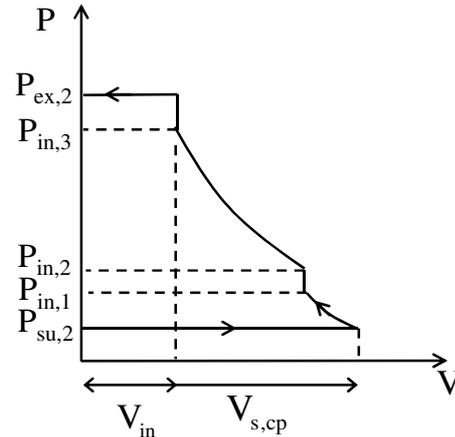


Figure 9: Representation of the compression process in the P-V diagram (under-compression)

The classical screw compressor model was modified to account for the vapor injection. This injection is assumed to occur instantaneously after a volume reduction of $r_{v,in1,cp}$ (Figure 9).

The internal built-in volume ratio of the machine is the product of both built-in volume ratios

$$r_{v,in,cp} = r_{v,in1,cp} \cdot r_{v,in2,cp} \tag{8}$$

The pressure at the end of the injection process is computed by expressing the energy balance over the compression chamber at the time of the injection.

$$\dot{M}_{in} \cdot u_{in1,cp} + \dot{M}_{econ} \cdot h_{ex,econ} = (\dot{M}_{in} + \dot{M}_{econ}) \cdot u_{in2,cp} \tag{9}$$

In the former expression, the second term is the enthalpy flow rate injected into the compressor. The pressure is a function of the specific internal energy and volume.

$$P_{in2,cp} = f(u_{in2,cp}, v_{in2,cp}) \quad \text{with} \quad v_{in2,cp} = \frac{\dot{V}_{s,cp}}{r_{v,in1,cp} \cdot (\dot{M}_{in} + \dot{M}_{econ})} \tag{10}$$

The internal compression power is finally given by:

$$\dot{W}_{in,cp} = \dot{M}_{s,cp} \cdot (w_{in1,cp} + w_{in2,cp}) + (\dot{M}_{s,cp} + \dot{M}_{econ}) \cdot (w_{in3,cp} + w_{in4,cp}) \tag{11}$$

with the different compression works computed by:

$$\begin{aligned} w_{in1,cp} &= h_{in1,cp} - h_{su2,cp} ; w_{in2,cp} = v_{in1,cp} \cdot (P_{in2,cp} - P_{in1,cp}) \\ w_{in3,cp} &= h_{in3,cp} - h_{in2,cp} ; w_{in4,cp} = v_{in3,cp} \cdot (P_{ex2,cp} - P_{in3,cp}) \end{aligned} \tag{12}$$

In this validation, the presence of oil in the refrigerant was not considered. It is probably implicitly taken into account in the estimation of the heat transfers coefficients. It can be observed hereunder that the model is able to predict with a very good accuracy the exhaust temperature (Figure 10 (a)) and the volumetric and isentropic efficiencies (Figure 10 (b)).

The chiller comprises two screw compressors characterized by different displacements. For the purpose of the modeling, it was assumed that the two compressors have equal displacements (average value of the actual ones). As indicated in Table 2, values of some parameters were slightly tuned in order for the chiller model to predict the performance with a better accuracy. Results of the calibration are shown in Figure 11.

