Eighth International IBPSA Conference
Eindhoven, Netherlands
August 11-14, 2003

ROOM AIR CONDITIONERS: DETERMINATION OF EMPIRICAL CORRELATIONS FOR PREDICTING BUILDING ENERGY CONSUMPTION

Gustavo Cherem-Pereira and Nathan Mendes
Pontifical Catholic University of Paraná, Curitiba-PR, Brazil
Thermal Systems Laboratory
R. Imaculada Conceição, n. 1155, 80.215-901

ABSTRACT

Two room air conditioners were modeled in order to predict the total cooling capacity, the sensible cooling capacity and the Energy Efficiency Ratio (E.E.R.) of each appliance. The mathematical correlations obtained were written in terms of room air wet bulb temperature and outdoor-side dry bulb temperature. The use of these correlations allows predicting building energy consumption, electric power demand and equipment performance characteristics for a wide range of outdoor-side dry-bulb and room-side wet-bulb temperatures. The experimental data, from which the correlations were created, were obtained in calorimeters according to standard ISO 5151 (1994). The uncertainty on measurements is also presented.

INTRODUCTION

In the last decades, engineers have put a lot of efforts into the development of computational programs that are able to simulate the indoor climate in buildings.

The main objectives of the thermal analysis in buildings as office buildings, residential buildings and shopping malls are:

i) Provide thermal comfort in a better way;

ii) avoid the energy waste to decrease the HVAC equipment operating cost and

iii) simulating buildings interacting with HVAC equipment.

However, there is still a lack of models to predict equipment performance, specially of room air conditioners which are largely used in Brazil. According to manufacturers, in the year of 2002, 928.000 appliances were soil in the country, with a 7% saturation.

Therefore, in order to provide some information for simulating buildings with room air conditioners, we present mathematical correlations obtained from experimental data, for predicting the total cooling capacity, the sensible cooling capacity and the E.E.R. – Energy Efficiency Ratio of two different air conditioners; one of them is equipped with reciprocating compressor and the other one with rotating compressor.

Those air conditioners use HCFC-22 as the refrigerant fluid. The studied vapor compression cycle is showed in Figure 1.

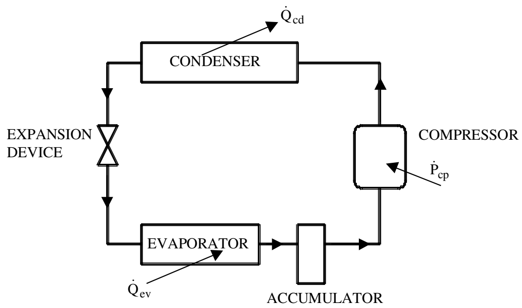

Figure 1 - Vapor-compression cycle.

The apparatus (calorimeter) utilized is described in details, according to standard ISO 5151 (1994). The uncertainty on the mathematical correlation was also investigated. The measurement uncertainty is also evaluated.

The applicability and functionality of this kind of modeling has been investigated by integrating to the building simulation program DOMUS (2001).

EXPERIMENTAL APPARATUS

In order to evaluate the air conditioning system performance during the product development, manufacturers use calorimeters as their test equipment. According to standard ISO 5151 (1994), there are three types of calorimeters: psychrometric calorimeter, calibrated calorimeter and balanced calorimeter. In this work, the necessary tests for air conditioners modeling were performed in both psychrometric and balanced calorimeters.

The psychrometric one consists of two chambers that have a wall in common when the air conditioner under test is installed. The chambers are able to control their temperature and relative humidity, simulating both outdoor-side and room-side psychrometrics conditions.

In order to measure the cooling capacity of the air conditioner, it is necessary to check only the room-side. In the region next to the equipment air suction, a psychrometer measures the dry and wet bulb temperatures, defining the psychometric air state at the evaporator inlet and then the inlet air enthalpy, $h_{int,i}$, is obtained.

