EMPIRICAL MODELING OF DIRECT-EXPANSION AIR CONDITIONERS FOR BUILDING ENERGY EFFICIENCY ANALYSES

Cherem Pereira G
Pontifical Catholic University of Paraná, Imaculada Conceição St., 1155, Curitiba-PR, 80.215-901, Brazil
guscherem@onda.com.br

Mendes N
Pontifical Catholic University of Paraná, Imaculada Conceição St., 1155, Curitiba-PR, 80.215-901, Brazil
nmendes@rla01.pucpr.br

Three 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 in any environment 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 experimental data, from which the correlations were created, were obtained in calorimeters according to standard ISO 5151 (1994). A mathematical model to integrate to the air conditioning models with a building simulation model is also presented and a simulation sample is carried out.

Keywords. *Air conditioner, Regression Models, HVAC Simulation, Building Modeling.*

1. Introduction

The air conditioners models found in literature can be analytical, semi-empirical or empirical. The analytical models depend on many parameters; they integrate the basic components of the vapor compression cycle models. In a similar way, the semi-empirical models depend on test data and fewer parameters, mixing analytical and empirical calculations.

In spite of these many-input-models are useful for studying refrigeration systems, they are too time consuming when integrated to yearly building simulation programs. In this way, this work is focused on presenting empirical models for room air conditioners and integrating them with a building simulation code.

Empirical models of direct-expansion air conditioners use only test data and can be much simpler, providing accurate results though. This kind of modeling let users spend much less efforts for predicting air conditioners performance. The models are based on experimental data obtained from tests carried out in a calorimeter according to standard ANSI/AHAM RAC-1 (1992). This calorimeter and associated measurement uncertainties are described by Pereira and Mendes (2003).

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

i) Providing thermal comfort in a better way;
ii) avoiding 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, especially of room air conditioners, which are largely used in Brazil. According to manufacturers, in the year of 2002, 928.000 appliances were sold in the country, with a 7% saturation degree in residences.

Simulation programs as DOE-2.0 (1993), DOMUS (2003) and Clim2000 (1993) use empirical modeling of air conditioning system, based on performance data map normally provided by equipment manufacturers. Nevertheless, room air conditioners (RAC) manufactures do not normally provide listings of performance.

Thereupon, in order to provide some information for simulating buildings equipped 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 three different air conditioners; one of them is supplied with reciprocating compressor and the other two with rotating compressor.

Those air conditioners use HCFC-22 as the refrigerant fluid. The studied vapor compression cycle is showed in Figure 1. For the reciprocating compressor, the compressor's case plays the accumulator role.

The applicability and functionality of this kind of modeling has been investigated by integrating it to the building simulation program DOMUS (Mendes *et al.*, 2003). The simulation model is based on a lumped approach formulation for calculating both room air temperature and humidity ratio. In the energy balance, loads of sensible and latent conduction heat transfer, convection heat transfer, short-wave solar radiation, inter-surface long-wave radiation, infiltration, ventilation and HVAC system related loads are considered. The sensible and latent conduction loads that crosses the control surface of each zone are described by Mendes *et al.* (2003). Considering the Domus program model, simulations are carried out and the air conditioners performance is discussed.

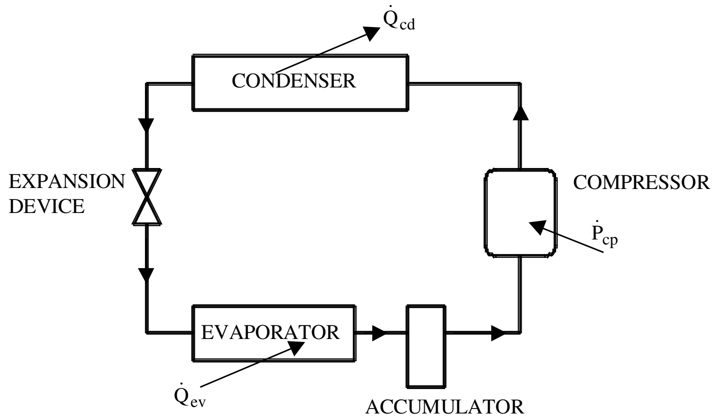


Figure 1. Vapor-compression cycle.

