Laboratoire
de Thermodynamique appliquée
Université de Liège

Département des Sciences et Gestion
de l'Environnement
Université de Liège

Bât. B49 - Parking 33
Chemin des Chevreuils, 7
B-4000 LIEGE
Belgique

Avenue de Longwy, 185
B-6700 ARLON
Belgique

Ref Nr: RAC modeling VTJL070704

MODELING OF A ROOM AIR CONDITIONER

-SUMMARY-

Vladut Teodorese and Jean Lebrun

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1. Introduction

A “reference” and a “simplified” room air conditioner models are proposed in this paper.
One example of “medium” class “split” air conditioners is considered.
In nominal *cooling* conditions (indoor air temperature 27°C, indoor air relative humidity 47% and outdoor air temperature 35°C), the capacity is 4500 W and the EER 3.1.
In nominal heating conditions (indoor air temperature 22°C and outdoor air temperature 7 °C), the capacity is 4357 W and the COP 3.4.
The part-load operating mode is ON-OFF.

2. Reference model

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. The main equations are already described in previously report RAC modelling VTJL061207.

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.

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

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

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 → su1).
2) Isentropic compression (su1 → in)
3) Compression at a fixed volume (in →ex1).
4) Cooling down (ex1→ex)

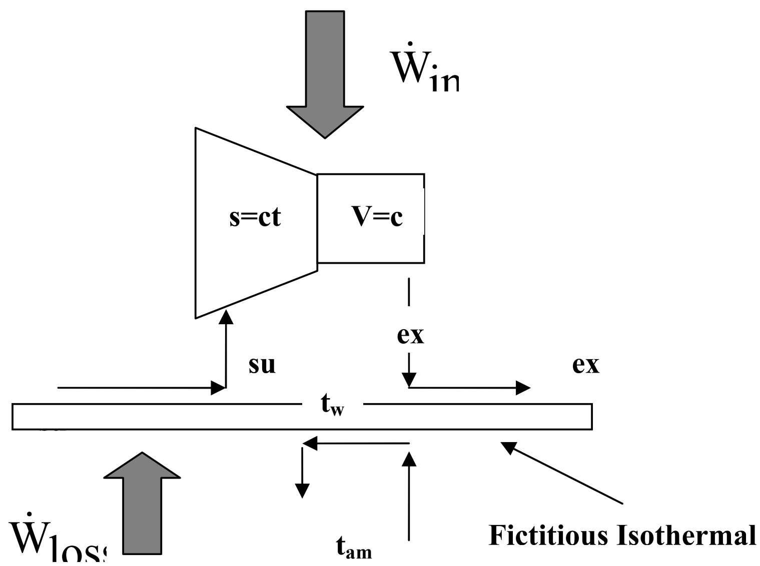

Figure 1: Conceptual scheme of the compressor model

2.4. Fan(s) model

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

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.

Polynomials are fitted to manufacturer's performance data.

3. 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 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 2**:

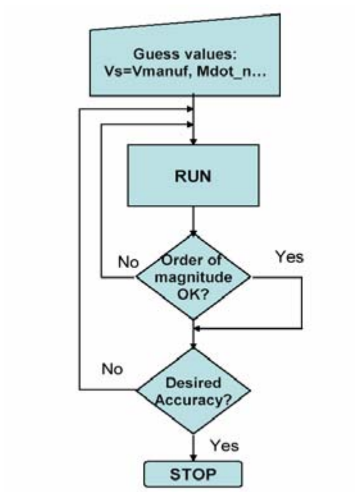

Figure 2: Identification process flowchart

Unfortunately we don't dispose of technical data other (hypothetical) nominal performances.

The following assumptions were made:

- evaporator flow rate: 530 m³/h (at 20°C and atmospheric pressure);
- condenser flow rate: 2373 m³/h (at 20°C and atmospheric pressure);
- superheating at the condenser supply: 32°C;
- compressor flow rate: 3.13 m³/h;
- compressor rotation speed: 3000 tr/min;
- indoor unit fan power consumption: 10 W;
- outdoor unit fan power consumption: 52W;
- electronic devices power consumption: 6 W.