The supply air enters into a nozzle chamber, where another psychrometer measures the dry and wet bulb temperatures, defining the psychrometric air state at the evaporator outlet, giving then the outlet air enthalpy, $h_{int,o}$. The pressure at the nozzle chamber entrance is maintained equal to the local atmospheric pressure so that the nozzle chamber does not interfere on the airflow of the tested air conditioner. The airflow rate can be calculated by measuring the pressure drop through a calibrated nozzle. The calculation method can be found in the standard ANSI/ASHRAE 51 (1985).

Equations 1 and 2 give the total and sensible cooling capacities of the air conditioning equipment under test, derived from the air-side energy balance in the evaporator. On the other hand, the E.E.R. is defined by the ratio of total cooling capacity to the total electric power (compressor and fan). An electrical transducer measures this total power input required by the air conditioner.

$$TC = \dot{m}_a \left(h_{int,i} - h_{int,o}\right) \quad (1)$$

$$SC = \dot{m}_a c_{pu} \left(Tdb_{int,i} - Tdb_{int,o}\right) \quad (2)$$

The psychrometric calorimeter is schematically shown in Figure 2 (in Annex).

The cooling capacity calculation in a psychrometric calorimeter involves a large quantity of parameters which causes - with the instruments and air conditioners used - cooling capacity measurement uncertainties from 1,5 up to 3,5%.

The balanced calorimeter also consists of two chambers disposed side by side. The air conditioner under test is installed in the same way. The main differences between the psychometrics and balanced calorimeters are the calculation and measuring methods.

In the balanced calorimeter, an energy balance in the room-side determines the cooling capacity. Figure 3 shows a scheme of the system and the involved energy fluxes. In steady-state conditions, the energy removed by the air conditioner is equivalent to the energy that the chamber has to supply to the system to keep the same temperature conditions. Therefore, this energy has to be measured.

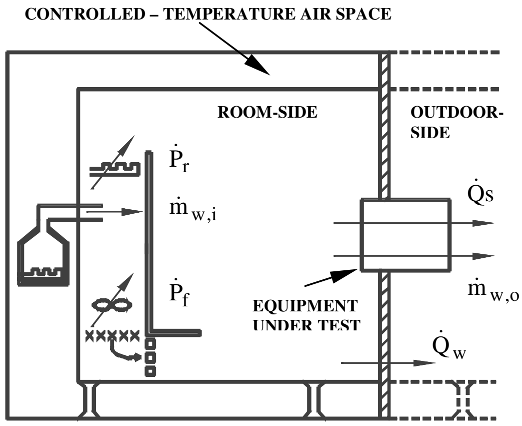

Figure 3 – Scheme of a balanced calorimeter.

The calorimeter surfaces, with the exception of the dividing wall, are considered adiabatic since the air space and the room-side have nearly the same temperatures.

Equation 3 represents an energy balance in the system. Starting from this equation, it is possible to calculate the total cooling capacity and the sensible cooling capacity.

$$0 = -\dot{P}_r - \dot{P}_f - \dot{Q}_w + \dot{Q}_s + \dot{m}_{w,i} h_{w,i} - \dot{m}_{w,o} h_{w,o} \quad (3)$$

where:

$\dot{P}_r$ is the dispersed power for the chamber's heaters.

$\dot{P}_f$ is the dispersed power for the fan that circulates the reconditioned air.

$\dot{Q}_w$ is the heat transferred through the dividing wall.

$\dot{Q}_s$ is the sensible heat that the air conditioner removes from the system.

$\dot{m}_{w,i}$ is the water flow supplied by the chamber's boiler.

$\dot{m}_{w,o}$ is the water flow removed from the system.

$h_{w,i}$ is the enthalpy of the water steam supplied by the chamber's boiler.

$h_{w,o}$ is the enthalpy of the water removed from the system.