2. Air conditioner models

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 calorimeters consist 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 psychrometric conditions. The main differences between the psychometric and balanced calorimeters are the calculation and measuring methods. In order to measure the cooling capacity of the air conditioner, the psychrometrics calorimeter defines the psychometric air state at the evaporator inlet and outlet and measures the supply airflow. In the balanced calorimeter, an energy balance in the room-side determines the cooling capacity. Both calorimeters successfully accomplish the required tests

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 (C_T), the sensible cooling capacity (C_S) and the Energy Efficiency Ratio (EER) factors (Z) in terms of room-side wet-bulb temperature and outdoor-side dry-bulb temperature as expressed by Eq. 1.

$$Z_{CT}, Z_{CS}, Z_{EER} = a_0 + a_1 T_{wb,\mathrm{int}} + a_2 T_{wb,\mathrm{int}}{}^2 + a_3 T_{db,ext} + a_4 T_{db,ext}{}^2 + a_5 T_{wb,\mathrm{int}} T_{db,ext} \tag{1}$$

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 three different air conditioners modeled and the tests performed with them. The nominal cooling capacity is a measured value obtained under standard conditions. Table 2 presents the equation-4 coefficients for calculating the total cooling capacity, the sensible cooling capacity and the Energy Efficiency Ratio.

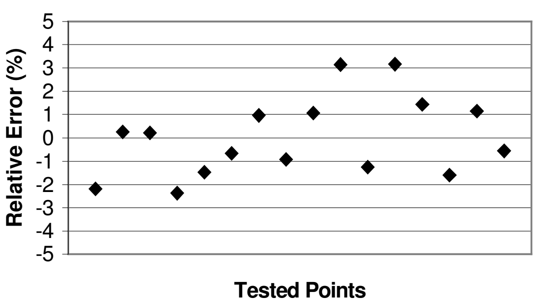
Table 1. Information about the air conditioners tested.

Air conditioner	Nominal Cooling Capacity (Btu/h)	Compressor	Compressor nominal EER (Btu/hW)	Evaporator Air flow under standard conditions (kg/s)	Number of Tests	Calorimeter	Tested Range of $T_{wb,int}$ (°C)	Tested Range of $T_{db,ext}$ (°C)
A	9060	Reciprocating	8.4	0,1438	16	Psychrometric	10.9 to 24.2	22.4 to 40.9
B	9900	Rotating	10.9	0,1346	9	Balanced	12.4 to 23.2	26.6 to 41.1
C	11960	Rotating	10.7	0,1495	11	Balanced	11.5 to 23.9	26.1 to 42.8

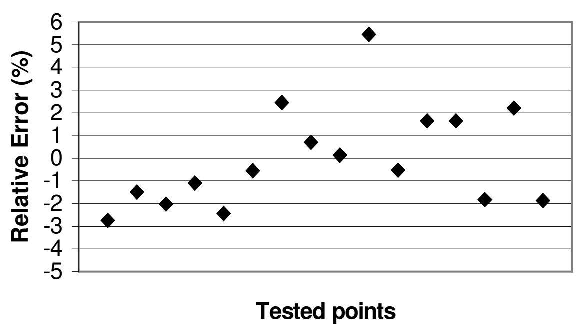
Table 2. Correlation Coefficients.

Air conditioner		a_0	a_1	a_2	a_3	a_4	a_5	R^2
A	Total capacity	0,25980104	0,08177077	-0,00061114	-0,00239426	-0,00022465	-0,00040407	0,992
	E.E.R.	0,63862143	0,10185857	-0,00019429	-0,02036571	0,00008	-0,00140143	0,990
	Sensible capacity	1,81083213	0,02111948	-0,00465104	-0,05035484	-0,00008299	0,00372398	0,64
B	Total capacity	-0,64923733	0,06520502	-0,00109588	0,04668802	-0,00101406	0,00058685	0,989
	E.E.R.	-0,38608209	0,13043817	-0,00139126	0,01649467	-0,00044456	-0,00094456	0,989
	Sensible capacity	0,25358180	-0,26485570	0,00494079	0,16734804	-0,00261926	0,00184341	0,701
C	Total capacity	0,89676	0,09595	-0,00277	-0,05400	0,00016	0,00144	0,973
	E.E.R.	1,67968	0,13061	-0,00300	-0,10015	0,00081	0,00066	0,961
	Sensible capacity	1,35820	-0,10118	0,00020	0,03615	-0,00115	0,00241	0,651