Information flow diagrams and simulation results in nominal cooling and heating modes are given in **Figures 3** and **4**, respectively.

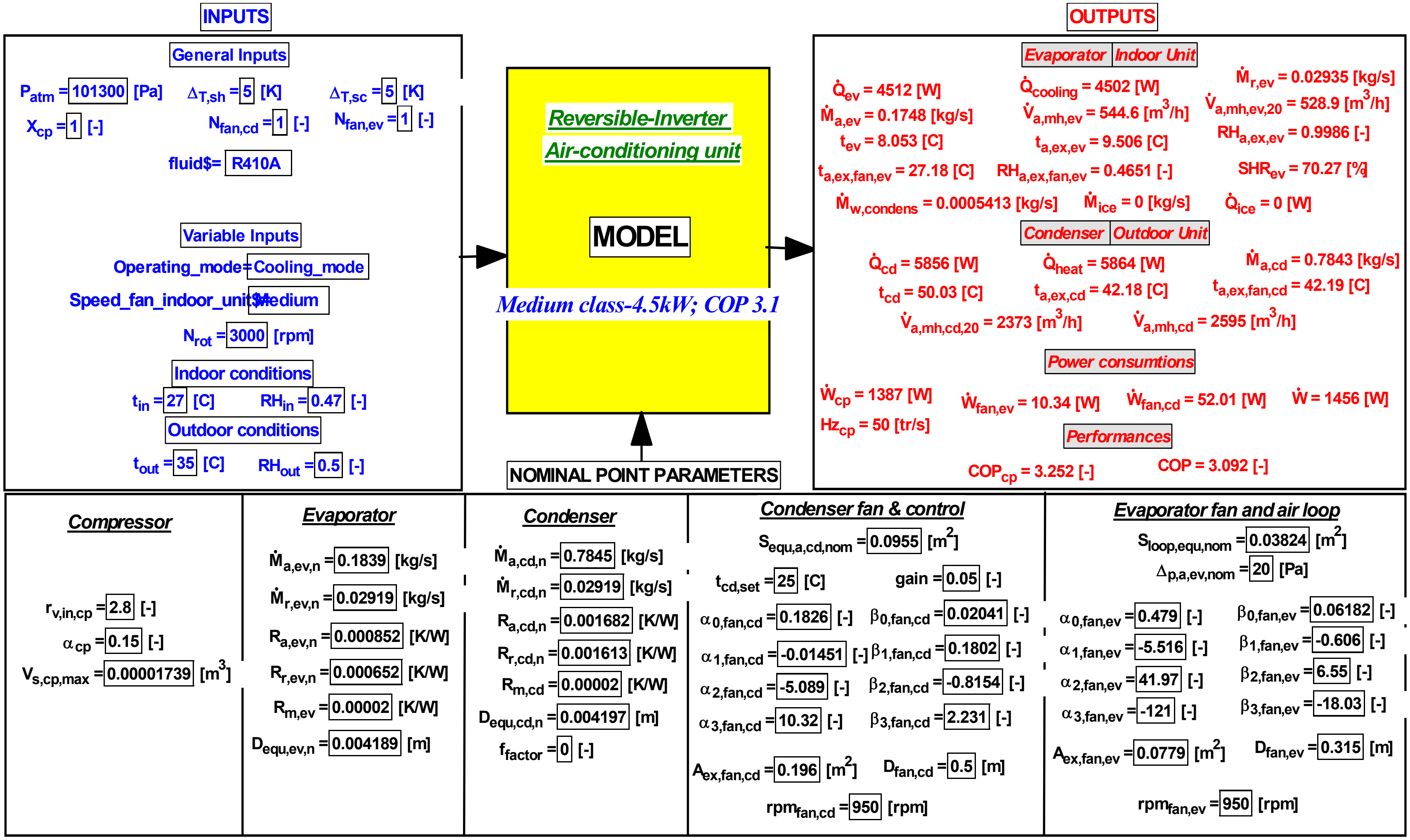

Figure 3: The room air conditioner model results: nominal cooling mode

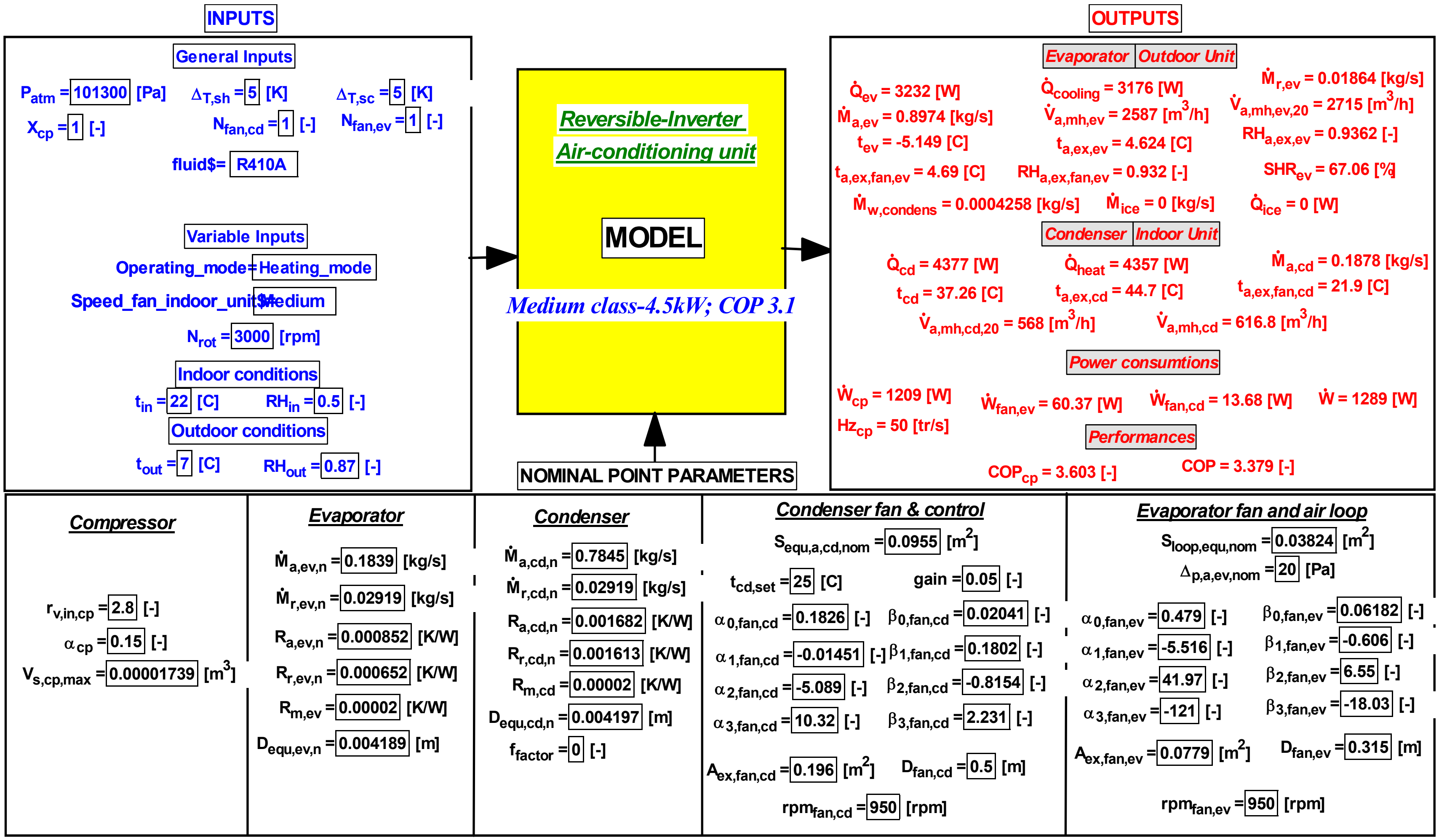

Figure 4: The room air conditioner model results: nominal heating mode

4. Simplified modelling in cooling mode

4.1 Modelling

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

1) The outdoor air temperature;
2) The indoor air temperature, associated to a reference relative humidity (50%).

