# MODELING AND TESTING OF ROOM AIR CONDITIONERS IN HEATING AND COOLING MODES

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**Abstract:** This paper presents some simulation models and testing methods applicable to reversible room air conditionners. The first part of the paper describes the reference (“mother”) model. This model is an assembly of real and/or fictitious physical processes. It needs some tuning, in order to fit with the behavior of a given component. The tuning is made with help of manufacturer information and/or of test results. The second part of the paper presents the test method, some test results and reference model validation. The testing method described here is based on a double energy balance: one on refrigerant side and one on air side. The third part of the paper presents a simplified (“daughter”) model, reduced to a set of polynomial expressions, easy to use and to integrate into a global building simulation software.

**Key Words:** ***Room air conditioners, simulation, experiment***

## 1 INTRODUCTION

The results presented in this paper were obtained in the frame of a European research program (Philippe Rivière, 2007).

As most other HVAC components, room air conditioners can be modelled at two levels:

- A *"reference” (or "mother")* model can be used in order to integrate laboratory test results and to predict performances in various conditions. This first model is, as much as possible, based on physical phenomena; it contents almost all what is known about the system considered. It needs nevertheless some parametric adjustment in order to fit with a given component.
- After having been tuned on experimental data available, the reference model can be used to generate a *"simplified" (or “daughter”)* model. This second model is designed in such way to reproduce the behaviour of the reference model, with just enough accuracy and with a maximum of computation robustness.

The testing method presented hereafter is based on a double energy balance:

- The *air side* energy balance ("calorimeter" method) consists in installing each (indoor and outdoor) unit inside a calorimeter. Each calorimeter contains all (water and heat) sources and sinks, required to absorb the emission of each unit and to maintain the required indoor and outdoor climate conditions.
- The *refrigerant side* energy balance ("enthalpy method") is based on the determination of the refrigerant enthalpy flow rate supplied to each element of the RAC (compressor, condenser, expansion valve and evaporator). This is made possible thanks to the use of a set of temperature and pressure sensors and of a Coriolis flow meter, introduced in the refrigerant circuit. But this method can only be

applied in one of the two (cooling/heating) regimes for a reversible RAC: the Coriolis flow meter can only work in one direction.

Two examples of ("medium" and "high" classes) "split" air conditioners are analysed. Both units have a nominal cooling capacity of 2700 W. In both cases also, the compressor is driven through an inverter.

## 2 DESCRIPTION OF THE REFERENCE MODEL

The reference model of a room air conditioner is built by assembling several component models described hereafter.

### 2.1 Heating Coil and Dry Cooling Coil

A same heating and dry cooling coil model is used to simulate both the condenser and the evaporator in dry regime. This model considers the coil as a one-zone fictitious semi isothermal heat exchanger. Equations are hereafter presented for a coil used as evaporator on refrigerant side and as dry cooler on air side.

The output variables are the coil thermal power and the exhaust temperatures of both fluids (air and refrigerant).

The parameters are: the nominal flow rates of both fluids and the nominal values of three thermal resistances (air side, metal and refrigerant side).

The input variables are the supply conditions on both sides of the coil.

The overall heat transfer coefficient of this heat exchange is calculated by considering three thermal resistances in series:

$$AU_{ev,dry} = \frac{1}{R_{a,ev} + R_{r,ev,dry} + R_{m,ev}} \tag{1}$$

$$R_{r,ev,dry} = R_{r,ev,n} \cdot \left[\frac{\dot{M}_{r,ev,n}}{\dot{M}_{r,ev,dry}}\right]^{0.8} \tag{2}$$

$$R_{a,ev} = R_{a,ev,n} \cdot \left[\frac{\dot{M}_{a,ev,n}}{\dot{M}_{a,ev}}\right]^{0.6} \tag{3}$$

Nominal values of the thermal resistances and of the flow rates are the model parameters, which must be identified on the basis of experimental results or catalogue data.