The calculation method used for the balanced calorimeter involves less parameters and measuring instruments, which makes it more accurate for estimating cooling capacity. The measuring uncertainties observed on the tests were between 1,0 and 1,5%.

AIR CONDITIONER MODELS

The models for predicting the air conditioners performance were empirically obtained by using the input and output data gathered from the acquisition data system connected to the calorimeter. Their performance was described by mathematical correlations explicitly written to provide the total cooling capacity (TC), the sensible cooling capacity (SC) and the Energy Efficiency Ratio (EER) in terms of room-side wet-bulb temperature and outdoor-side dry-bulb temperature as expressed by Eq. 4.

$$TC, SC, EER = a_0 + a_1 T_{wb,int} + a_2 T_{wb,int}^2 + + a_3 T_{db,ext} + a_4 T_{db,ext}^2 + a_5 T_{wb,int} T_{db,ext} \quad (4)$$

This steady-state modeling is also used by the building simulation program DOE-2.0 (1993) and recommended by ASHRAE (1997).

Table 1 shows the data about the two different air conditioners modeled and the tests performed with them. The experimental tests were performed with chosen values for both dry and wet bulb temperatures within the range showed in Table 1. Table 2 presents the equation-4 coefficients for calculating the total cooling capacity, the sensible cooling capacity and the Energy Efficiency Ratio.

The errors found between the calculated and measured quantities are presented in Figs. 4-9. Figs. 4, 5 and 6 show the errors found by using the mathematical correlations obtained for the air conditioner *A* that was tested in the psychrometric calorimeter. On the other hand, Figs. 7, 8 and 9 show the errors found by using the formulation for the air conditioner *B,* which was tested in the balanced calorimeter. We notice that the model for sensible capacity presents higher errors, which are mostly due to their low sensitivity to the room air wet bulb temperature.

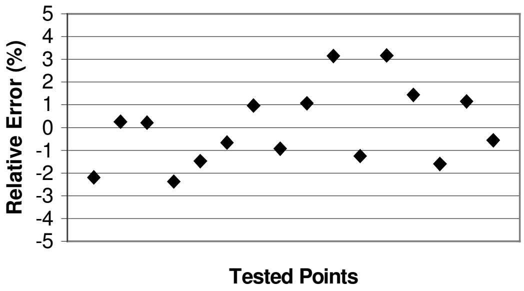

Figure 4 - Relative errors for total cooling capacity – Air Conditioner *A*.

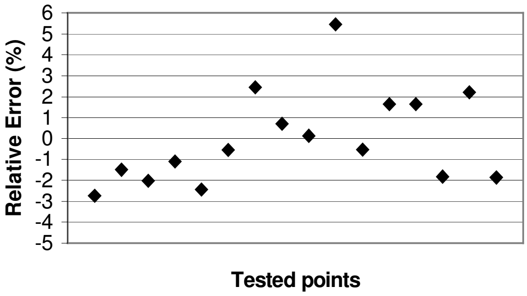

Figure 5 - Relative errors for E.E.R. – Air Conditioner *A*.

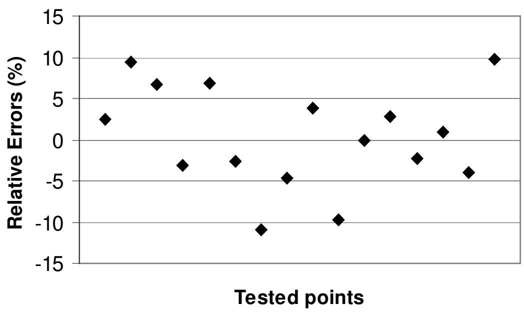

Figure 6 - Relative errors for sensible cooling capacity – Air Conditioner *A*.