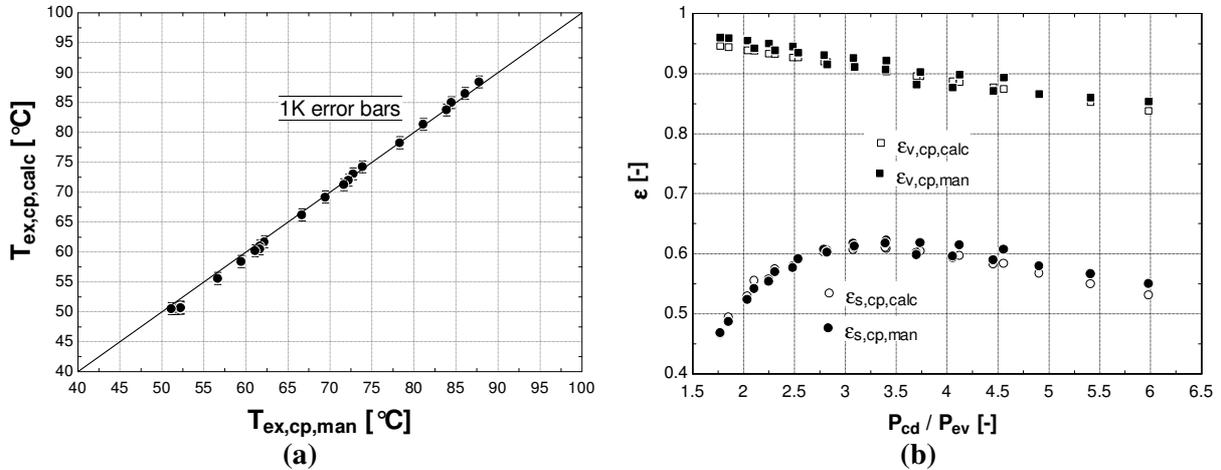


Figure 10: Prediction of (a) the exhaust temperature; (b) the volumetric and isentropic efficiencies (SCRE-AIR)

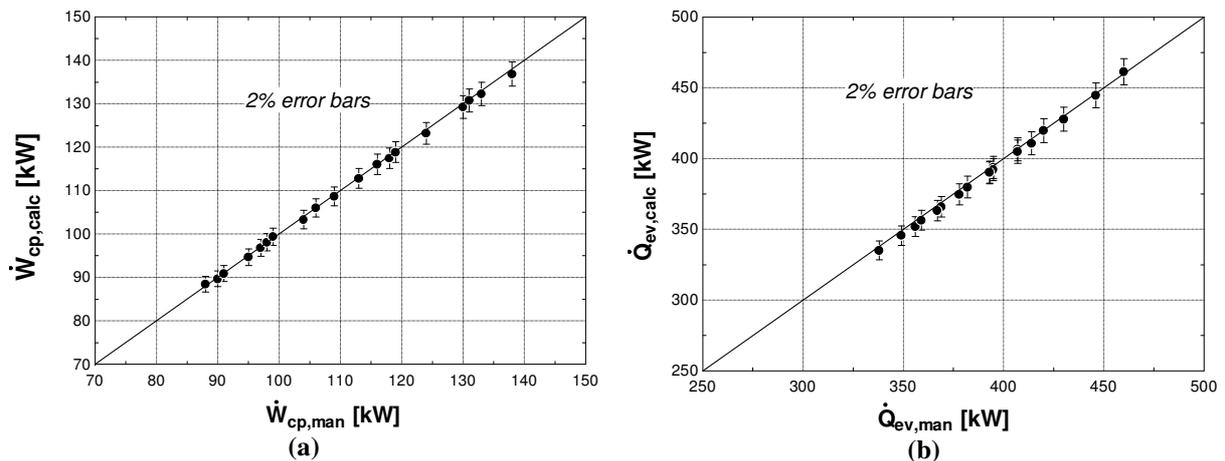


Figure 11: Prediction of (a) the full load compressor consumption; (b) the cooling capacity (CH-SCRE-AIR)

Air-cooled chiller with reciprocating compressors

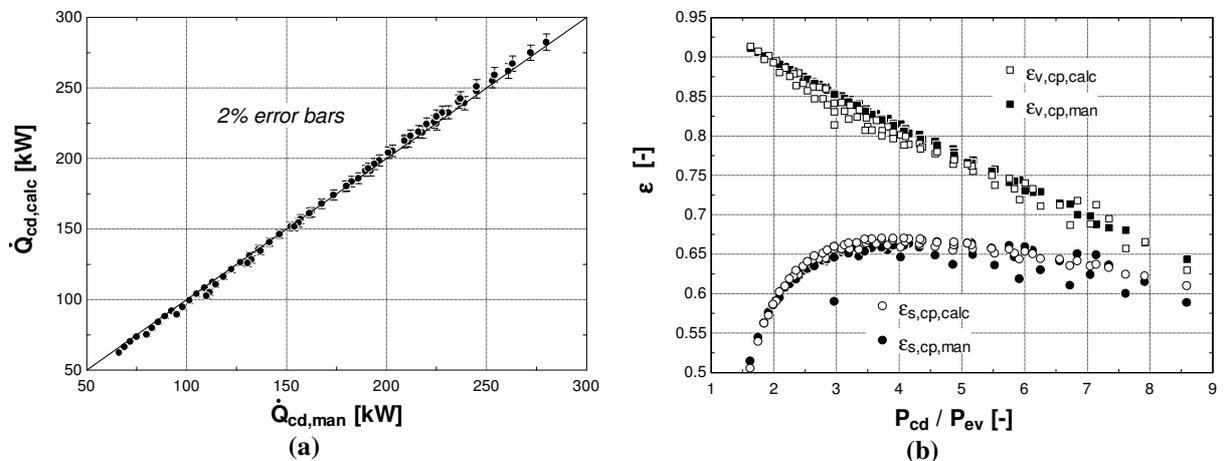


Figure 12: Prediction of (a) the exhaust temperature; (b) the volumetric and global isentropic efficiencies (RECI-AIR)