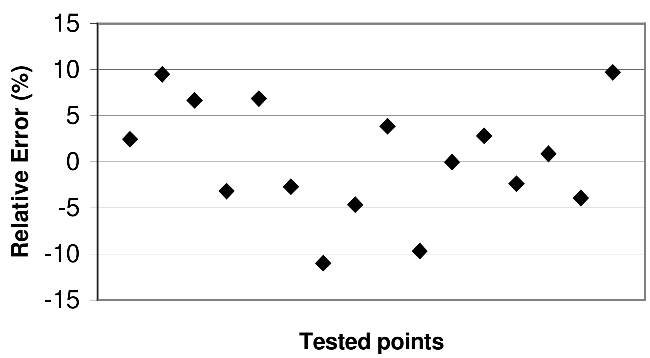
The errors found between the calculated and measured quantities are represented in Figs. 2a, 2b and 2c. Figs. 2a, 2b and 2c show the errors found by using the mathematical correlations obtained for the air conditioner *A* that was tested in the psychrometric calorimeter. The errors found by using the formulation for the air conditioners *B* and *C* are similar, which were tested in the balanced calorimeter. We notice that the model for sensible cooling capacity presents higher errors, which are mostly due to their lower sensitivity to the room air wet bulb temperature. At this point, recommendations are addressed on the development of new regression models for predicting more accurately the RAC sensible cooling capacities.



(a)



(b)



(c)

Figure 2. Relative errors for Air Conditioner *A*.. (a) Total cooling capacity. (b) E.E.R. (c) Sensible cooling capacity

The relative errors shown in Fig. 2 validate the use of the mathematical correlations (see the coefficients in Table 2) to predict with reasonable accuracy the performance of 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: $T_{db,int}$=26,7°C; $T_{wb,int}$=19,4°C, $T_{db,ext}$=35°C and $T_{wb,ext}$=23,9°C (ANSI / AHAM RAC-1-1992). However, thermal performance and energy consumption strongly vary with temperature and humidity as we can see in Figs. 3-5.

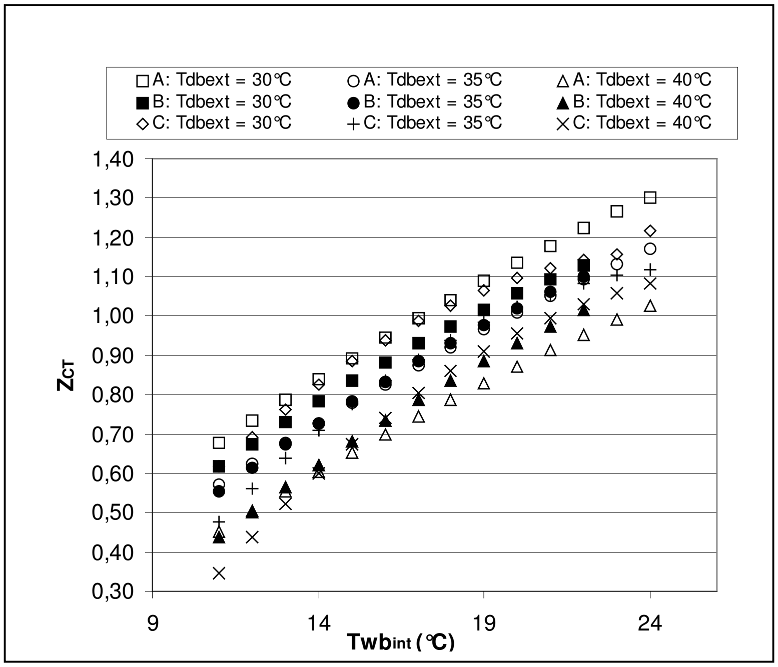


Figure 3. Comparative of total cooling capacity factor under several room-side and outdoor-side temperature conditions.

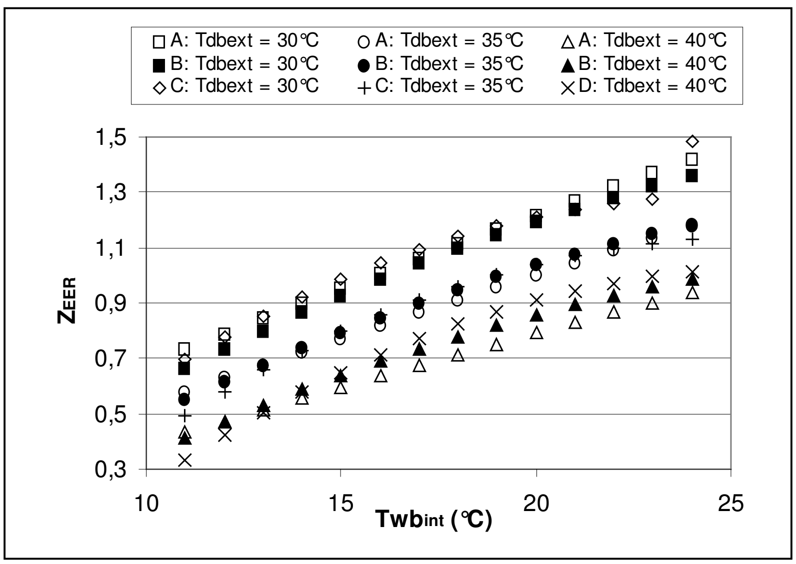


Figure 4. EER factor comparison under several room-side and outdoor-side temperature conditions.

