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# MODELING OF A ROOM AIR CONDITIONER

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# 1. Introduction

A "reference" and a "simplified" room air conditioner models are proposed in this paper.

Two examples of ("medium" and "high" classes) "split" air conditioners are considered hereafter. Both units have a nominal cooling capacity of 2500 W. In both cases also, the compressor is driven through an inverter.

The mains characteristics of these two units are presented in **Table 1**. Focus is given hereafter to the cooling mode only.

| | | | | |
|---|---|---|---|---|
| Cooling capacity | | min-nom-max | kW | 1.3 - 2.5 - 3 |
| Heating capacity | | min-nom-max | kW | 1.3 - 3.4 - 4.5 |
| Nominal input | cooling | min-nom-max | kW | 0.3 - 0.685 - 0.96 |
| | heating | min-nom-max | kW | 0.29 - 0.92 - 1.43 |
| EER | | | | 3.65 |
| COP | | | | 3.7 |
| Energy label | cooling | | | A |
| | heating | | | A |
| Anual energy consumption | cooling | | kWh | 343 |

Note: 1) VM = 1~,220~240/220-230V,50Hz/60Hz / V1 = 1~, 220-240V, 50Hz
2) Nominal cooling capacities are based on: indoor temperature 27°CDB/19°CWB • outdoor temperature 35°CDB • refrigerant piping length 7.5m • level difference 0m.
3) Nominal heating capacities are based on: indoor temperature 20°CDB • outdoor temperature 7°CDB/6°CWB • refrigerant piping length 7.5m • level difference 0m.
4) Capacities are net, including a deduction for cooling (an addition for heating) for indoor fan motor heat.
5) Units should be selected on nominal capacity. Max. capacity is limited to peak periods.
6) The sound pressure level is measured via a microphone at a certain distance from the unit (for measuring conditions: please refer to the technical databooks).
7) The sound power is an absolute value indicating the "power" which a sound source generates.

a) medium class

| | | | | |
|---|---|---|---|---|
| Cooling capacity | | min-nom-max | kW | 0.9 - 2.5 - 3.0 |
| Heating capacity | | min-max | kW | 0.9 - 3.2 - 4.5 |
| | | at -7°C | | 2.14 |
| Nominal input | cooling | min-nom-max | kW | 0.19 - 0.64 - 0.85 |
| | heating | min-max | kW | 0.2 - 0.76 - 1.3 |
| EER | | | | 3.91 |
| COP | | | | 4.21 |
| Energy label | cooling | | | A |
| | heating | | | A |

b) high class

**Table 1:** Main characteristics of the two air conditioners

## 2. Reference models

A so-called "reference" ("detailed", "mechanistic", or "mother") model is, as much as possible, based on real physics. It's an assembly of meaningful equations, describing the dominant physical phenomena, as they are understood and as they can be represented. But the model is nevertheless application-oriented: its realism is only required in a given domain of use and for relevant (input and output) variables.

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

### *2.1 Heating 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.

The coil is supposed here to behave as a fictitious semi-isothermal heat exchanger. Laminar and turbulent regimes are considered on air and "refrigerant" sides respectively.

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 three nominal thermal resistances (air side, metal and refrigerant side).

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

The main equations of this model are presented hereafter.

Air side heat balance:

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

Refrigerant side heat balance:

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

$$\Delta_{h,ev,dry} = h_{r,ex,ev,dry} - h_{r,su,ev,dry}$$

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

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

$$\dot{C}_{a,ev,dry} = \dot{M}_{a,ev} \cdot cp_{a,su,ev}$$

$$\varepsilon_{ev,dry} = 1 - \mathbf{exp}(-NTU_{ev,dry})$$

$$NTU_{ev,dry} = \frac{AU_{ev,dry}}{\dot{C}_{a,ev,dry}}$$

Definition of a weighted average temperature on the refrigerant side of the evaporator:

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

with

$$t_{ev,dry} = \frac{t_{r,su,ev,dry} + t_{r,sat,ex,ev,dry}}{2}$$

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

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

Both convective resistances are defined by reference to nominal values:

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

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

## *2.2 Cooling coil in dry and wet regimes*

This model is based on Merckel theory (combination of latent and sensible heat transfer), with a very slight adaptation: air enthalpy is here replaced by wet bulb temperature as total heat transfer potential).

Jim Braun's hypothesis is also used (replacing partially dry-wet by completely dry or completely wet regimes).

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:

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

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

The air state at the evaporator exhaust is calculated, according to a classical ASHRAE procedure, by identifying fictitious contact effectiveness (i.e. by considering a fictitious air side isothermal surface):

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

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

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

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

According to Jim Braun's proposal, the (dry or wet) regime giving the highest cooling power is selected as nearest to reality:

$$\dot{Q}_{ev} = \textbf{If}\ (\dot{Q}_{ev,dry}, \dot{Q}_{ev,wet}, \dot{Q}_{ev,wet}, \dot{Q}_{ev,dry}, \dot{Q}_{ev,dry})$$

The sensible power is the power that would be obtained with same supply and exhaust air temperatures, but without any change of air water content:

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

The latent power is defined by difference between total and sensible powers:

$$\dot{Q}_{ev,lat} = \dot{Q}_{ev} - \dot{Q}_{ev,sens}$$

## *2.3 Compressor*

The model used here is well adapted to the simulation of most rotary compressors. It includes heat transfer at the supply, at the exhaust and to the ambient. The pressure drop during the suction and discharge are neglected, as well as any 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 → su1).
2) Isentropic compression (su1 → in)
3) Compression at a fixed volume (in →ex1).
4) Cooling down (ex1→ex)