The cooling power can be defined, on air and refrigerant sides, by Equations 4 and 5 respectively:

$$\dot{Q}_{ev,dry} = \dot{M}_{a,ev} \cdot cp_{a,su,ev} \cdot (t_{a,su,ev} - t_{a,ex,ev,dry}) \tag{4}$$

$$\dot{Q}_{ev,dry} = \dot{M}_{r,ev,dry} \cdot \Delta_{h,ev,dry} \tag{5}$$

Heat transfer through the (fictitious) semi-isothermal-flow heat exchanger is give as:

$$\dot{Q}_{ev,dry} = \varepsilon_{ev,dry} \cdot \dot{C}_{a,ev,dry} \cdot (t_{a,su,ev} - t_{ev,mean,dry}) \tag{6}$$

$$\varepsilon_{ev,dry} = 1 - \mathbf{exp}(-NTU_{ev,dry}) \qquad NTU_{ev,dry} = \frac{AU_{ev,dry}}{\dot{C}_{a,ev,dry}} \tag{7}$$

The temperature on refrigerant side is defined in weighted average:

$$t_{ev,mean,dry} = \frac{t_{ev,dry} \cdot (h_{r,sat,ex,ev,dry} - h_{r,sat,su,ev,dry}) + t_{r,ex,ev,dry} \cdot (h_{r,ex,ev,dry} - h_{r,sat,ex,ev,dry})}{h_{r,sat,ex,ev,dry} - h_{r,sat,su,ev,dry} + h_{r,ex,ev,dry} - h_{r,sat,ex,ev,dry}} \tag{8}$$

## 2.2. Cooling Coil in Dry and Wet Regimes

This model is adapted from Merckel theory (combination of latent and sensible heat transfer): air enthalpy is here replaced by wet bulb temperature as total heat transfer potential).Fully dry and fully wet regimes are described simultaneously and the regime selected is the one leading to the highest cooling capacity (Braun and co. 1989).

Selected outputs are:
- Coil “emissions” (total, sensible and latent cooling power);
- Air state at coil exhaust (temperature, moisture content and relative humidity);
- Water condensate flow rate;
- Refrigerant temperature at coil exhaust.

The parameters and the input variables are the same as in dry regime.

The equations already developed are transposed to the wet regime by substituting to the air a fictitious ideal gas, whose temperature is the actual air wet bulb temperature.

The air side heat balance and heat transfer equations become:

$$\dot{Q}_{ev,wet} = \dot{M}_{a,ev} \cdot (h_{a,su,ev} - h_{a,ex,ev,wet} - (w_{a,su,ev} - w_{a,ex,ev,wet}) \cdot c_{w,ev} \cdot t_{c,ev,wet}) \tag{9}$$

$$\dot{Q}_{ev,wet} = \varepsilon_{ev,wet} \cdot \dot{C}_{a,ev,wet} \cdot (t_{wb,su,ev} - t_{ev,mean,wet}) \tag{10}$$

The air side thermal resistance and the fictitious specific heat are defined by Equations 11 and 12:

$$R_{af,ev} = R_{a,ev} \cdot \frac{cp_{a,su,ev}}{cp_{f,a,ev}} \tag{11}$$

$$cp_{f,a,ev} = \frac{h_{a,su,ev} - h_{a,ex,ev,wet} - (w_{a,su,ev} - w_{a,ex,ev,wet}) \cdot c_{w,ev} \cdot t_{c,ev,wet}}{t_{wb,su,ev} - t_{wb,ex,ev,wet}} \tag{12}$$

The air state at the evaporator exhaust is calculated by identifying an air-side fictitious contact effectiveness:

$$\varepsilon_{c,ev,wet} = \frac{h_{a,ev} - h_{b,ev}}{h_{a,ev} - h_{c,ev,wet}} \tag{13}$$

$$\varepsilon_{c,ev,wet} = \frac{w_{a,ev} - w_{b,ev}}{w_{a,ev} - w_{c,ev,wet}} \tag{14}$$

$$\varepsilon_{c,ev,wet} = \frac{w_{a,ev} - w_{b,ev}}{w_{a,ev} - w_{c,ev,wet}} \tag{15}$$

$$NTU_{c,ev,wet} = \frac{1}{R_{a,ev} \cdot \dot{C}_{a,ev}} \tag{16}$$

$h_{a,ev} = h_{a,su,ev}$ (air enthalpy at the evaporator supply)

$h_{b,ev} = h_{a,ex,ev,wet}$ (air enthalpy at the evaporator exhaust)

$h_{c,ev,wet} = \mathbf{h}('AirH2O', T = t_{c,ev,wet}, P = P_{atm}, R = 1)$ (contact air enthalpy)

## 2.3. Compressor

The model used here is well adapted to the simulation of most rotary compressors. It includes heat transfers at the supply, at the exhaust and to the ambient. The suction and discharge pressure drops are neglected, as well as the lubricant circulation. The compression is considered as isentropic, up to the internal pressure and then at constant volume until the exhaust pressure. The conceptual schema of the compressor is presented in Figure 1.