Figure 3 shows that the total cooling capacity factor 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 augmentation 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 conditioner efficiency.

Figure 4 illustrates that the EER factor 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.

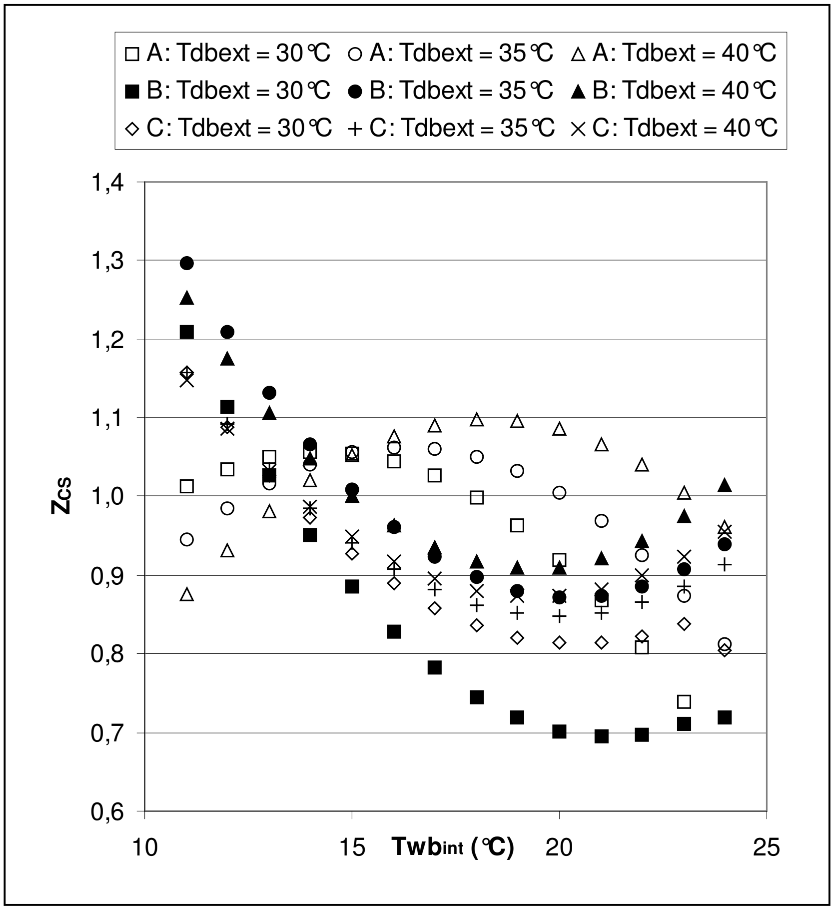


Figure 5. Comparative of sensible capacity under several room-side and outdoor-side conditions.

Figure 5 shows sensible cooling capacity curves for both air conditioners. Differently from what was presented in Figs. 3 and 4, the sensible capacity behavior was quite different for the equipments modeled in all range of room-side wet bulb temperature.

3. Integration to a building model

The present air conditioner empirical model can be integrated to the building simulation program DOMUS (Mendes *et al.*, 2003), which uses a dynamic model for the analysis of a whole-building hygrothermal behavior. In this case, a lumped formulation for calculating both room air temperature and humidity ratio is considered for each building zone. Equation 2 describes the energy balance, for a zone submitted to loads of conduction, convection, short-wave solar radiation, inter-surface long-wave radiation, infiltration, ventilation and HVAC system related loads.

$$\dot{E}_t + \dot{E}_g = \rho_{air} c_{air} V_{air} \frac{dT_{int}}{dt} \tag{2}$$

where:

$\dot{E}_t$ energy flow that crosses the room (W)

$\dot{E}_g$ internal energy generation rate (W)

ρ_{air} air density (kg/m^3)

c_{air} specific heat of air (J/kg-K)

V_{air} room volume (m^3)
T_{int} room air temperature (^oC)

The term $\dot{E}_t$, on the energy conservation equation, includes loads associated to the building envelope (sensible and latent conduction heat transfer), furniture (sensible and latent), fenestration (conduction and solar radiation), openings (ventilation and infiltration) and HVAC systems. The total conduction heat flux that crosses the control surface of each zone is described by Mendes *et al.* (2003).