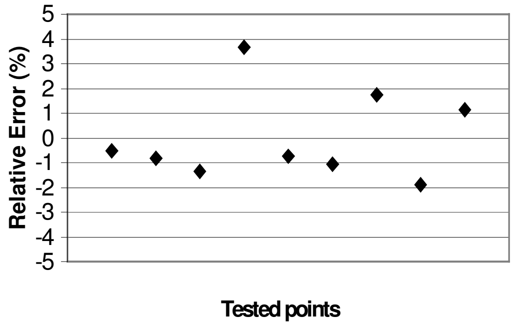

Figure 7 - Relative errors for total cooling capacity – Air Conditioner *B*.

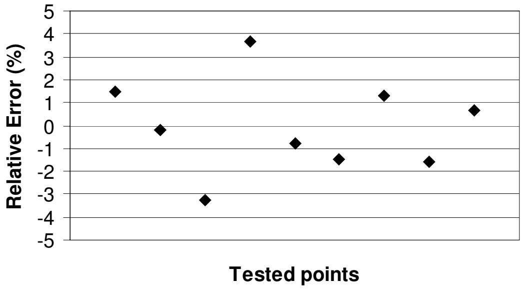

Figure 8 - Relative errors for E.E.R. – Air conditioner *B*.

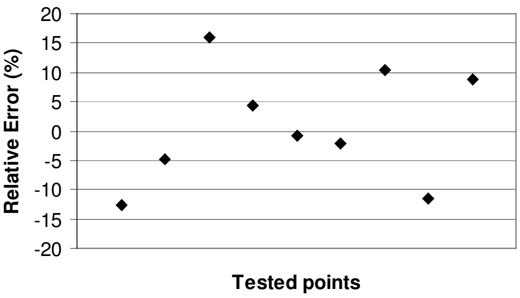

Figure 9 - Relative errors for sensible cooling capacity – Air Conditioner *B*.

The relative errors shown in Figs. 4-9 validate the use of the mathematical correlations (see the coefficients in Table 2) to predict with reasonable accuracy the performance of those two air conditioners within the temperature range tested in this work, shown in Table 1.

The nominal cooling capacity and the nominal E.E.R. are obtained by using standard conditions:

$T_{db,int}$=27°C; $T_{wb,int}$=19°C, $T_{db,ext}$=35°C and $T_{wb,ext}$=24°C (ISO 5151). However, themal performance and energy consumption strongly vary with temperature and humidity as we can see in Figs. 10-12.

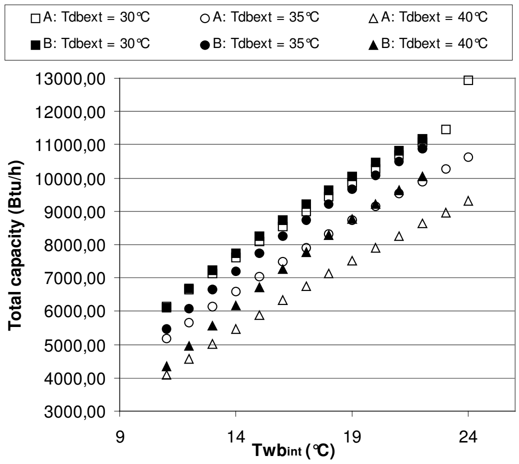

Figure 10 – Comparative of total cooling capacity under several room-side and outdoor-side temperature conditions.

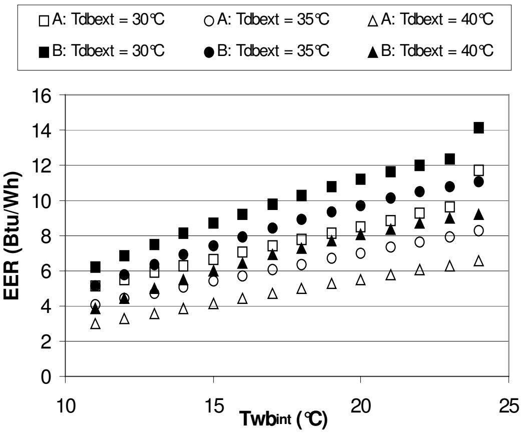

Figure 11 – EER comparison under several room-side and outdoor-side temperature conditions.