The evolution of the refrigerant is decomposed into four steps:

1) Heating-up (su $\rightarrow$ su1).
2) Isentropic compression (su1 $\rightarrow$ in)
3) Compression at a fixed volume (in $\rightarrow$ex1).
4) Cooling down (ex1$\rightarrow$ex)

The evolution of the refrigerant state through the compressor is presented in Figure 2.

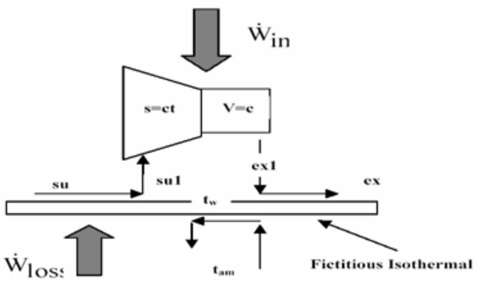


**Figure 1: Conceptual scheme of the compressor model**

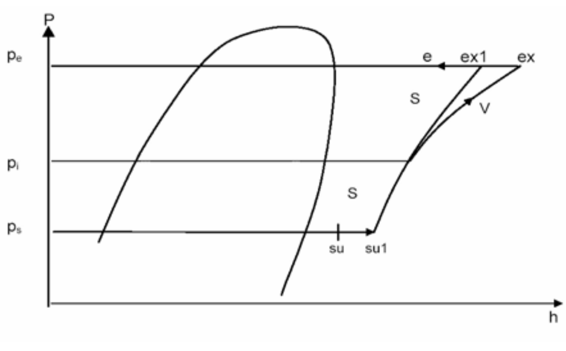


**Figure 2: Refrigerant states evolution inside the compressor**

### 2.3.1. Prediction of the refrigerant mass flow rate

The refrigerant mass flow rate is given by:

$$\dot{M}_{r,cp} = \frac{\dot{V}_{s,cp}}{v_{r,su1,cp}} \tag{17}$$

$$\dot{V}_{s,cp} = \dot{V}_{s,cp,max} \cdot X_{cp} \tag{18}$$

With *X_cp* = load factor *(control variable)*

### 2.3.2. Heat Transfer

The different heat exchanges are represented by reference to a unique isothermal wall, which is supposed to be in contact with the refrigerant at both (supply and exhaust) sides and with the ambient. Electromechanical losses are also supposed to be directly transmitted (as equivalent heat) to this wall (Figure 1).

The supply wall-to-refrigerant heat transfer can be described through the following equations:

$$h_{r,su1,cp} = h_{r,su,cp} + \frac{\dot{Q}_{r,su,cp}}{\dot{M}_{r,cp}} \tag{19}$$

$$\dot{Q}_{r,su,cp} = \varepsilon_{r,su,cp} \cdot \dot{C}_{r,su,cp} \cdot (t_{w,cp} - t_{r,su,cp}) \tag{20}$$

$$\varepsilon_{r,su,cp} = 1 - \mathbf{exp}(-NTU_{r,su,cp}) \tag{21}$$

$$NTU_{r,su,cp} = \frac{AU_{r,su,cp}}{\dot{C}_{r,su,cp}} \tag{22}$$

$$AU_{r,su,cp} = 70 \cdot \left[\frac{\dot{M}_{r,cp}}{\dot{M}_{r,ref}}\right]^{0.8} \tag{23}$$

with *M_dot_r_ref* = reference refrigerant mass flow rate (0.25 kg/s)

The same set of equations is used for the exhaust heat transfer. The ambient-to-compressor heat transfer is given by:

$$\dot{Q}_{out,cp} = AU_{amb,cp} \cdot (t_{out,cp} - t_{w,cp}) \tag{24}$$

$$AU_{amb,cp} = 10 \cdot \left[\frac{\dot{M}_{r,cp}}{\dot{M}_{r,ref}}\right]^{0.8} \tag{25}$$

### 2.3.3. Wall balance

The wall balance can be established as follows:

$$\dot{Q}_{out,cp} = \dot{Q}_{r,su,cp} + \dot{Q}_{r,ex,cp} - \dot{W}_{loss,cp} \tag{26}$$

### 2.3.4. Exhaust conditions

The compression process is decomposed into two steps:

1) An adiabatic, reversible and therefore also isentropic compression, up to the adapted internal pressure
2) An isochoric evolution (compression or expansion) from internal to exhaust pressures.