For the combined heat and moisture transfer problem through the building zone porous walls, Mendes *et al.* (2002) discretized the conservation equations by using the control-volume formulation method with a central difference scheme and linearized vapor concentration difference at the boundaries in terms of temperature and moisture content. The resulting algebraic equations were solved using the MultiTriDiagonal-Matrix Algorithm (MTDMA) as described by Mendes and Philippi (2003).

In terms of water vapor balance, different contributions were considered: ventilation, infiltration, internal generation, porous walls, furniture, HVAC system and people breath. In this way, the lumped formulation becomes:

$$(\dot{m}_{inf} + \dot{m}_{vent})(W_{ext} - W_{int}) + J_b + J_{ger} + J_{porous\ surface} + J_{HVAC} = \rho_{air} V_{air} \frac{dW_{int}}{dt} \tag{3}$$

where:
$\dot{m}_{inf}$ air mass flow by infiltration (kg dry air/s)
$\dot{m}_{vent}$ air mass flow by ventilation (kg dry air/s)
W_{ext} external humidity ratio (kg water/kg dry air)
W_{int} internal humidity ratio (kg water/kg dry air)
J_b water vapor flow from the breath of occupants (kg/s)
J_{ger} internal water-vapor generation rate (kg/s)
$J_{porous\ surfaces}$ water vapor flow from porous surfaces (walls, partitions and furniture) (kg/s)
J_{HVAC} vapor flow from HVAC systems (kg/s)
ρ_{air} air density (kg dry air/s)
V_{air} room volume (m3)

The water-vapor mass flow from the people breath is calculated as shown in ASHRAE (1993), which takes into account the room air temperature, humidity ratio and physical activity as well.

The HVAC model integration to the building zone is done by inserting the total cooling capacity (C_T) on the second left-hand term of the energy conservation equation applied to the room control volume.

C_T is calculated by multiplying Z_{CT} factor by the equipment nominal total cooling capacity ($C_T = Z_{CT} C_{T,Nom}$). However, as the evaporator absorbs heat from the building zone, the C_T signal is negative on Eq. (4).

In the water vapor mass balance equation, the term J_{HVAC} is calculated as:

$$J_{HVAC} = -\frac{C_T - C_S}{h_{LV}(\overline{T})} \tag{4}$$

where $\overline{T}$ is the mean value between the room and supply air temperatures.

The room air temperature (T_{int}) is calculated by the Energy Conservation Equation, while the supply air temperature (T_{ins}) by the following expression:

$$T_{ins} = T_{int} - \frac{C_S}{\dot{m} c_p} \tag{5}$$

The energy consumption (E_c, kWh) is calculated as:

$$E_c = \frac{1}{1000} \int_0^{\tau} \frac{C_T}{EER} dt \tag{6}$$

where τ is the simulation period, in which the air conditioning can be either turned on or off.

3.1. Results

Figures 6 and 7 shows the Curitiba weather data file, in terms of temperature and relative humidity and total solar radiation, from Jan. 1st to Jan. 31st. These hourly data were obtained from the Umidus program (Mendes *et al.*, 1999).

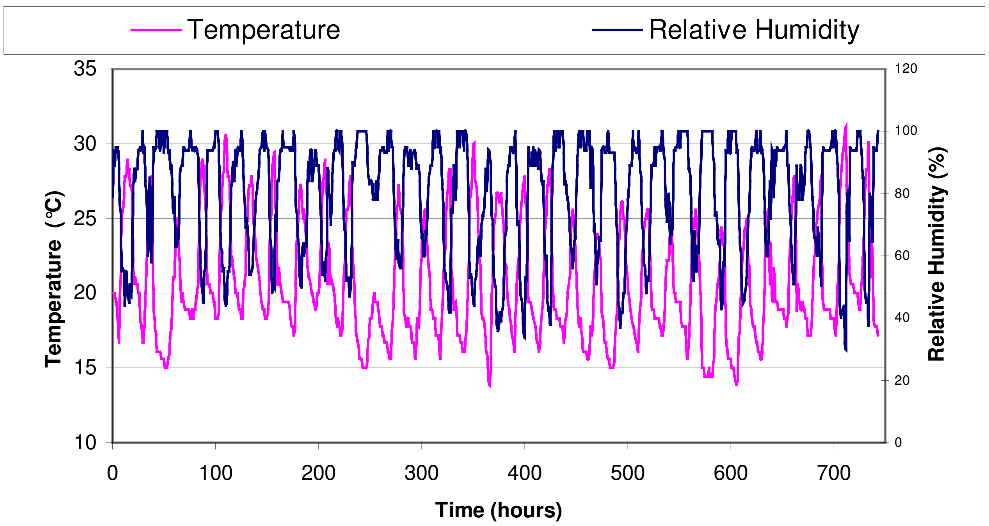


Figure 6: Temperature and relative humidity for the city of Curitiba-PR from Jan. 1st to Jan. 31st.