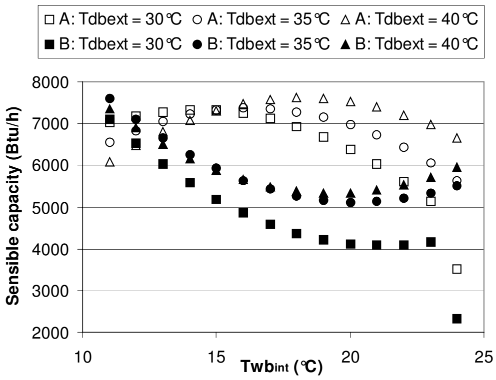

Figure 12 – Comparative of sensible capacity under several room-side and outdoor-side conditions.

Fig. 10 shows that the total cooling capacity increases with the room-side wet bulb temperature. We also notice a similar sensitivity for the total capacity to room-side dry bulb temperature for the two products. This increasing of total cooling capacity is due to a higher difference between the evaporation temperature and the room temperature when the dry bulb temperature is increased and also due to a greater latent load when the wet bulb temperature is augmented.

Nevertheless, the performance of the air conditioner *A* decreases more rapidly with the accession of the outdoor-side dry bulb temperature, which decreases the difference between the condensation and outdoor-side temperatures; As expected, the lower the condenser heat loss the lower the air conditioning efficiency.

Fig. 11 illustrates that the EER behavior for both equipment is quite similar; it increases with the room-side wet bulb temperature and decreases when the outdoor-side dry bulb temperature rises. Remarkably, the energy consumption of the air conditioner *B* is much lower, which is mostly due to the energy efficient rotating compressor.

Rotating compressors present some remarkable efficiency-related advantages:

i) they do not need devices to convert shaft kinetics energy into plunger kinetics energy, which increases their mechanical efficiency;

ii) the inlet gas enters directly into the compression chamber. In the reciprocating compressor, the HCFC-22 fluid absorbs heat from the electrical motor, increasing its specific volume before entering into the compression chamber.

iii) the gas suction and gas discharge occur in a very continuous way, consuming less electrical power.

Fig. 12 shows sensible cooling capacity curves for both air conditioners. Differently from what was presented in Figs. 10 and 11, the sensible capacity behavior was quite different for the two equipment modeled in all range of room-side wet bulb temperature.

MEASURING UNCERTANTIES

All the measuring instruments were calibrated by a certifying laboratory. The measuring uncertainties of the main sensors are presented on Table 3. These uncertainties combines both sensor and data acquisition system errors.

The cooling capacities calculation incorporates many parameters and equations, involving moist air properties. Therefore, the measuring uncertainties for cooling capacity, mentioned on previous sessions, were calculated by using the error propagation theory (Taylor, 1988).

CONCLUSIONS

We presented an experimental apparatus – psychrometric and balanced calorimeters according to standard ISO 5151 (1994) – used to test two room air conditioners. These air conditioners were modeled in order to predict, in an easy way, the total cooling capacity, the sensible cooling capacity and the E.E.R. – Energy Efficiency Ratio of each appliance for different thermal conditions. The mathematical correlations obtained were written in terms of room air wet bulb temperature and outdoor-side dry bulb temperature. The use of these correlations allows predicting building energy consumption, electric power demand and equipment performance characteristics for a wide range of outdoor-side dry-bulb and room-side wet-bulb temperatures.

The derived mathematical correlations are considerably accurate so that they can provide good results and are easily implementable in building simulation programs.

A suggestion for further work is the determination of a new correlation to increase the accuracy on predicting the sensible capacity. Another suggestion is the investigation of thermal characteristics variation of room air conditioners with time. This could also motivate air conditioners manufacturers to provide those correlations on their products catalogue.