The refrigerant enthalpy after this process can be calculated as follows:

$$h_{r,ex1,cp} = h_{r,su1,cp} + w_{r,in,cp} \tag{27}$$

$$w_{r,in,cp} = w_{r,in1,cp} + w_{r,in2,cp} \tag{28}$$

$$w_{r,in1,cp} = h_{r,in,cp} - h_{r,su1,cp} \tag{29}$$

$$h_{r,in,cp} = \mathbf{h}\,(\,fluid\$\,,\,s=s_{r,in,cp}, v=v_{r,in,cp}) \tag{30}$$

$$s_{r,in,cp} = s_{r,su1,cp} \tag{31}$$

$$v_{r,in,cp} = \frac{v_{r,su1,cp}}{r_{v,in,cp}} \tag{32}$$

$$w_{r,in2,cp} = v_{r,in,cp} \cdot (p_{r,ex1,cp} - p_{r,in,cp}) \tag{33}$$

with *r_v_in_cp* = compressor internal volume ratio *(parameter to be identified)*

### 2.3.5. Prediction of the compressor power

The compressor power can be split into two terms:

$$\dot{W}_{cp} = \dot{W}_{in,cp} + \dot{W}_{loss,cp} \tag{34}$$

The electro-mechanical loss can also be split into three terms:

$$\dot{W}_{loss,cp} = \dot{W}_{loss0,cp} + \alpha_{cp} \cdot \dot{W}_{in,cp} + \dot{W}_{loss0,cp} \cdot \left[\frac{N_{rot}}{N_{rot,cp,0}}\right]^2 \tag{35}$$

With *W_dot_loos0_cp* = constant electro-mechanical compressor loss;
*α_ cp* = loss factor *(parameter to by identified)*
*N_rot_cp_0* = reference rotation speed.

## 2.4. Fan(s) model

The fans are modelled with the help of similarity variables: flow, pressure and power factors. These factors can be correlated to each other by polynomial expressions:

$$\lambda_{fan,ev} = \frac{\phi_{fan,ev} \cdot \psi_{fan,ev}}{\varepsilon_{s,fan,ev}} \tag{36}$$

$$\phi_{fan,ev} = \alpha_{0,fan,ev} + \alpha_{1,fan,ev} \cdot \psi_{fan,ev} + \alpha_{2,fan,ev} \cdot \psi_{fan,ev}^2 + \alpha_{3,fan,ev} \cdot \psi_{fan,ev}^3 \tag{37}$$

$$\lambda_{fan,ev} = \beta_{0,fan,ev} + \beta_{1,fan,ev} \cdot \psi_{fan,ev} + \beta_{2,fan,ev} \cdot \psi_{fan,ev}^2 + \beta_{3,fan,ev} \cdot \psi_{fan,ev}^3 \tag{38}$$

## 2.5. Tuning of the reference models

The parameters identification process is “manual” and iterative; it is performed in two phases: The first phase consists in separate pre-tuning of the fans, heat exchangers and compressors models, with the help of default values and/or parameter identification models. In the second phase, the parameters are tuned again, in such a way to obtain the best fit with all experimental results and/or manufacturer data available.

As the room air conditioners considered here are reversible, a same model is built in such a way to allow the simulation of both operating modes.

A fair approximation consists in assuming that each refrigerant-side and air side heat transfer coefficients can be expressed as a unique function of the corresponding flow rate inside each exchanger used in both modes (evaporator and condenser).

## 3 DESCRIPTION OF THE TEST BENCH AND TEST METHOD

Testing conditions are fixed in priority according to the international standard ISO 5151. In order to cover a larger domain of use, six other combinations of test conditions are also considered. One example RAC to be tested is given in Figure 3.

**Figure 3: Example of a tested RAC**

The “indoor” test room is a volume (conform to ISO 5151), in which the required test conditions are maintained within specified tolerances (0.5 K for the temperatures). The air velocity is never exceeding 2.5 m/s near the tested equipment. The “outdoor” test room has also a sufficient volume (conform to ISO 5151) to avoid any perturbation of the “normal” air circulation pattern. The distances between all walls, except for the floor, and all equipment surfaces are larger than 0.9m. The manufacturer’s installation instructions are also carefully respected.

### 3.1. Calorimeter method

The calorimeter provides a method for determining the capacity simultaneously on both indoor and outdoor sides of the equipment. In cooling mode, this method is used only to check the air-conditioning unit performances obtained by the refrigerant-enthalpy method. Heat and water mass balances are used to determine the indoor unit capacity.

A 7000 W electrical heaters and a 4000 W steam boiler provide the sensible and latent heat respectively. The saturated steam is superheated in order to compensate the ambient heat losses with help of 800 W super-heater. A power transducer measures the sensible heat input. The latent heat input is determined by measuring, the amount of water consumed by the steam boilers.