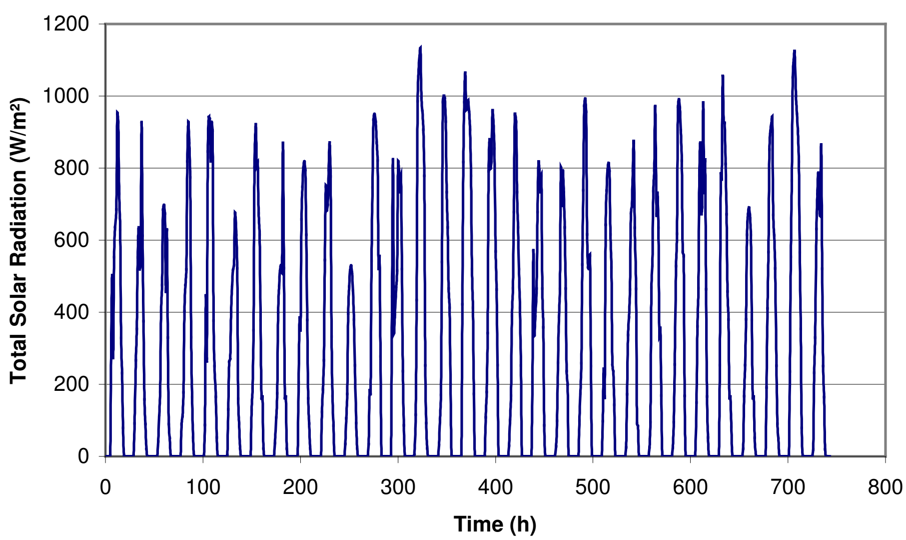


Figure 7: Total solar radiation for the city of Curitiba-PR from Jan. 1st to Jan. 31st.

The total solar radiation presented in Fig. 7 was calculated by the DOE-2 (Winkelmann et. al., 1993) model which is based on the sky cloudiness.

The single-zone building simulated is illustrated in Fig. 8. No thermal gains such as people, equipment and lighting were considered in the simulation, neither moisture sources. The room area is 25 m^2 and its height is equal to 2.5 m^2.