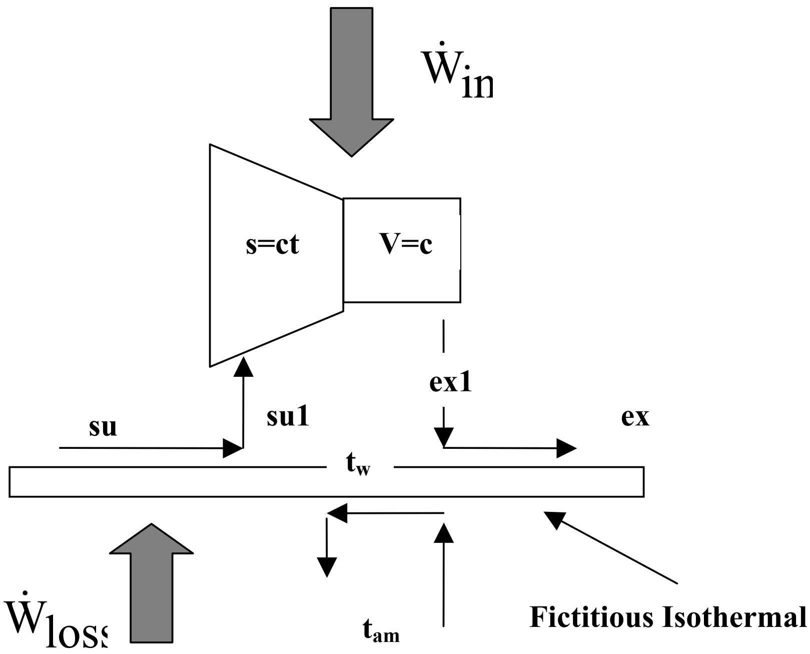



**Figure 1:** Conceptual scheme of the compressor model

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

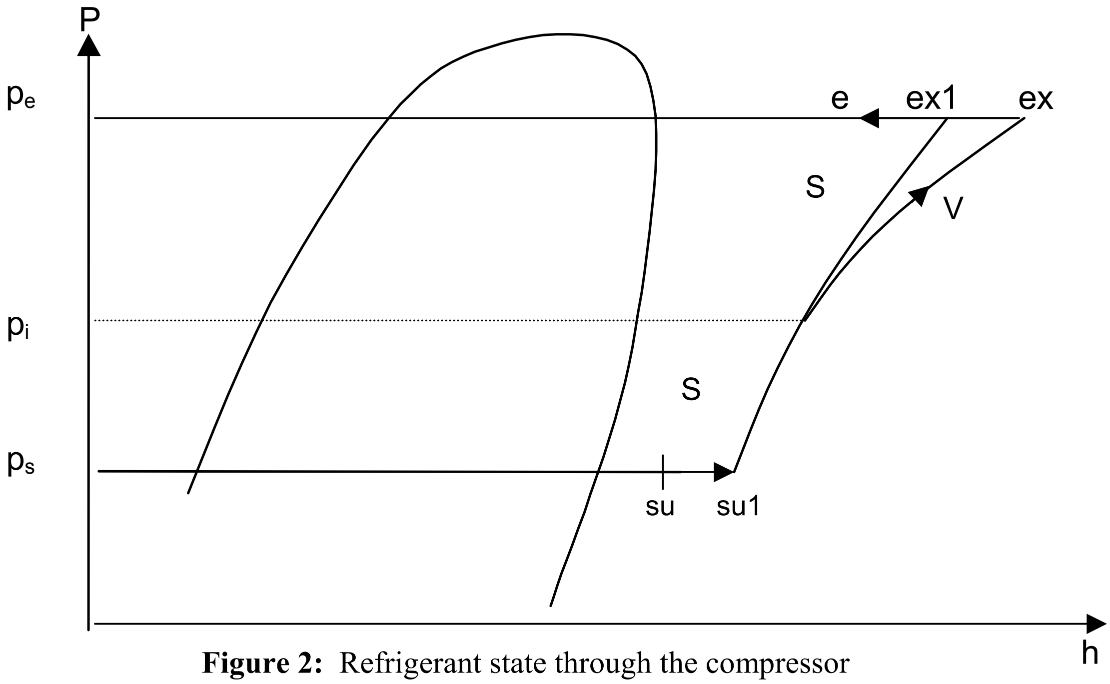



**Figure 2:** Refrigerant state through the compressor

## Refrigerant mass flow rate

The refrigerant mass flow rate is given by:

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

with

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

and with

- *M_dot_cp* - refrigerant mass flow rate [kg/s];
- *v_ su1_cp* – refrigerant volume at the compressor supply after overheating [m³/kg];
- V_dot_s_cp - refrigerant compressor flow rate [m³/s];
- *V_dot_s_cp_max* - maximal refrigerant flow rate [m³/s];
- *X_cp* – load factor (<u>*control variable)*</u> [-].

## 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:

$$h_{su1,cp} - h_{su,cp} = \frac{\dot{Q}_{su,cp}}{\dot{M}_{cp}}$$

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

$$\dot{C}_{su,cp} = \dot{M}_{cp} \cdot c_{p,su,cp}$$

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

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

with

- *Q_dot_su_cp* – supply heat transfer [W];
- *h_su_cp* – refrigerant enthalpy at the compressor supply [J/kg];
- *h_su1_cp* – refrigerant enthalpy after heating-up [J/kg];
- *c_p_su_cp* – refrigerant specific heat [J/kg-K];
- *ε_su_cp* - supply heat transfer effectiveness [-];

- *C_dot_su_cp* - thermal capacity flow rate [W/K];
- *t_w_cp* - fictitious wall uniform temperature [°C];
- *t_su_cp* – refrigerant temperature at compressor supply [°C];
- *NTU_su_cp* - number of transfer units [-];
- *AU_su_cp* – supply heat transfer coefficient [W/K].

The same set of equations is used for the exhaust heat transfer.

The ambient-to-compressor heat transfer is given by:

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

with

- *t_amb_cp* – "ambient" temperature (corresponding to the air temperature at condenser exhaust in the present case!) [°C];
- *AU_amb_cp* - fictitious ambient heat transfer coefficient [W/K].

**Wall balance**

The wall balance is give by the following relationship:

$$\dot{W}_{loss,cp} - \dot{Q}_{su,cp} - \dot{Q}_{ex,cp} + \dot{Q}_{amb,cp} = 0$$

with

- *W_dot_loss_cp* - compressor electro-mechanical loss [W];
- *Q_dot_ex_cp* – wall-to-refrigerant heat transfer [W].

**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) until the exhaust pressure.

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

$$h_{ex1,cp} = h_{su1,cp} + w_{in,cp}$$

with

$$w_{in,cp} = w_{in1,cp} + w_{in2,cp}$$

$$w_{in1,cp} = h_{in,cp} - h_{su1,cp}$$

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

$$s_{in,cp} = s_{su1,cp}$$

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

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

with:

- *h_ex1_cp* - refrigerant enthalpy after isochoric compression [J/kg];
- *w_in_cp* - internal compression work [J/kg];
- *w_in1_cp* - isentropic work [J/kg];
- *h_in_cp* - refrigerant enthalpy after isentropic compression [J/kg];
- *s_in_cp* – corresponding entropy [J/kg-K];
- *v_in_cp* –corresponding volume [m³/kg];
- *w_in2_cp* – isochoric work [J/kg];
- *p_in_cp* – internal pressure [Pa];
- *p_ex1_cp* –pressure after isochoric evolution [Pa];
- *r_v_in_cp* – compressor internal volume ratio ( *parameter to be identified*) [-].

**Prediction of the compressor power**

The compressor power can be split into two terms:

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

with

- *W_dot_in_cp* – internal power [W];
- *W_dot_loss_cp* – electro-mechanical loss [W]

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

$$\dot{W}_{loss,cp} = \dot{W}_{loss0,cp} + \alpha_{cp} \cdot \dot{W}_{in,cp}$$

with

- *W_dot_loos0_cp* – constant electro-mechanical compressor loss [W];
- *α_ cp* – loss factor (*parameter to by identified)* [-];

*2.4. Fan(s) model*

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

The main output of a fan model is the flow rate expressed here in “specific” value (in kg/s of *dry* air), as usually in air conditioning. Other outputs are: flow rate and pressure factors, exhaust air speed, total pressure difference, isentropic power and isentropic temperature increase across the fan (these two last outputs can be used as checking information).

The fan is supposed to be characterised by the diameter of its impeller (scale variable), the exhaust area and the coefficients of two polynomial correlations.

Supply air conditions (temperature, pressure and moisture content), rotation speed and supply-to-exhaust *static* pressure difference are taken as input variables.

This gives an information flow diagram such as presented in **Figure 3** (NB: the numbers contained in this diagram have nothing to do with the fans considered here).

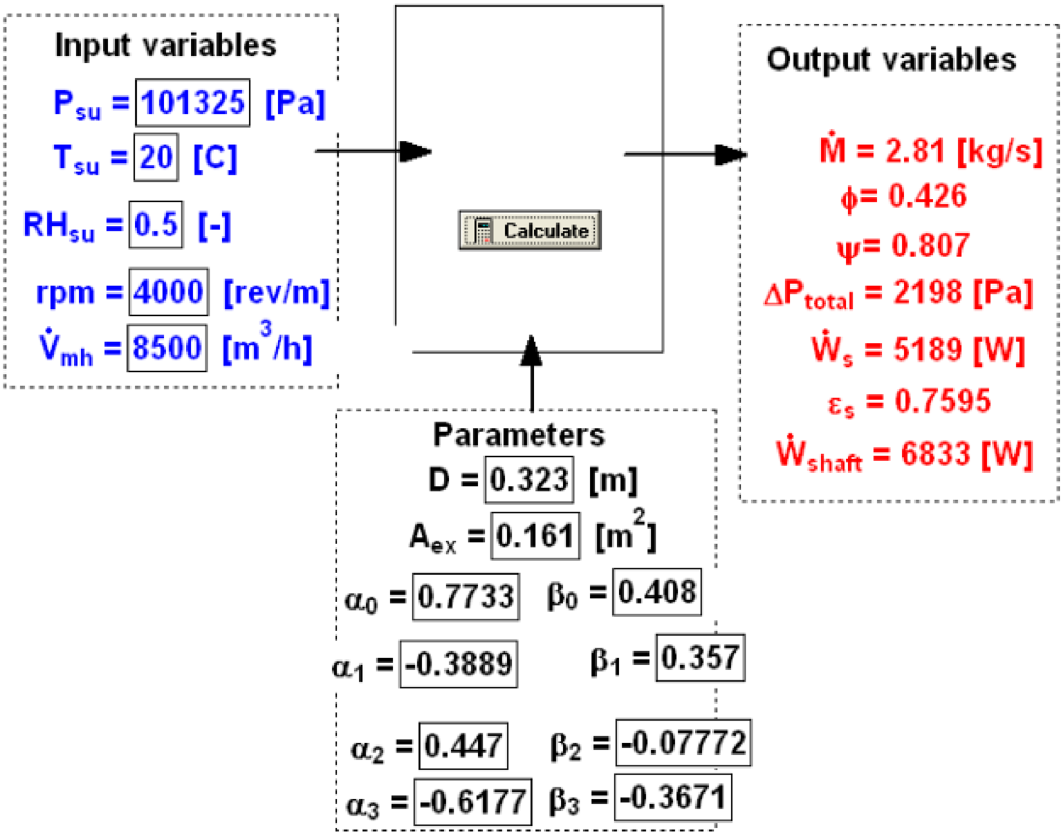



**Figure 3:** Example of fan model

The equations of this model are built on the basis of the definitions of two (flow and pressure) similarity factors:

$$\phi = \frac{\dot{V}}{A \cdot U}$$

Reference area:

$$A = \pi \cdot \frac{D^2}{4}$$

Peripheral speed:

$$U = \pi \cdot D \cdot N$$

rotation speed:

$$N = \frac{rpm}{60}$$

Pressure factor:

$$\psi = \frac{\Delta P_{total}}{P_{dynam,periph}}$$

$$\Delta P_{total} = \Delta P_{stat} + P_{dynam,ex}$$

Two dynamic pressures are considered: one at the exhaust and the other one at the periphery of the impeller:

Exhaust dynamic pressure:

$$P_{dynam,ex} = \frac{C_{ex}^{2}}{2 \cdot v}$$

$$C_{ex} = \frac{\dot{V}}{A_{ex}}$$

Peripheral dynamic pressure:

$$P_{dynam,periph} = \frac{U^2}{2 \cdot v}$$

The other non-dimensional variables considered are the isentropic effectiveness and the power factor:

Fan isentropic power:

$$\dot{W}_s = \dot{V} \cdot \Delta P_{total}$$

Isentropic effectiveness:

$$\varepsilon_s = \frac{\dot{W}_s}{\dot{W}_{shaft}}$$

Power factor:

$$\lambda = \frac{\phi \cdot \psi}{\varepsilon_s}$$

The three factors are inter-correlated through polynomial laws such as:

$$\phi = \alpha_0 + \alpha_1 \cdot \psi + \alpha_2 \cdot \psi^2 + \alpha_3 \cdot \psi^3$$

$$\lambda = \beta_0 + \beta_1 \cdot \psi + \beta_2 \cdot \psi^2 + \beta_3 \cdot \psi^3$$

This is an "orphan" model (not generated from a reference model), but from long time well validated and easy to tune…

Polynomials are fitted to manufacturer's performance data. Attention is paid to the distinction between total and static pressures: manufacturers present fan performance in terms of total pressure rise, whereas the measurements are usually made in terms of static pressures.

## 3. Tuning, validation and evaluation of reference models

### *3.1 How to make it?*

Even if "mechanistic", any component or system model has to be *tuned* before attempting to validate it.

The *validation* consists in verifying that, after having been tuned, the model can well reproduce the behaviour of the component considered in the whole domain of use.

The output of the validation procedure is an accuracy estimate (i.e. a correlation analysis between simulated and experimental data).

The quality *evaluation* must integrate, not only the validation results, but other considerations, such as tuning "easiness" (according to how much information is required and how much "skill" is required to use this information) and model "robustness" (i.e. its ability to stay realistic in all circumstances, even when going outside its tuning domain…)

One example of "tuning" and validation is presented hereafter.

## *3.1. Tuning*

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 slowly tuned, in such a way to obtain the best fit with all experimental results and/or manufacturer data available. The identification process is presented in **Figure 4**:

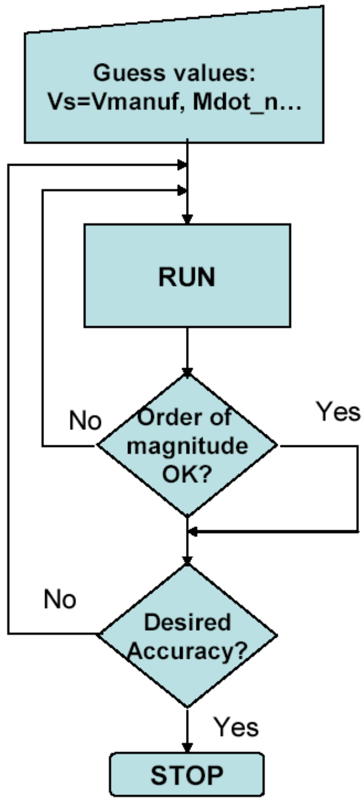



**Figure 4:** Identification process flowchart

The information flow diagram of the whole room air conditioner model is presented in **Figure 5**.

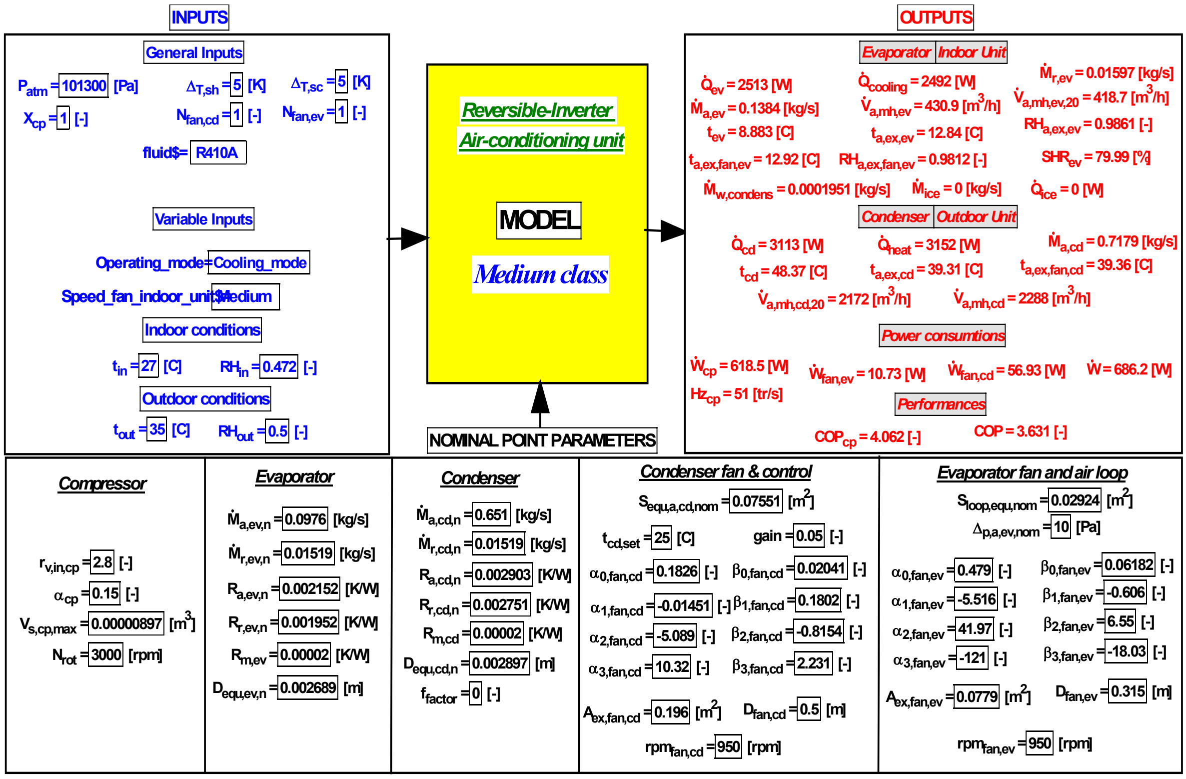


**Figure 5:** The room air conditioner model

## Phase 1: Pre-tuning of the component models

*Fans and air loops*

Generally the fans parameters are identified with help of the curves (or tables) given by the Manufacturer. An equivalent cross section area of the air loop (evaporator or condenser loop) can be defined from might be is identified from the nominal fan head announced by the manufacturer, or (as in the present case) given in the literature for a similar fan.

The utilitarian EES program used for this identification is presented in **Figures 6a and 6b**.

The coefficients $\alpha_i$ and $\beta_i$ are identified by selecting four specifics points on the fan characteristic curves available.