The outdoor unit capacity is also determined in order to check indoor side heat balance. The air is cooled by a fan-coil. The cooling coil power is determined from a water side energy balance. Supply and exhaust water temperatures are measured by thermocouples located in glove fingers; the water flow rate is measured every 15 minutes with the help of a balance and continuously checked with the help of a counter.

The calorimeter is completely insulated. An insulated wall separates also the “indoor” and “outdoor” rooms of the calorimeter. The air temperatures are controlled in the air channels surrounding the calorimeter. Heat gains or losses of all calorimeter walls are determined with the help of heat flow meters. Air temperatures and relative humidity are measured at the supply of each (indoor/outdoor) unit. Air temperatures are also measured at the exhaust of each unit and at two times four levels in each room. Globe temperatures are also measured at different points. The main sizes and the general arrangement of the calorimeter are shown in Figures 4.

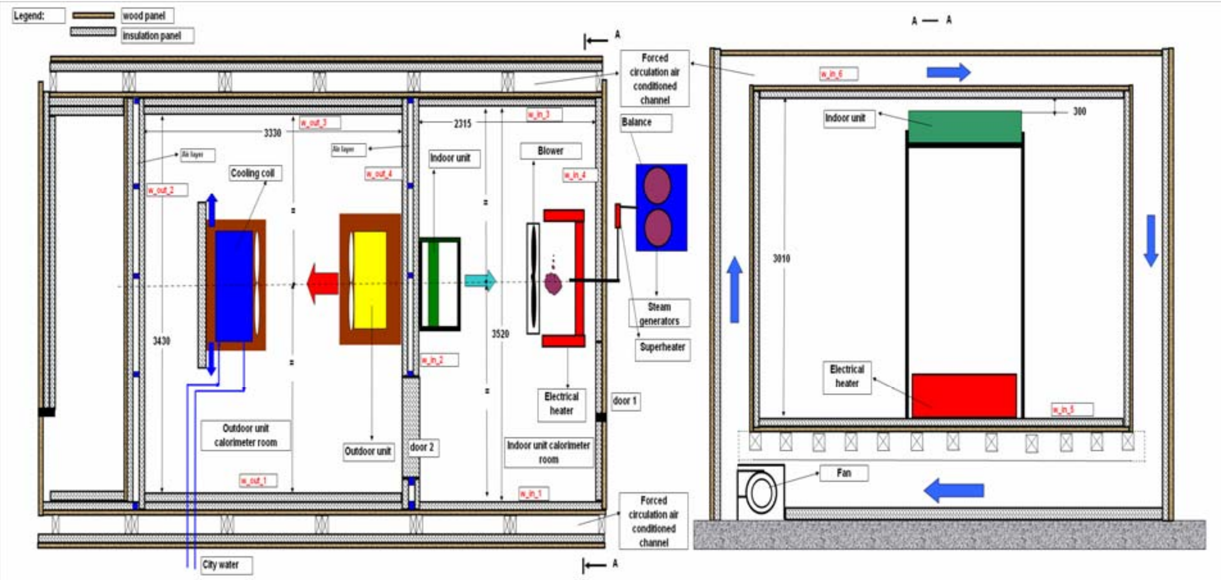


Figure 4: General view of the calorimeter rooms (cooling mode arrangement)

## 3.2. Refrigerant Enthalpy Method

Temperature and pressure measurements are used to determine the enthalpy change through each component. The refrigerant mass flow rate is measured with help of a Coriolis flow meter, located downstream of the condenser, in the liquid line (Figure 5). Enthalpy changes and flow rate are combined together in order to determine the cooling and heating capacities.

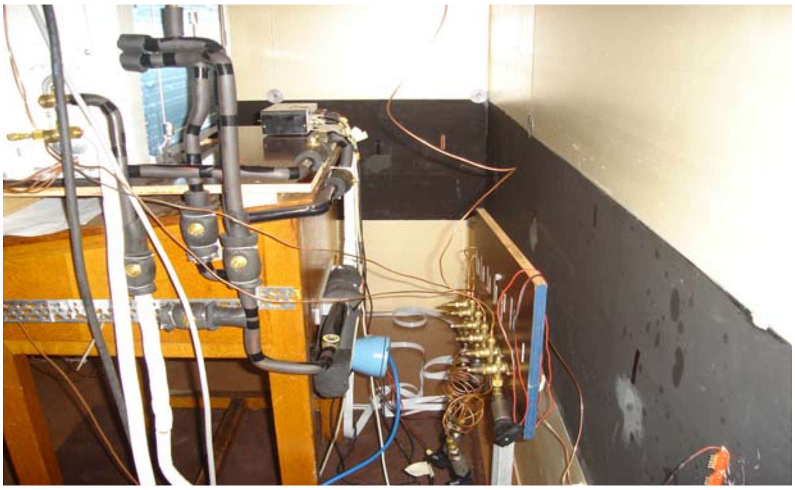

Figure 5: Flow-meter integrated in the refrigerant circuit

All measurements are recorded with a sampling rate of 1 min.