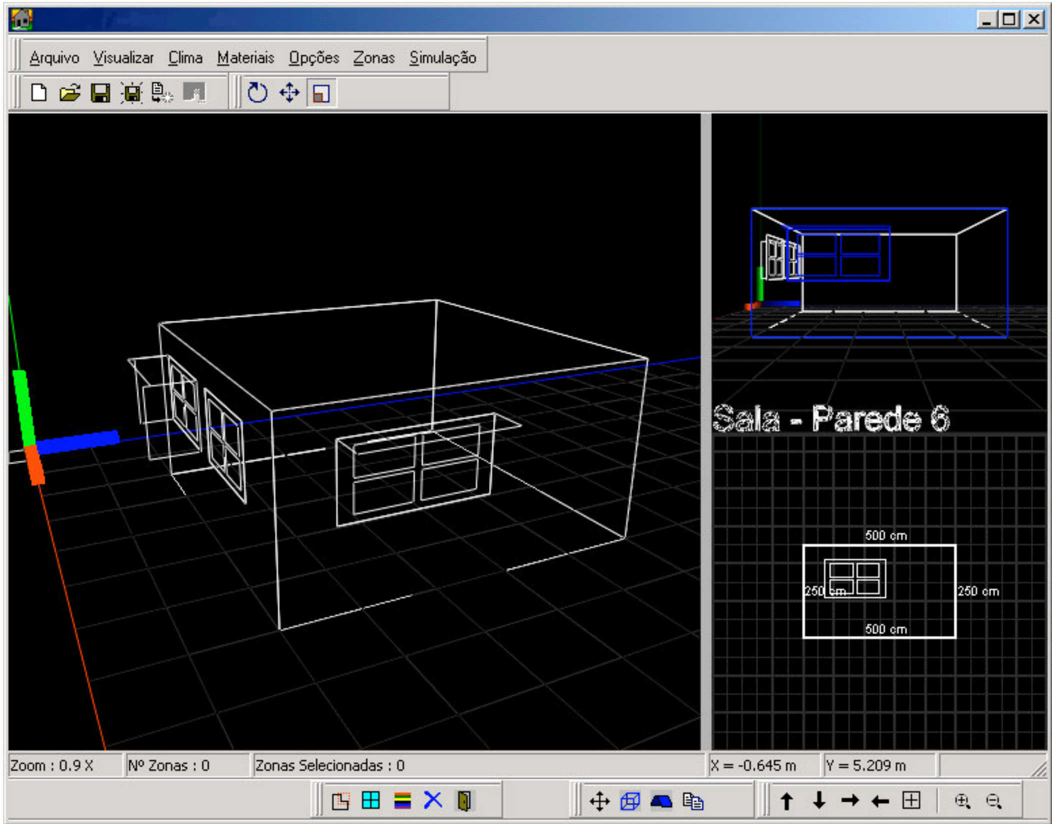


Figure 8: Program Domus main menu showing the simulated single-zone building .

The input parameters are described in Fig. 9. A 1-min simulation time step was used in order to avoid non-steady information loss, which could affect the temperature and relative humidity histories within the room. Another point to consider using very small time steps is the accuracy on the equipment power time integration for calculating the energy consumption.

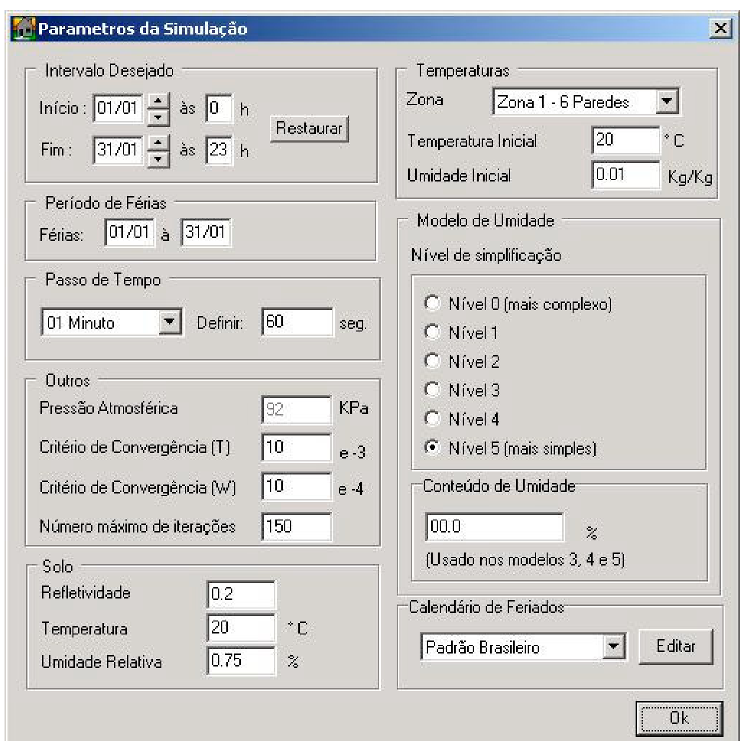


Figure 9: Program Domus input parameters for simulating the single-zone building .

Fig. 10 presents a comparison in terms of room air temperature, showing simulation results with no air conditioning and comparing then with temperature data obtained by simulating the single-zone building with two different air conditioners; one equipped with reciprocating compressor (RAC A) and the other one with rotating compressor (RAC B). No difference could be noticed between the room air temperatures obtained by using the two air conditioners, which is explained by the hypothetical PID control system used in the simulation. The PID (proportional, Integral, derivative) constants used are respectively equal to 1, 0.5 and 500.