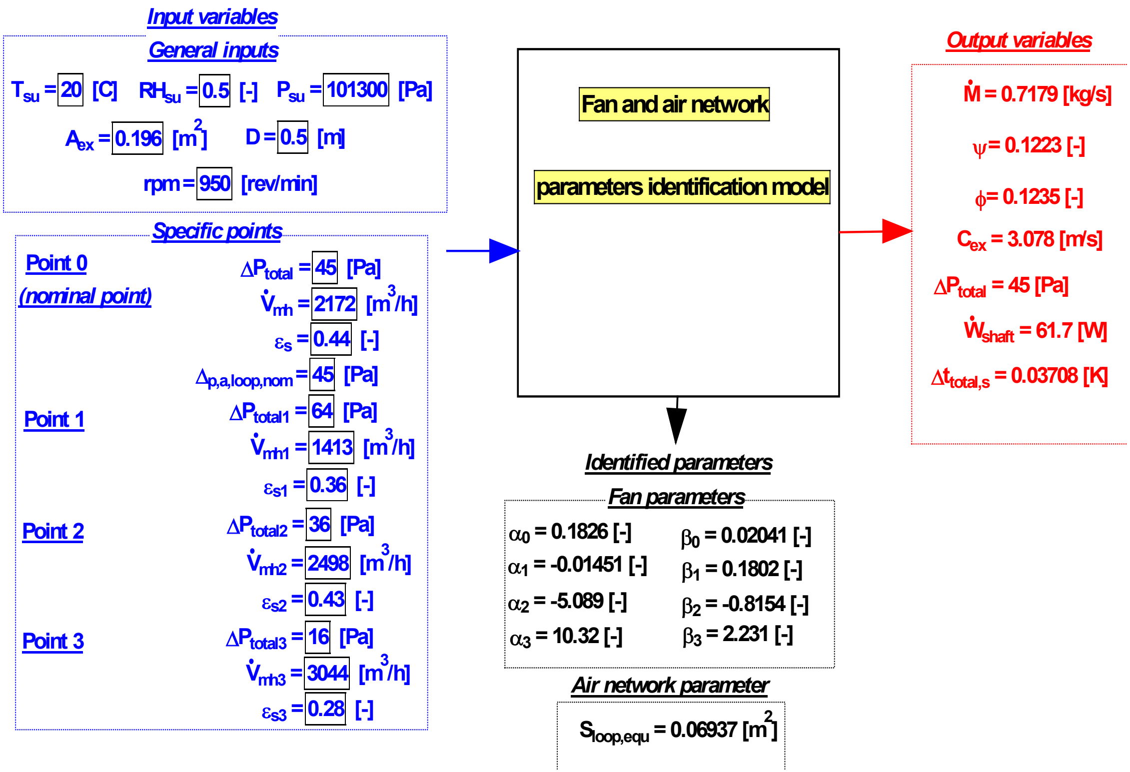



**Figure 6a:** Parameter identification for the condenser fan

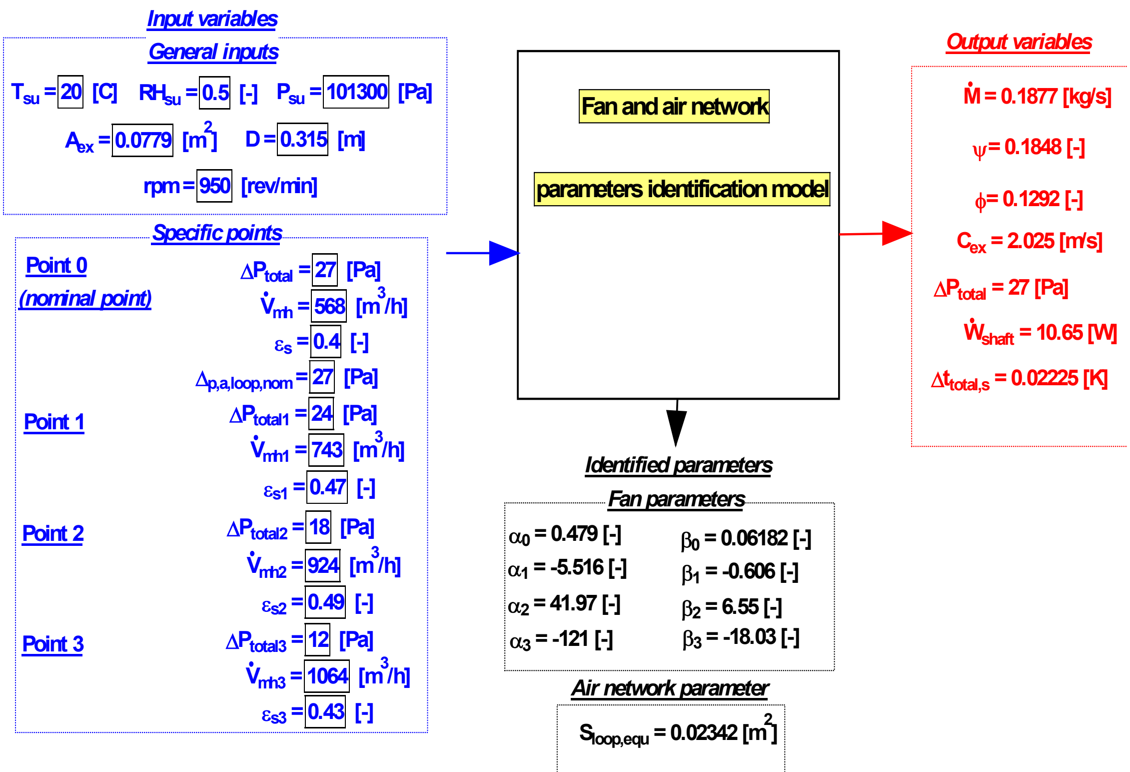



**Figure 6b:** Parameter identification for the evaporator fan

*Heat exchangers*

An other utilitarian EES program (**Figure 7**) is used to tune the parameters of both heat exchangers.

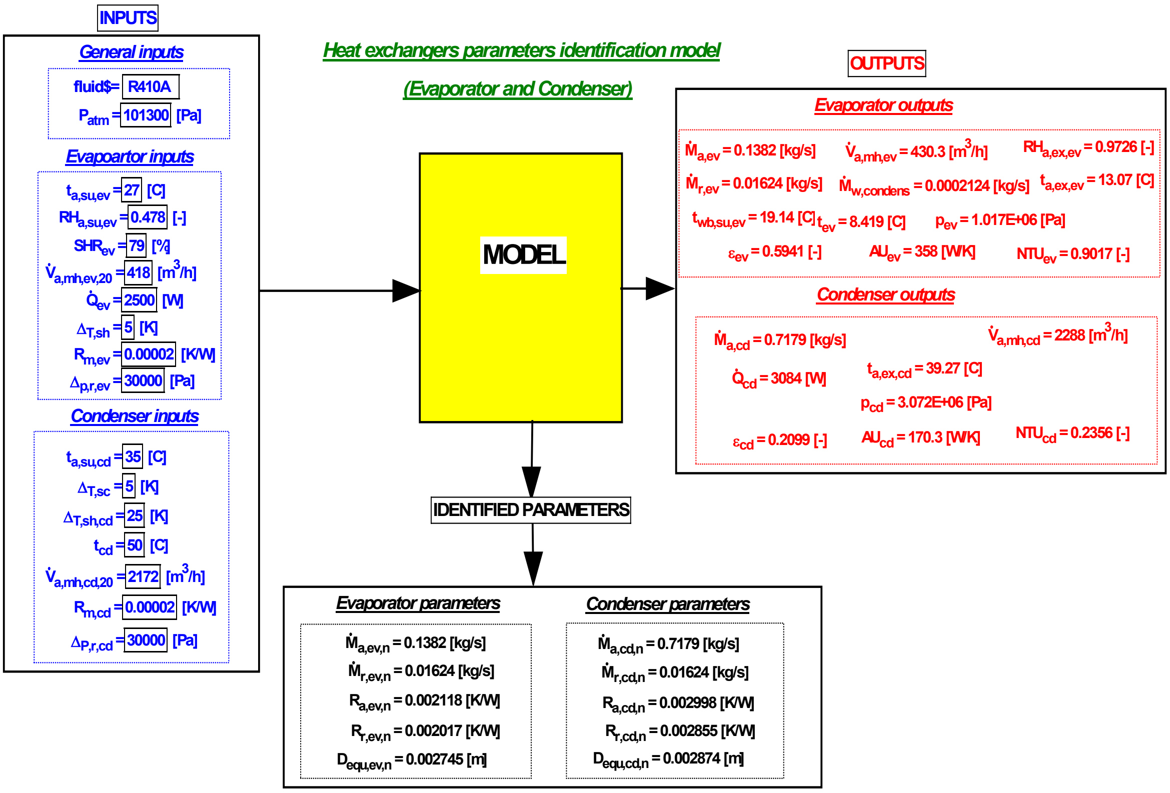



**Figure 7:** Parameter identification for both heat exchangers

The model inputs are the sizing values determined in nominal conditions (given by the manufacturer).