# 4 TEST RESULTS

Examples of test results obtained with one RAC are presented in Table 1.
The results available show among others that:

1) The discreancies between both experimental methods are currently not over-passing 5 %;
2) The ERR coefficient in *cooling* mode is here varying from 2.9 to 4.6, according to the class and mainly according to the conditions of use;
3) The ERR coefficient in *heating* mode is varying from 2.7 to 3.7;
4) In the case considered, passing from “medium” to “high” class of RAC is giving a significant increase of EER (about 14%) in *cooling* mode, but not in *heating* mode.

The main “bottle neck” for increasing EER seems located at the level of the condenser in both modes, i.e. at the level of the outdoor heat exchanger in cooling mode and at the level of the indoor heat exchanger in heating mode. In the present case, indeed, the main difference between “high” and “medium” classes consists in an increase of area of the outdoor heat exchanger.

| No | Test | Indoor temperature | Indoor relative humidity | Outside temperature | Indoor unit cooling power | | | Evaporating Temperature | Exhaust Air Temperature -Evaporator | SHR | Outdoor unit heating power | | | Condensing temperature | Exhaust air temperature - Condenser |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Enthalpy Method | Calorimeter. Method | Rel diff | | | | Enthalpy Method | Calor. Method | Rel diff | | |
| | | °C | - | °C | W | W | % | °C | °C | - | W | W | % | °C | °C |
| 1 | 120207A | 27,08 | 0,44 | 35,35 | 2225 | 2255 | 1,37 | 11,39 | 12,58 | 0,846 | 2666 | 2588 | -2,93 | 44,96 | 39,4 |
| 2 | 160207A | 20,76 | 0,5283 | 27,28 | 2786 | 2806 | 0,7357 | 4,186 | 5,561 | 0,703 | 3335 | 3322 | -0,3897 | 39,56 | 32,17 |
| 3 | 260207A | 28,78 | 0,4016 | 45,99 | 1860 | 1852 | -0,4195 | 15,15 | 15,93 | 1 | 2371 | 2289 | -3,46 | 54,11 | 49,79 |
| 4 | 130207A | 27,0067 | 0,31 | 33,96 | 2326 | 2379 | 2,261 | 9,816 | 10,87 | 1 | 2797 | 2692 | -3,754 | 44,03 | 38,15 |
| 5 | 140207A | 27,396 | 0,61 | 33,73 | 3262 | 3216 | -1,417 | 11,9 | 13,33 | 0,513 | 3966 | 3926 | -0,9961 | 47,5 | 39,84 |
| 6 | 220207A | 23,57 | 0,6074 | 30,42 | 2962 | 2906 | -1,892 | 8,008 | 9,356 | 0,58 | 3565 | 3634 | 1,933 | 43,36 | 35,83 |
| 7 | 210207A | 23,37 | 0,3453 | 31,01 | 2149 | 2209 | 2,785 | 6,944 | 7,994 | 1 | 2578 | 2572 | -0,2088 | 40,79 | 35,03 |
| 8 | 150207A | 27,46 | 0,4463 | 20,86 | 2370 | 2443 | 3,074 | 10,57 | 12,1 | 0,797 | 2619 | 2584 | -1,327 | 30,18 | 24,59 |
| 9 | 230207A | 20,88 | 0,5409 | 20,38 | 2397 | 2389 | -0,3558 | 5,481 | 6,698 | 0,684 | 2748 | 2792 | 1,578 | 30,4 | 24,24 |
| 10 | Dry_nom | 27,046 | 0,2723 | 36,6 | 2528 | / | / | 9,975 | 11,17 | 1 | 3132 | / | / | 47,39 | 40,99 |