ACKNOWLEDGEMENTS

The authors thank both CAPES - Coordenação de Aperfeiçoamento de Pessoal de Nível Superior - of the Secretary for Education and Culture of Brazil and CNPq (Conselho Nacional de Desenvolvimento Científico e Tecnológico) of the Secretary for Science and Technology for support of this work. Special thanks are given to Electrolux of Brazil for disposing the calorimeters.

REFERENCES

ANSI / ASHRAE 51-1985 ANSI / AMCA standard 210-85, Laboratory methods of testing fans for rating, 1985.

ASHRAE, American Society of Heating, Refrigerating and Air Conditioning Engineers, Handbook of Fundamentals. Atlanta:ASHRAE, 1997.

ISO 5151: 1994 (E), Non-ducted air conditioners and heat pumps – Testing and rating for performance, 1994.

Mendes N., Oliveira R.C.L.F. e Santos G.H., DOMUS 1.0: A Brazilian PC Program for Building Simulation, Seventh International Conference on Building Performance Simulation (IBPSA´01), V.1, n.1, p.83-89, Rio de Janeiro, Brazil, 2001.

Taylor, J. L., Fundamentals of Measurement Error, Neff Instrument Corporation, 1988.

Winkelmann F. C., Birdsall B. E., Buhl W. F., Ellington K. L., Erdem A. E. DOE-2 Supplement, V. I, Version 2.1E, Energy and Environment Division. Lawrence Berkeley Laboratory. University of California, Berkeley, CA, 1993a.

Winkelmann F. C., Birdsall B. E., BuhlW. F., Ellington, K. L., Erdem A. E. DOE-2 Supplement, V. II, version 2.1E, Energy and Environment Division. Lawrence Berkeley Laboratory. University of California, Berkeley, CA, 1993b.

NOMENCLATURE

a_0 to a_5 Correlation Coefficients

c_{pu} Specific heat at constant pressure of moist air, J/kg K

E.E.R. Energy Efficiency Ratio, W/W [1 W = 3,4121 Btu/h]

$h_{int,i}$ Input air enthalpy, J/kg

$h_{int,o}$ Output air enthalpy, J/kg

$h_{w,i}$ Input enthalpy of the water steam, J/kg

$h_{w,o}$ Output enthalpy of the condensed water , J/kg

$\dot{m}_a$ Air flow, kg/s

$\dot{m}_{w,i}$ Input water flow, kg/s

$\dot{m}_{w,o}$ Output water flow, kg/s

$\dot{P}_f$ Dispersed power for the motor-fan, W

$\dot{P}_r$ Dispersed power for the chamber's heaters, W

$\dot{Q}s$ Sensible heat flux removed by the air conditioner, W

$\dot{Q}_w$ heat flux through dividing wall, W

R^2 Correlation coefficient

SC Sensible cooling capacity, W [1 W = 3,4121 Btu/h]

TC Total cooling capacity, W [1 W = 3,4121 Btu/h]