***General inputs :***

*fluid$* - the refrigerant used name [R407C];
*P_atm* – atmospheric pressure [Pa];

***Evaporator inputs:***

*t_a_su_ev* – dry bulb temperature at evaporator supply [°C];
*RH_a_su_ev* – relative humidity at evaporator supply [-];
*Q_dot_ev* – evaporator total cooling power [W];
*Δ_t_sh* - refrigerant superheating at evaporator exhaust [K];
*R_m_ev* - metal thermal resistance (current order of magnitue: 0.000011 K/W and 0.00002 ) [K/W];
*V_dot_a_mh_ev_20* - air flow rate at 20 °C, 50% and 101325 Pa [m³/h];
*SHR_ev* – evaporator sensible heat ratio [-];
*Δ_p_r_ev* - refrigerant pressure drop [Pa].

## Condenser inputs:

*t_a_su_cd* – supply air dry bulb temperature [°C];
*Δ_t_sc* – refrigerant subcooling at condenser exhaust [K];
*Δ_t_sh_cd* - refrigerant superheating at condenser supply [K];
*t_cd* – condensing temperature [°C];
*V_dot_a_mh_cd_20* – air flow rate at 20°C, 50% and 101325 Pa [m³/h];
*R_m_cd* - metal thermal resistance (0.00001 to 0.00002) [K/W];
*Δ_p_r_cd* - refrigerant pressure drop [Pa].

## Evaporator parameters identified:

*M_dot_a_ev_n* - nominal air mass flow rate [kg/s];
*M_dot_r_ev_n* – nominal refrigerant mass flow rate [kg/s];
*R_a_ev_n* - nominal air thermal resistance [K/W];
*R_r_ev_n* - evaporator nominal refrigerant thermal resistance [K/W];
*D_equ_ev_n* - nominal equivalent diameter (refrigerant side) [m].

## Condenser parameters:

*M_dot_a_cd_n* - nominal air mass flow rate [kg/s];
*M_dot_r_cd_n* - nominal refrigerant flow rate [kg/s];
*R_a_r_cd_n* - nominal air thermal resistance [K/W];
*R_r_cd_n* - nominal refrigerant thermal resistance [K/W];
*D_equ_cd_n* – nominal equivalent diameter (refrigerant side) [m].

## Evaporator outputs:

*M_dot_a_ev* – air mass flow rate (nominal value) [kg/s];
*V_dot_a_mh_ev* - air volume flow rate [m³/h];
*t_a_ex_ev* – exhaust dry bulb temperature [°C];
*RH_a_ex_ev* – exhaust relative humidity [-];
*M_dot_r_ev* – refrigerant mass flow rate (nominal value) [kg/s];
*M_dot_w_condens* – condensed water mass flow rate [kg/s];
*t_ev* – evaporating temperature [°C];
*p_ev* – evaporating pressure [Pa];
*ε_ev* – fictitious effectiveness [-];
*AU_ev* – fictitious global heat transfer coefficient [W/K];
*NTU_ev* – fictitious number of transfer units [-].

## Condenser outputs:

*Q_dot_cd* – condenser thermal power [W];
*M_dot_a_cd* – air mass flow rate (nominal value) [kg/s];
*V_dot_a_mh_cd* - air volume flow rate [m³/h];
*t_a_ex_cd* – exhaust dry bulb temperature [C];

*p_cd* – condensing pressure [Pa];
*ε_cd* – fictitious effectiveness [-];
*AU_cd*– fictitious global heat transfer coefficient [W/K];
*NTU_cd* – fictitious number of transfer units [-].

A starting assumption consists in giving almost the same values to air and refrigerant sides resistances. For example (in the present case):

$$R_{a,n} = R_{r,n} \cdot 1.05$$

*Compressor*

In the present case, the manufacturer doesn't give any specific information about the compressor.

A reference swept volume flow rate is iteratively identified, by starting from a default value.

Two other parameters are then identified in such a way to get correct values of both the compressor power and the amount of heat rejected through the condenser:

- *r_v_in_cp* – compressor internal volume ratio (between 2.5 and 3.5) [-];
- *ά_cp* - loss factor (between 0.2 and 0.5) [-].

To complete the parameter identification, the following similarity laws can be used:

a) Constant electro-mechanical loss [W] :

$$\dot{W}_{loss0,cp} = \dot{W}_{loss0,cp,reference} \cdot \left[\frac{V_{s,cp,max}}{V_{s,cp,max,reference}}\right]^{0.98}$$

b) Fictitious overall heat transfer coefficient at compressor exhaust [W/K]:

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

c) Fictitious ambient overall heat transfer coefficient [W/K]:

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

d) Fictitious overall heat transfer coefficient at compressor supply [W/K]:

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

with
*W_dot_loss0_cp_reference* = 500 - reference constant electro-mechanical loss [W];
*V_s_cp_max_reference =0.000213* -reference maximal fictitious volume [$m^3$].

$M\_dot\_r\_ref = 0.25$ - reference refrigerant mass flow rate [kg/s].

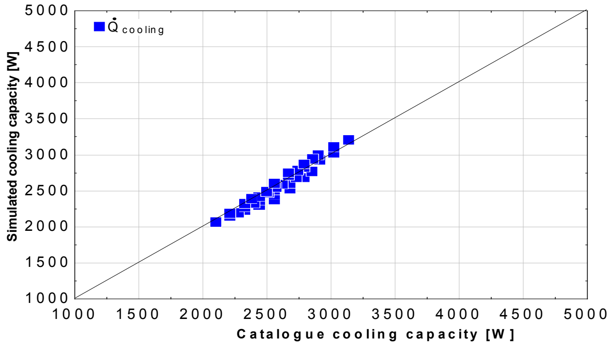
**Phase 2: Final tuning**

The compressor and heat exchangers parameters are slightly modified (**if necessary**), in order to obtain the desired results accuracy in nominal conditions:

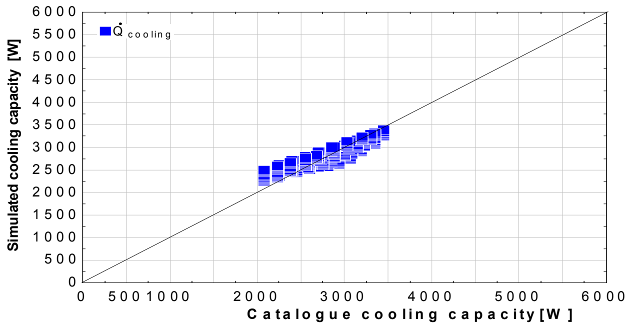
1) The swept volume is re-tuned in order to get the correct cooling power
2) The internal volume ratio is re-tuned in order to get the correct compressor electrical power
3) The evaporator air thermal resistance is also re-tuned (if necessary) in order to get the correct air exhaust temperature.

## *3.2. Validation*

By lack of other experimental data, the validation is limited here to a comparison with what is given in the catalogue of the manufacturer. The results are fairly satisfactory, as shown in **Figures 8, 9** and **10**.