| No | Test | Power Consumption of the RAC | Power consumption of the compressor | Compressor Effectiveness | EER | Compressor Rotation Speed** | Power Consumption of the evaporator fan | Level speed of the fan | Evaporator fictitious effectiveness | Evaporator fictitious global heat transfer coefficient | Condenser fictitious effectiveness | Condenser fictitious global heat transfer coefficient |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | W | W | - | - | tr/min | W | - | - | [J/K] | - | [J/K] |
| 1 | 120207A | 584,1 | 507,1 | 0,691 | 3,8 | 2320 | 11,02 | Medium | 0,8372 | 651,2 | 0,3519 | 278,5 |
| 2 | 160207A | 741,3 | 664 | 0,6951 | 3,8 | 3438 | 11,24 | Medium | 0,8718 | 600,8 | 0,3601 | 298,6 |
| 3 | 260207A | 639,8 | 561,7 | 0,6542 | 2,9 | 1974 | 12,09 | Medium | 0,943 | 414,9 | 0,3242 | 238,5 |
| 4 | 130207A | 609,2 | 530,8 | 0,7046 | 3,8 | 2522 | 12,47 | Medium | 0,9838 | 1526 | 0,3542 | 285,7 |
| 5 | 140207A | 885,9 | 809,3 | 0,6879 | 3,7 | 3481 | 10,6 | Medium | 0,8661 | 726,1 | 0,3786 | 303,9 |
| 6 | 220207A | 796,1 | 719,4 | 0,6816 | 3,7 | 3356 | 10,75 | Medium | 0,8686 | 642,7 | 0,3695 | 299 |
| 7 | 210207A | 551,4 | 473,1 | 0,7044 | 3,9 | 2417 | 12,32 | Medium | 0,8929 | 743 | 0,3506 | 276,9 |
| 8 | 150207A | 340,5 | 313,6 | 0,6299 | 7.0 | 2142 | 10,92 | Medium | 0,8201 | 571,2 | 0,3752 | 328,8 |
| 9 | 230207A | 422,9 | 346 | 0,7502 | 5,7 | 2544 | 10,88 | Medium | 0,8785 | 581,4 | 0,3609 | 311,8 |
| 10 | Dry_nom | 710,5 | 630,7 | 0,7114 | 3,6 | 2731 | 13,81 | High | 1 | 5340 | 0,3374 | 281,6 |

**Table 1: Medium class room air conditioner test results in cooling mode**

# 5 SIMPLIFIED MODEL

The simplest modelling consists in expressing the nominal cooling capacity and the corresponding electrical consumption as polynomial functions of two independent variables: the outdoor air temperature and the indoor air temperature, associated to a reference relative humidity (50%).

$$Q_{cooling,full,nom} = C1_{capacity}+C2_{capacity}*t_{in}+C3_{capacity}*t_{in}^{2}+C4_{capacity}*t_{in}^{3}+C5_{capacity}*t_{out}+C6_{capacity}*t_{out}^{2}$$
$$+C7_{capacity}*t_{out}^{3}+C8_{capacity}*t_{in}*t_{out}+C9_{capacity}*t_{in}*t_{out}^{2}+C10_{capacity}*t_{in}^{2}*t_{out}+C11_{capacity}*t_{in}^{2}*t_{out}^{2}$$

$$W_{cooling,full,nom}=C1_{power}+C2_{power}*t_{in}+C3_{power}*t_{in}^{2}+C4_{power}*t_{in}^{3}+C5_{power}*t_{out}+C6_{power}*t_{out}^{2}+C7_{power}*t_{out}^{3}$$
$$+C8_{power}*t_{in}*t_{out}+C9_{power}*t_{in}*t_{out}^{2}+C10_{power}*t_{in}^{2}*t_{out}+C11_{power}*t_{in}^{2}*t_{out}^{2}$$

Correction factors can be identified, in order to take into account the effects of the actual indoor relative humidity and of the actual fan speed:

$$\dot{W}_{cooling,full} = \dot{W}_{cooling,full,nom} \cdot C_{power,relhum} \cdot C_{speed,power}$$

$$\dot{Q}_{cooling,full} = \dot{Q}_{cooling,full,nom} \cdot C_{capacity,relhum} \cdot C_{speed,capacity}$$