In the calibration of the reciprocating compressor model, the impact of the internal leakage, the re-expansion of the clearance volume and the exhaust pressure drop had a significant effect. In the particular case of this machine, the exhaust temperature of the compressor was not given. Instead, the heat rejection at the condenser for a given liquid subcooling was given. In order to predict with a good accuracy this power, only half of the motor electrical loss was injected into the fictitious wall (Figure 12 (a)). This might be explained by the fact the compressor is semi-hermetic. Figure 12 (b) indicates that the model is able to predict both the volumetric and isentropic efficiencies with a very good accuracy. The much lower volumetric efficiency achieved with reciprocating compressors (in comparison with scroll compressors) is due to the re-expansion of the clearance volume.

4. CONCLUSIONS

Semi-empirical models can predict the performance of components (compressors, chillers, heat pumps) with a very good accuracy. This paper showed how such models can be calibrated on the basis of manufacturer's submittal, which is most of the time the only available source of information. A calibration methodology has been proposed and illustrated for different technologies of vapor-compression machines, pointing out tips and tricks. The tailor-made formalism of the model allows describing the main features of the machine (presence of a reed valve in the compressor, economized configuration of the chiller), which largely improves the accuracy of the model.

NOMENCLATURE

A,	Area, m ²	cd	Condenser
AU,	Heat transfer coefficient, W/K	cp	Compressor
c	Specific heat, J/kg-K	econ	Economizer
d	Diameter, m	ex	Exhaust
h	Specific enthalpy, J/kg	ev	Evaporator
\dot{M}	Mass flow rate, kg/s	in	Internal
N	Number, -	leak	Leakage
P	Pressure, Pa	loss	Electro-mechanical losses
\dot{Q}	Heat transfer rate, W	m	Metal
R	Heat transfer resistance, K/W	man	Manufacturer
$r_{v,in}$	Built-in volume ratio, -	n	Nominal
T	Temperature, °C	p	Isobaric
u	Specific internal energy, J/kg	r	Refrigerant
v	Specific volume, m ³ /kg	s	Isentropic
\dot{V}	Volume flow rate, m ³ /s	sf	Secondary fluid
w	Specific work, J/kg	sh	Superheat
\dot{W}	Power, W	su	Supply
		tp	Two-phase
		w	Envelope, water
Subscript		Greek letters	
0	Constant	α	factor of proportionality
a	Air	Δ	Difference
amb	Ambient	ε	efficiency
calc	Calculated		

REFERENCES

- Bourdouxhe, J-P H., M. Grodent, J.J. Lebrun, C. Saavedra, and K.L. Silva. 1994. A toolkit for primary HVAC system energy calculation—Part 2: Reciprocating chiller models. *ASHRAE Transactions* 100(2): 774-786.
- Elson, J., N. Kaemmer, S. Wang, and M. Perevozchikov. 2008. Scroll Technology: An Overview of Past, Present and Future Developments. *Proceedings of the International Compressor Engineering Conference at Purdue*: Paper 1204.
- Hydeman, M., N., Webb, P. Sreeharan and S. Blanc. 2002. Development and Testing of a Reformulated Regression-Based Electric Chiller Model.

-
- Jin, H., and J.D. Spitler. 2002a. A Parameter Estimation Based Model of Water-to-Water Heat Pumps for Use in Energy Calculation Programs, *ASHRAE Transactions* 2002, Vol 108, Part 1.
- Jin, H., and J. Spitler. 2002b. Parameters Estimation Based Model of Water-to-Water Heat Pumps with Scroll Compressors and Water/Glycol Solutions, *Proceedings of the Sixth International Conference on System Simulation in Buildings*.
- Lemort, V., J. Lebrun, and C. Felsmann. 2009. Testing and validation of simulation tools of HVAC mechanical equipment including their control strategies Part III: validation of an Air-cooled chiller Model. In: *Proceedings of the 11th International IBPSA Conference*. Glasgow, Scotland, July 27-30, 2009.
- Morisot, O., D. Marchio. 2002. REFLIQ – Groupe Frigorifique de production d’eau glacée à condensation à air ou à eau : modélisation simplifiée dans l’optique d’un calcul des consommations d’énergie d’une installation de climatisation dans un bâtiment tertiaire. *ConsoClim, Cahier des Algorithmes*, réf.99.079R.
- Winandy, E., C. Saavedra, and J. Lebrun. 2002. Experimental analysis and simplified modelling of a hermetic scroll refrigeration compressor, *Applied Thermal Engineering* 22: 107-120.