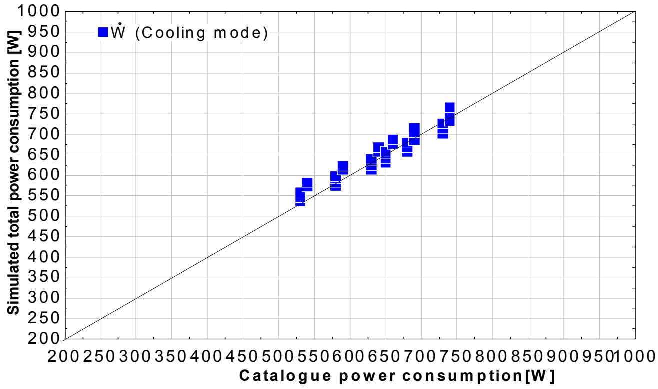



a) medium class unit

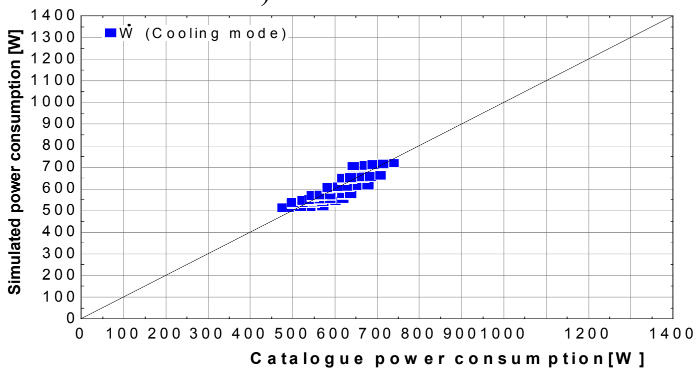



b) high class

**Figure 8:** Cooling capacity

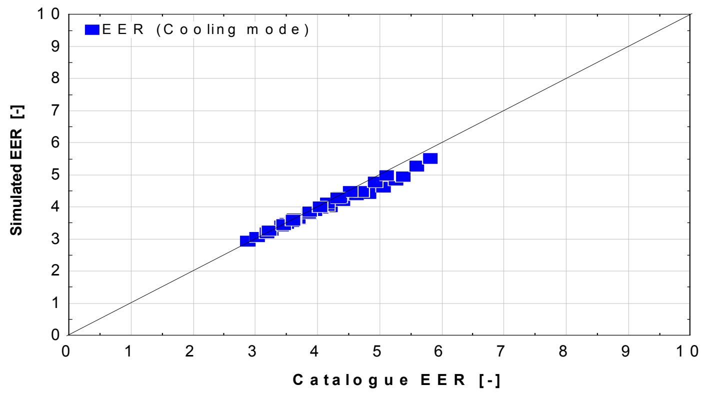



a) medium class

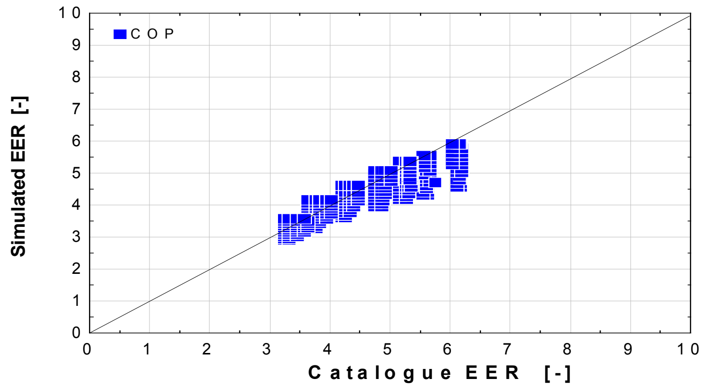



b) high class

**Figure 9:** Electrical power




a) medium class




b) high class

**Figure 10:** EER

After having been tuned and validated, this “reference” model can be used to reproduce the performances curves of the manufacturer. Examples are given in **Figure 11** and **12.**

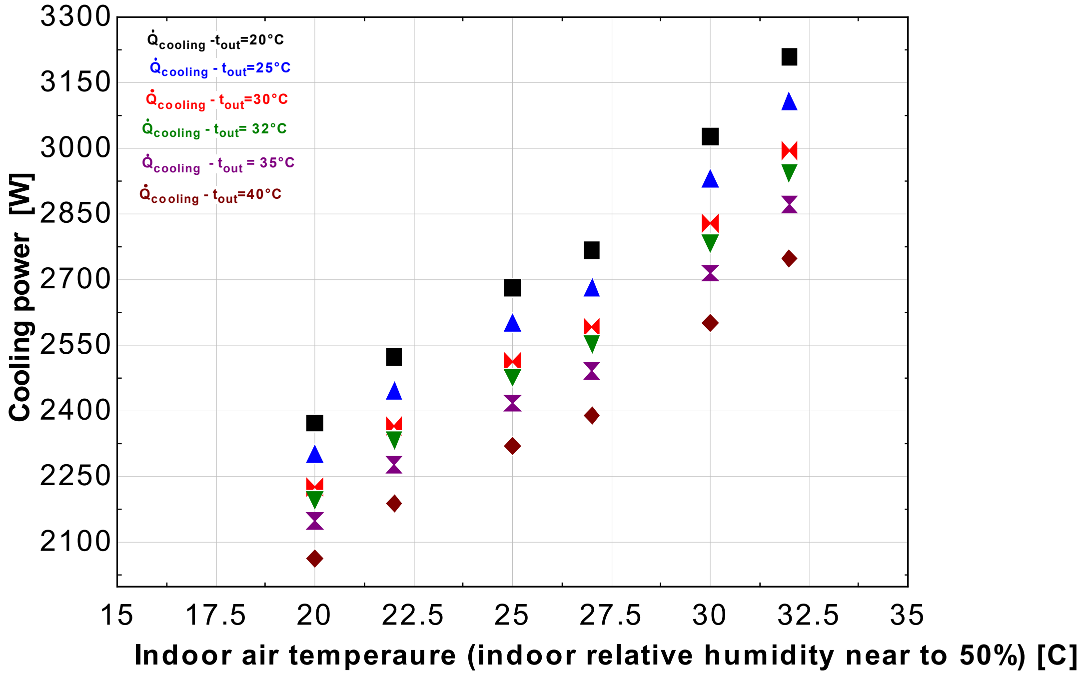



**Figure 11:** Simulated cooling capacity as function of indoor and outdoor air temperatures (medium class air conditioner with medium fan speed)

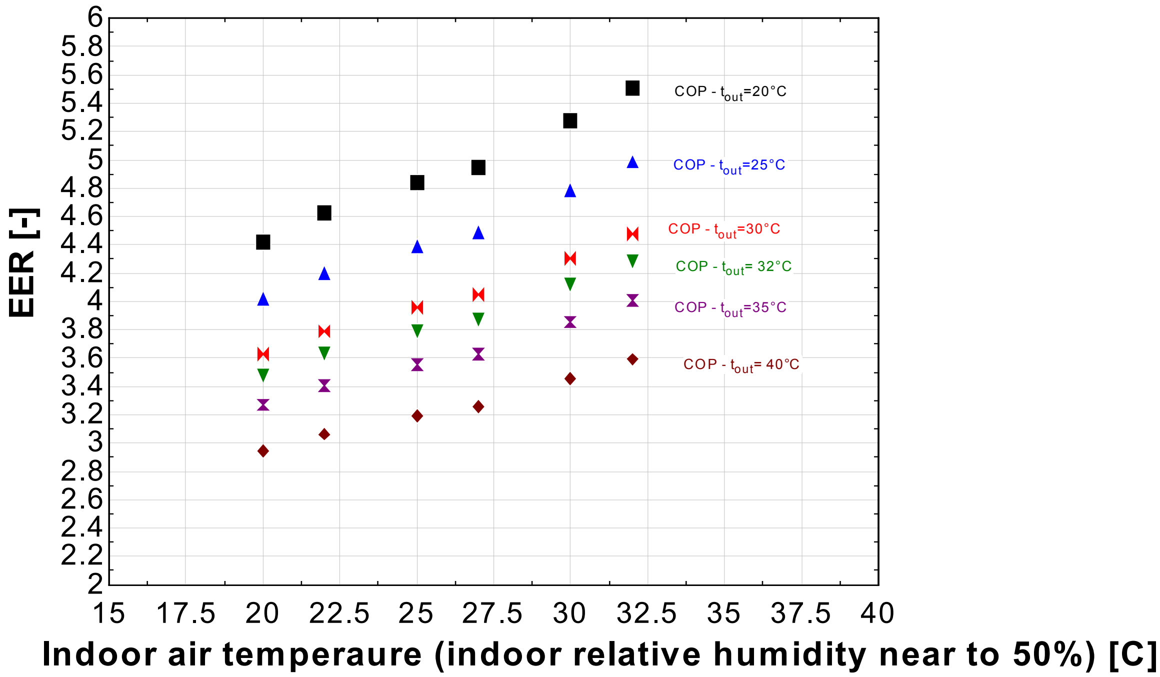



**Figure 12:** Simulated EER as function of indoor and outdoor air temperatures (medium class air conditioner with medium fan speed)

## 4. Simplified model

The simplest modeling consists in expressing the cooling full load nominal capacity and the electrical consumption as polynomial functions of two independent variables:

1) The outdoor air temperature. The relative humidity doesn't affect the system performances because the outdoor heat exchanger (condenser) works in dry regime
2) The indoor air temperature. This temperature must be then associated with a constant relative humidity (for example 50%).