$Tdb_{int,i}$ Input dry bulb temperature of room-side, °C

$Tdb_{int,o}$ Output dry bulb temperature of room-side, °C

Tdb_{ext} Dry bulb temperature of outdoor-side, °C

Twb_{int} Wet bulb temperature of room-side, °C

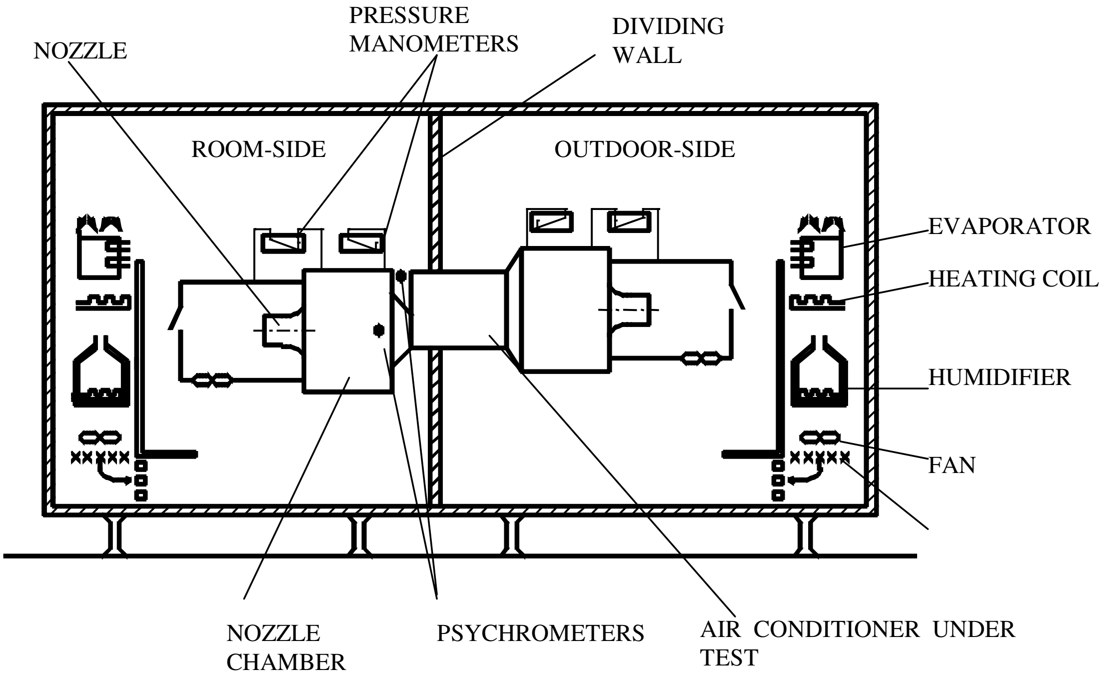

Figure 2 – Psychrometric calorimeter [ISO 5151].

Table 1 – Information about the air conditioners tested.

Air conditioner	Cooling Capacity (Btu/h)	Compressor	Compressor nominal EER (Btu/Wh)	Number of Tests	Calorimeter	Tested Range of $T_{wb,int}$ (°C)	Tested Range of $T_{db,ext}$ (°C)
A	9500	Reciprocating	8,4	16	Psychrometric	10,9 to 24,2	22,4 to 40,9
B	10000	Rotating	10,9	9	Balanced	12,4 to 23,2	26,6 to 41,1

Table 2 – Correlation Coefficients.

Air conditioner		a_0	a_1	a_2	a_3	a_4	a_5	R^2
A	Total capacity	2353,7974	740,84314	-5,53694	-21,69204	-2,03529	-3,66084	0,992
	E.E.R.	4,47035	0,71301	-0,00136	-0,14256	0,00056	-0,00981	0,990
	Sensible capacity	11317,7008	131,99676	-29,069	-314,71777	-0,51866	23,27489	0,641
B	Total capacity	-6427,44961	645,52971	-10,84917	462,21141	-10,03916	5,80981	0,989
	E.E.R.	-3,62145	1,22351	-0,01305	0,15472	-0,00417	-0,00886	0,989
	Sensible capacity	1488,70996	-1554,89597	29,00601	982,45497	-15,37698	10,82215	0,701

Table 3 – Measuring uncertainties.

Parameter	Sensor	Uncertainty
Dry and wet bulb temperatures	RTD [0 – 60 °C]	± 0,05 - 0,08°C
Air flow rate	Pressure manometers [0 – 50 mm H_20]	± 0,05 mm H_20
	Nozzle [88,0 mm]	± 0,0001 m
Power input	Electrical transducer – Voltage [220V]	± 0,7 V
	Electrical transducer – Current [5 – 10 A]	± 0,02 A
Water flow rate	Flowmeter [0 – 100 L/h]	± 1%