## 5.1 Relative humidity correction factor

Both correction factors can be defined as functions of a relative humidity ratio:

$$ratio_{RH,in} = \frac{1}{RH_{in}}$$

An example of such correction factor is shown in **Figure 6.**

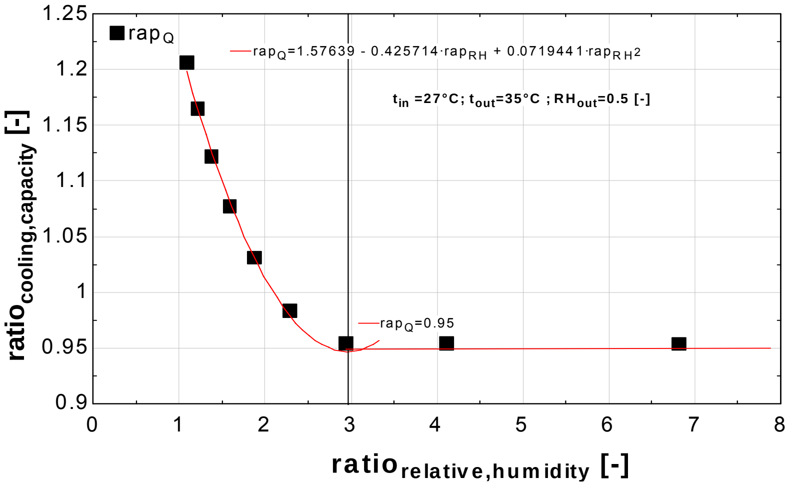


**Figure 6: Cooling capacity ratio as function of relative humidity ratio**

Two different zones of relative humidity ratio must be distinguished: 1) from 0 to 2.9, where both the cooling capacity and electrical power ratios present a (second degree) parabolic variation; 2) above 2.9, where both ratios stay constant. A similar law can be used to define the sensible heat ratio.

## 5.2 Fan speed correction factors:

A set of constant values are identified also for the fan speed correction factors:

## 5.3 Part load or over-load correction factor

In each unit considered, the compressor is driven by an inverter, which allows the cooling capacity to be adapted between 0.35 and 1.35 of full cooling capacity. The control variable is called “cooling capacity ratio” hereafter. If the cooling capacity ratio is smaller than 0.35, the RAC is working in ON-OFF. This control mode generates some performance degradations related to:

- frequent start-up of the compressor;
- system instabilities at the start-up (the steady-state condition are generally achieved after 5 – 10 min after start-up);
- (small) auxiliary power consumption (6W for the electronic device).

## 6 CONCLUSIONS

The accuracy of the reference modelling is of the order of 5%. The maximal relative difference between the two experimental methods is also of the order of 5%. EER varying from 2.9 to 4.6 and from 2.7 to 3.7 are currently reached with the two RAC's considered, in cooling and heating modes respectively. In Belgian climate, yearly EER's higner than 3 are surely achievable. Future efforts shoud be devoted to improve the heating EER in priority.

## 7 NOMENCLATURE

| | | |
|---|---|---|
| *AU* | =overall heat transfer coefficient, | W/K |
| *c* | = specific heat, | J/kg/K |
| *COP* | = performance coefficient | - |
| $\dot{C}$ | = capacity flow, | W/K |
| *EER* | = energy efficiency ratio | - |
| *h* | = specific enthalpy, | J/kg |
| $\dot{M}$ | = mass flow rate, | kg/s |
| *N* | = rotation speed | tr/min |
| *NTU* | = number of transfer units, | - |
| *P* | = pressure, | Pa |
| $\dot{Q}$ | = heat transfer, | W |
| *R* | = thermal resistance, | K/W |
| *RH* | = relative humidity, | - |
| *SHR* | = sensible heat ratio | - |
| *T* | = temperature, | °C |
| U | = peripheral speed | |
| v | = specific volume, | $m^3/kg$ |
| $\dot{V}$ | = volume flow rate, | $m^3/s$ |
| *X* | = control variable, | - |
| *w* | = humidity ratio, | - |
| *w* | = specific work; | J/kg |
| $\dot{W}$ | = power consumption, | W |

### Subscripts

| | |
|---|---|
| *a* | = air |
| amb | = ambient |
| *c* | = contact |
| *cd* | = condenser, condensing |
| *cp* | =compressor |
| *ev* | = evaporator, evaporating |
| *ex* | = exhaust |
| *f* | = fictitious |
| *in* | = internal |
| *iu* | = indoor unit |
| *m* | = metal |
| *n* | = nominal |
| *ou* | = outdoor unit |
| *r* | = refrigerant |
| *ref* | = reference |
| rot | = rotation |
| sat | = saturation |

*sens* = sensible
*sh* = superheated
*su* = supply
v = volume
w = wall
*wb* = wetbulb

**Greek Symbols**

*α* = pump or fan flow factor coefficient
*β* = pump or fan pressure factor coefficient
*ε* = effectiveness
Φ = flow factor
Ψ = pressure factor
λ = power factor

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