The system performances are already predicted with the reference model (**Figure 11** and **Figure 12**). Third degree polynomials laws can be fitted on these results:

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

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

With the help of the reference model, these laws can be corrected in order to take into account the effects of the indoor relative humidity of the fan speed:

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

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

*Relative humidity correction factors*

Both correction factors (cooling capacity and power consumption ratios) can be defined as functions of a relative humidity ratio:

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

These functions are identified in **Figures 13** and **14:**

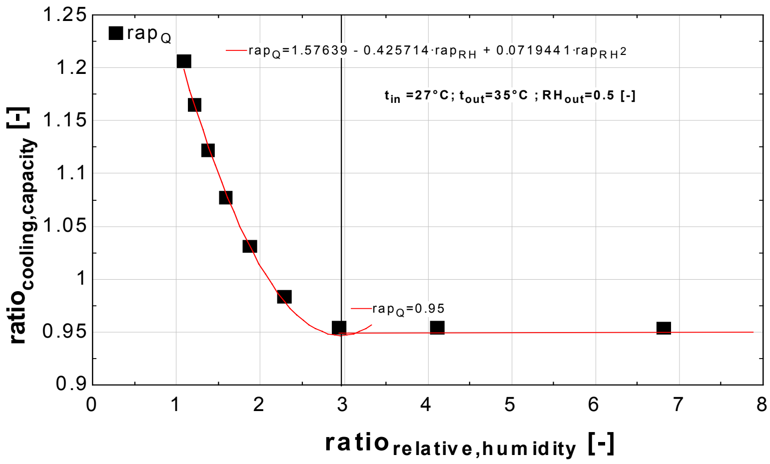



**Figure 13:** Cooling capacity ratio as function of relative humidity ratio

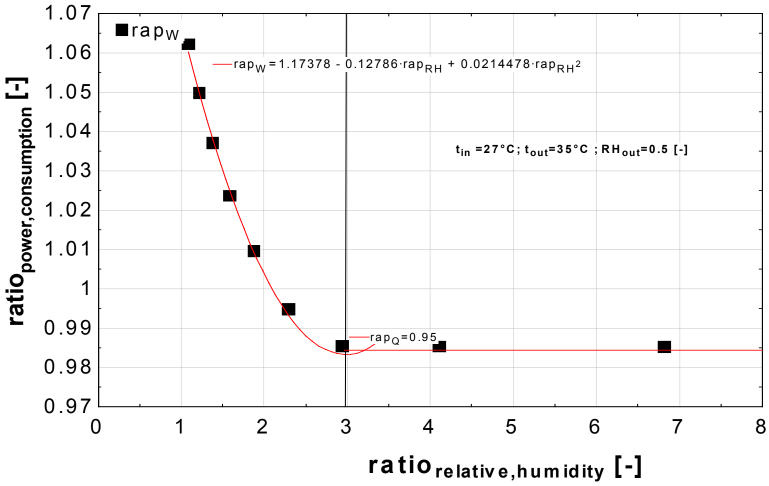



**Figure 14:** Power consumption ratio as function of relative humidity ratio

Two different zones can be distinguished:

1) from 0 to 2.9 of relative humidity ratio, where both cooling and power ratios present a (second degree) parabolic variation;
2) above 2.9 of relative humidity ratio, where the cooling and power ratios are constant.

For these two zones, the corrections factors can be defined as follows:

*Cooling capacity relative humidity correction factor:*

$C_{capacity,relhum}$ = **CoolingCapacityRHCorection** ( $ratio_{RH,in}$ )

If ( $ratio_{RH,in}$ >= 2.9 ) Then

CoolingCapacityRHCorection := 0.95

CoolingCapacityRHCorection := $1.57639 - 0.425714 \cdot ratio_{RH,in} + 0.0719441 \cdot ratio_{RH,in}^2$

*Power consumption relative humidity correction factor:*

$C_{power,relhum}$ = **PowerRHCorection** ( $ratio_{RH,in}$ )

If ( $ratio_{RH,in}$ >= 2.9 ) Then

PowerRHCorection := 0.985

PowerRHCorection := $1.17378 - 0.12786 \cdot ratio_{RH,in} + 0.0214478 \cdot ratio_{RH,in}^2$

A similar law can be used to define the sensible heat ratio (**Figure 15**):

SHR = **SensibleHeatRatio** ( $ratio_{RH,in}$ )

If ( $ratio_{RH,in}$ >= 2.9 ) Then

$$SensibleHeatRatio := 100$$

$$SensibleHeatRatio := -0.550793 + 0.95039 \cdot ratio_{RH,in} - 0.144445 \cdot ratio_{RH,in}^{\ 2}$$

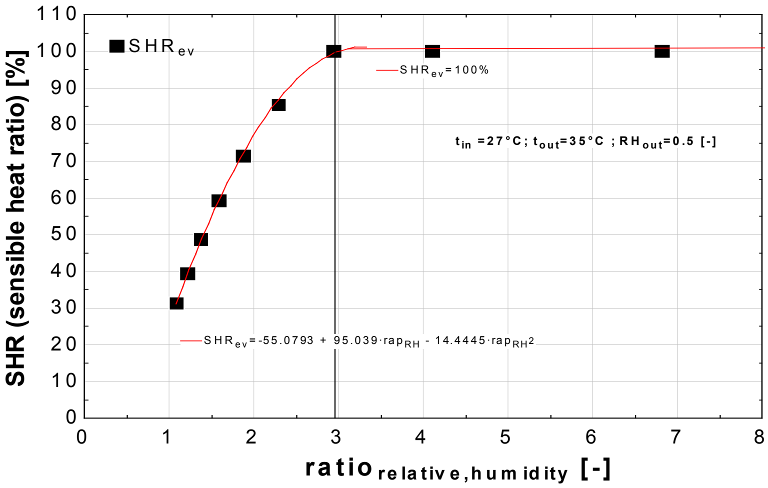



**Figure 15:** Sensible heat ratio as function of relative humidity ratio

*Fan speed correction factors*

Constant values can be identified for the fan speed correction factors:

$$C_{speed,capacity} = \mathbf{SpeedFanEvCorectionFactor}\ (Speed_{fan,indoor,unit\$})$$

If $(Speed_{fan,indoor,unit\$} = \text{'High'})$ Then

$SpeedFanEvCorectionFactor := 1.019$

If $(Speed_{fan,indoor,unit\$} = \text{'Medium'})$ Then

$SpeedFanEvCorectionFactor := 1$

If $(Speed_{fan,indoor,unit\$} = \text{'Low'})$ Then

$SpeedFanEvCorectionFactor := 0.994$

If $(Speed_{fan,indoor,unit\$} = \text{'Silent Operation'})$ Then

$SpeedFanEvCorectionFactor := 0.983$

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

If $(Speed_{fan,indoor,unit\$} = \text{'High'})$ Then

$SpeedFanCompressorFactor := 1.016$

If $(Speed_{fan,indoor,unit\$} = \text{'Medium'})$ Then

$SpeedFanCompressorFactor := 1$

If ( $Speed_{fan,indoor,unit\$}$ = 'Low' ) Then

SpeedFanCompressorFactor := 0.964

If ( $Speed_{fan,indoor,unit\$}$ = 'Silent Operation' ) Then

SpeedFanCompressorFactor := 0.953

The empirical model flow chart is presented in **Figure 18**:

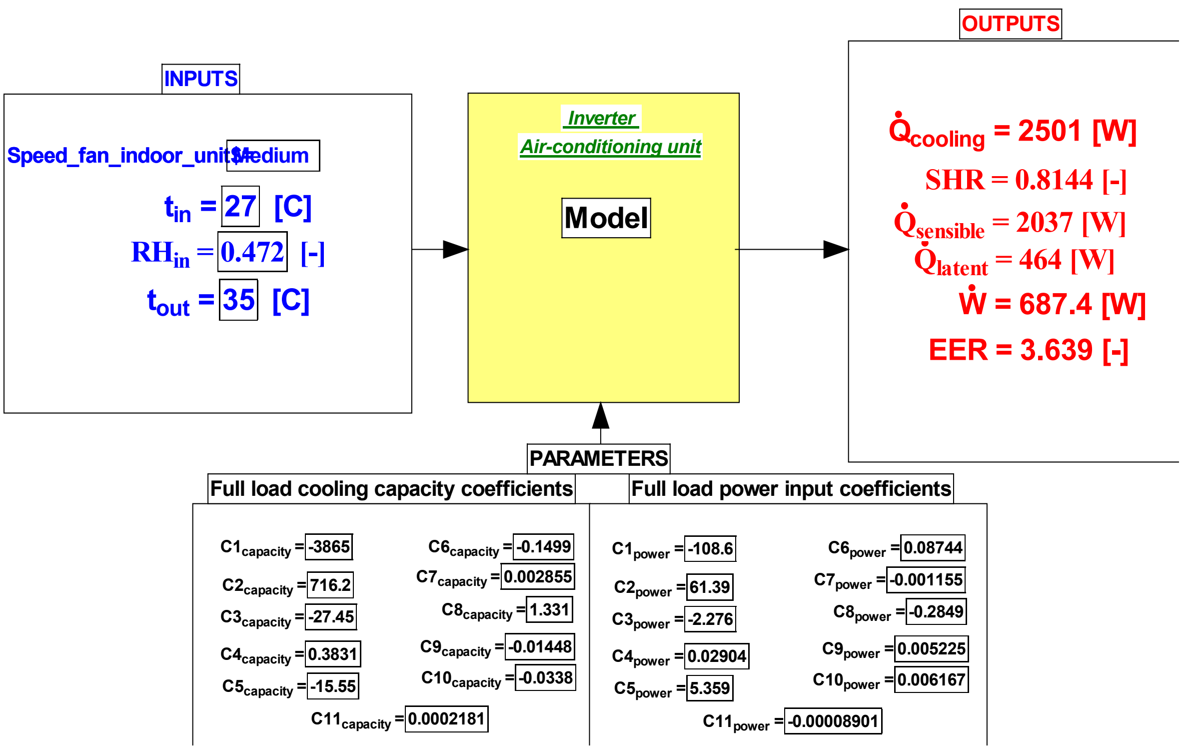



**Figure 18:** Empirical model flow chart-medium class

### *4.2 Simulation results*

Comparative simulation results (obtained with both reference and simplified models) are shown in **Figure 19** to **21**.