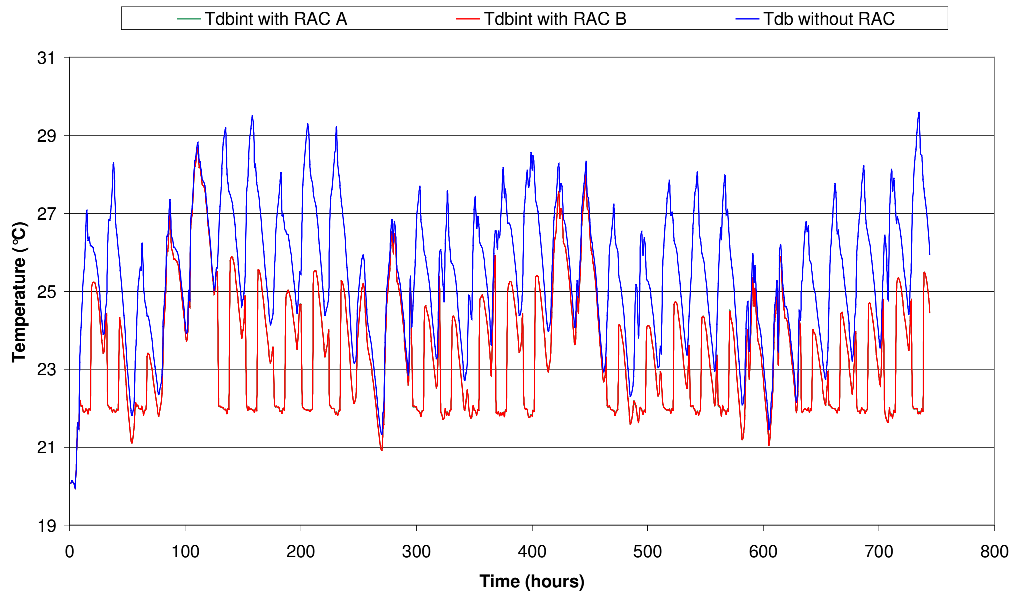


Figure 10: Room air temperature from Jan. 1st to Jan. 31st.

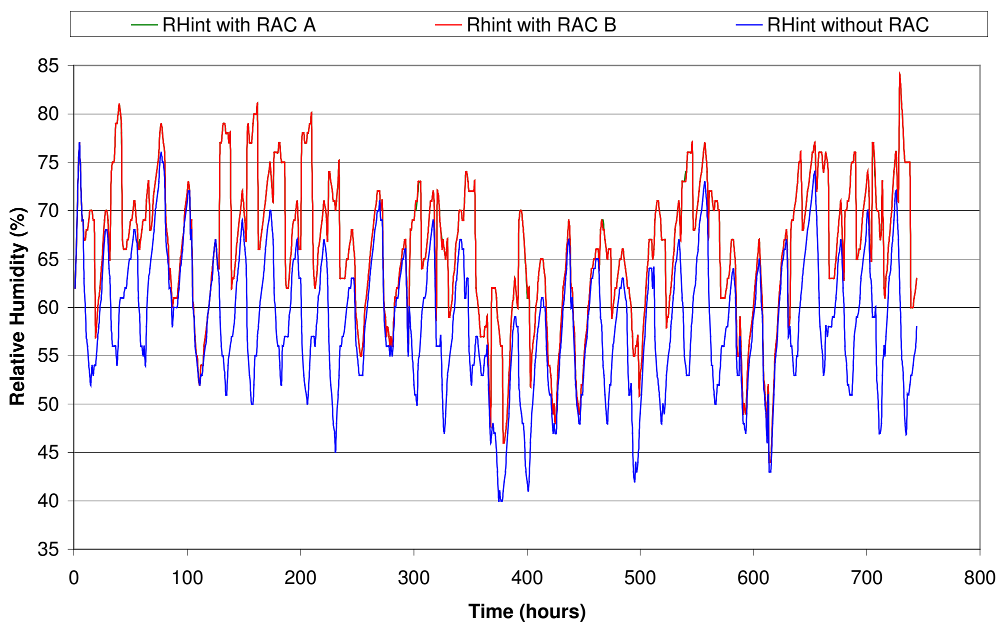


Figure 11: Room air relative humidity from Jan. 1st to Jan. 31st.

Figure 11 illustrates the room air relative humidity from Jan. 1st to Jan. 31st. Similarly to what was noticed for temperature, the two air conditioners provided the same room air relative humidity. However, a 17% energy consumption reduction was observed when the 10.000-Btu/h rotating-compressor air conditioner was used rather than the reciprocating one. This energy-efficient behavior for RAC B was previously expected as commented on Section 2 about the rotating compressor efficiency.

4. Conclusions

We presented empirical models for three room air conditioners tested in two experimental apparatus (psychrometric and balanced calorimeters according to standard ISO 5151 (1994)). These regression models were developed in order to predict, in an easy way, the total cooling capacity, the sensible cooling capacity and the Energy Efficiency Ratio (E.E.R.) 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 to be inserted in building simulation programs as shown in section 3 for a single-zone building. The integration results showed that the rotating-compressor can considerably reduce the energy consumption. For the case simulated, this air conditioning presented the same performance in terms of temperature and relative humidity than the reciprocating one, but with a 17% energy consumption reduction.

A suggestion for further work is the determination of a new regression model to increase the accuracy on predicting the sensible cooling capacity of room air conditioners.

5. Acknowledgement

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