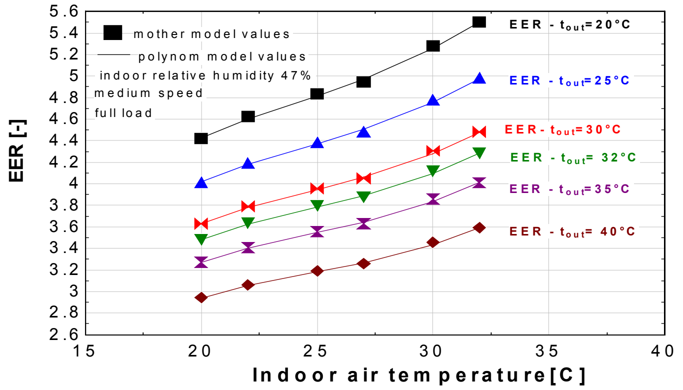



**Figure 19**: Performance coefficient as function of indoor and outdoor air temperatures (medium speed)

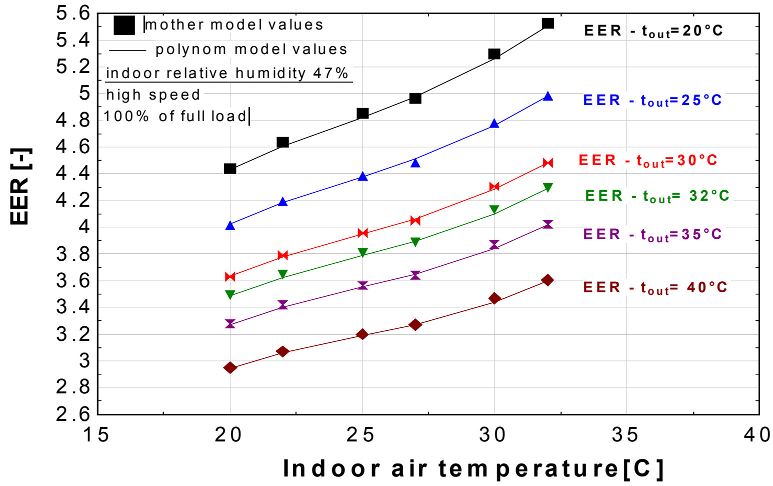
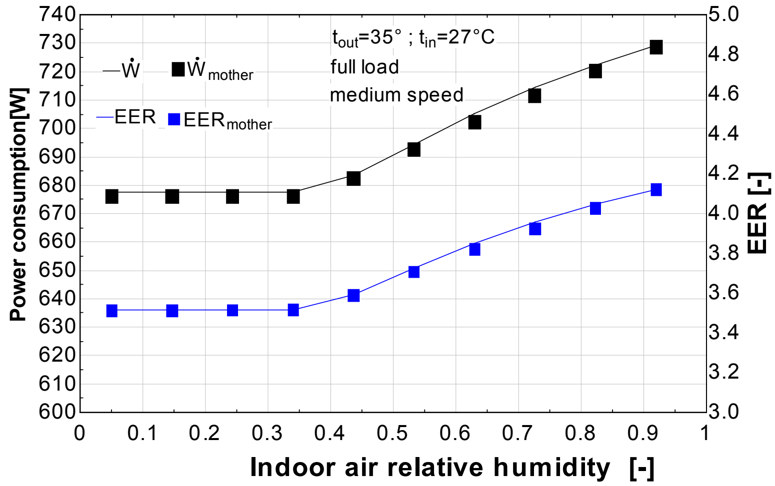
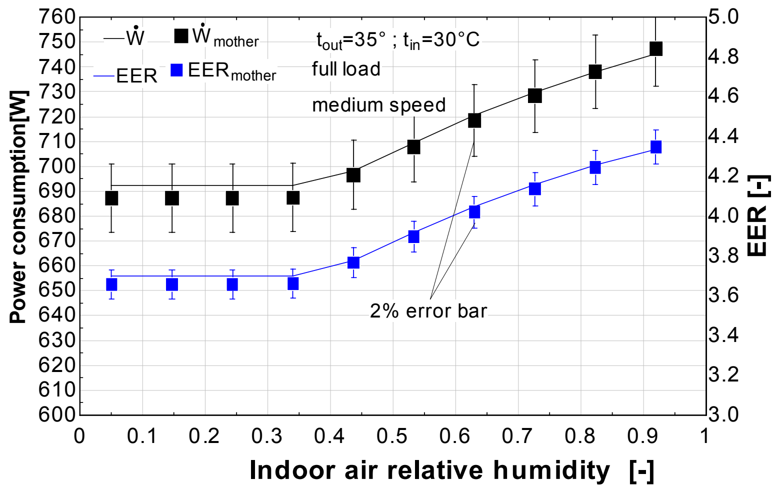


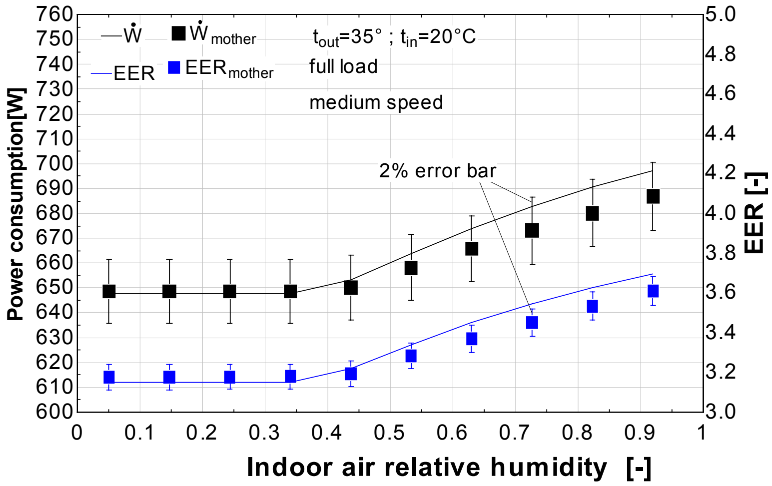

**Figure 20**: Performance coefficient as function of indoor and outdoor air temperatures (high speed)




a) t_in=27°C; t_out=35°C; full load; medium speed




b) t_in=30°C; t_out=35°C; full load; medium speed




c) t_in=20°C; t_out=35°C; full load; medium speed

**Figure 21**: Performance coefficient and power consumption as functions of indoor air relative humidity

## 5. Conclusion

Both (reference and simplified) models are fairly well reproducing the performance data given by the manufacturer.

More experimental results will be welcome to go further in the validation of the reference model.

After validation, the reference ("mother") model will be used to generate a more reliable simplified ("daughter") model if required.

## 6. References

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**Lebrun J.** – *System Simulation Synthesis Report*, University of Liège, Belgium, *report AN10 881020-RF, 1988, 90pp*

**André Ph., Aparecida C., Hannay Jules, Lebrun J., Lemort V, Teodorese I.V** - *Simulation of HVAC systems: development and validation of simulation models and examples of practical applications, Keynote presented at Mercofrio 2006, Porto Alegre, Brazil, October17-20 2006*

**Teodorese I.V.** - *Modélisation d'une machine frigorifique fonctionnant avec un mélange zéotropique. Thèse de doctorat. Liège, Belgique, 2003*

**Winandy E.L.,** - *Contribution to the performance analysis of reciprocating and scroll refrigeration compressors - Ph. D. Thesis. Conception, Chile, Decembre 1999*