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

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

With the help of the reference model, 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:

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

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

Relative humidity correction factors

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

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

These functions are identified in **Figures 5** and **6:**

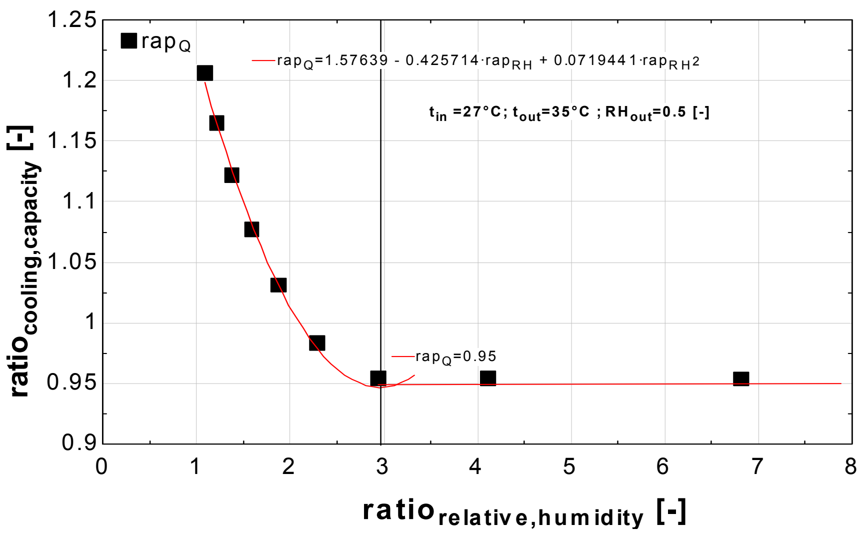

Figure 5: Cooling capacity ratio as function of relative humidity ratio

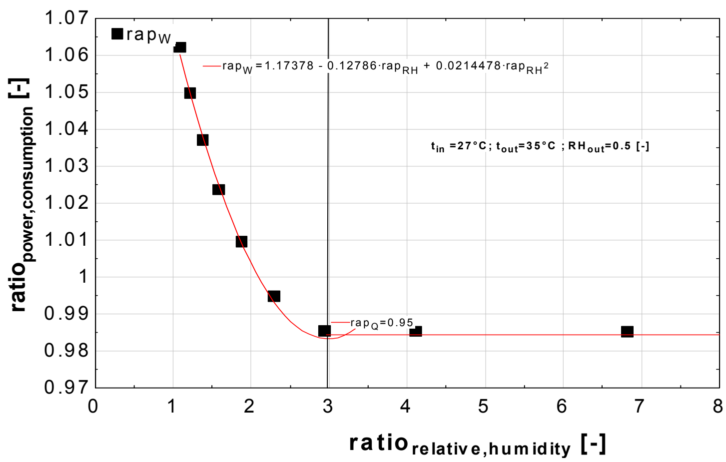

Figure 6: Power consumption ratio as function of relative humidity ratio

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.

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

For cooling capacity:

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

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

CoolingCapacityRHCorection := 0.95

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

For electrical power consumption:

$C_{power,relhum}$ = **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 7**):

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

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

$$SensibleHeatRatio := 100$$

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

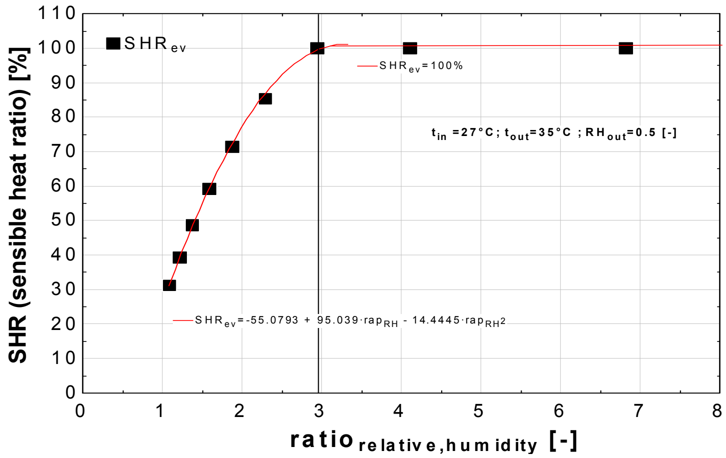

Figure 7: Sensible heat ratio as function of relative humidity ratio

Fan speed correction factors

A set of constant values can be identified for the fan speed correction factors:

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

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

SpeedFanEvCorectionFactor := 1.019

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

SpeedFanEvCorectionFactor := 1

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

SpeedFanEvCorectionFactor := 0.994

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

SpeedFanEvCorectionFactor := 0.983

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

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

SpeedFanCompressorFactor := 1.016

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

Part load correction factor

In part load, we suppose that the RAC is working in ON-OFF. This control mode generates some performances losses due 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).

On the basis of literature information gathered in **Task 4,** we suppose that the system performance degradation can be described by the curves given in **Figure 8**.

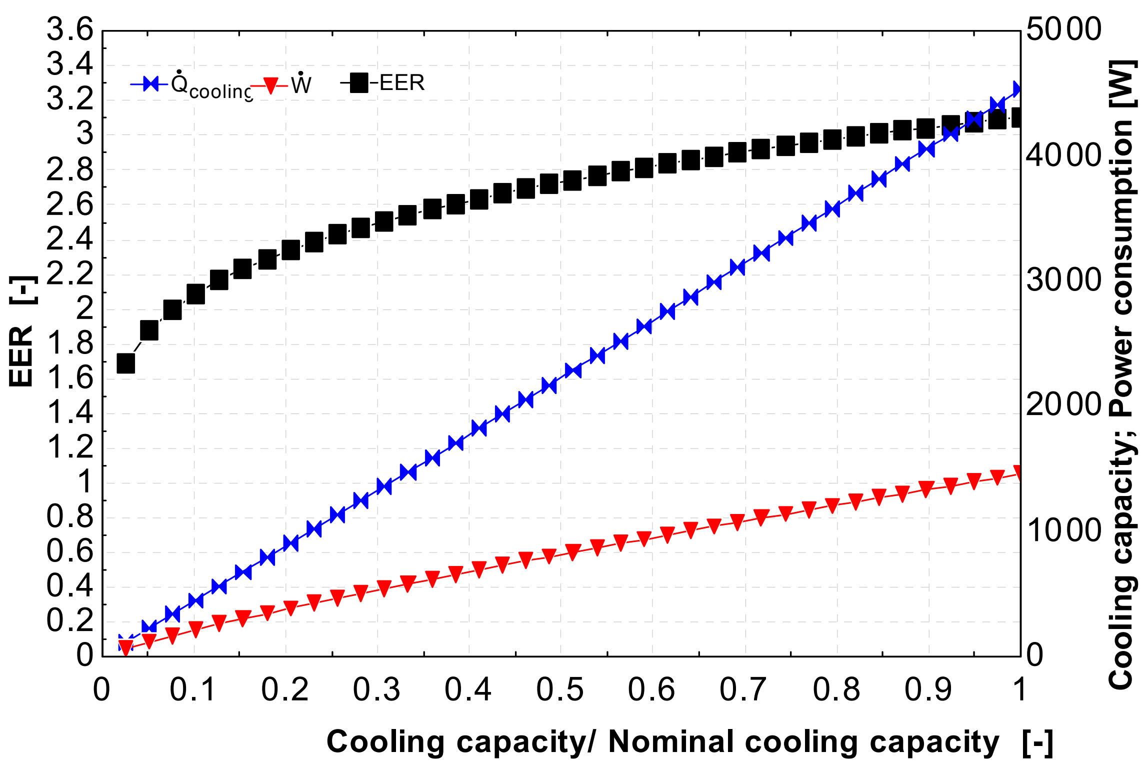

Figure 8: Part load system EER

In order to obtain such performance curves, we use as control variable the ON time and the total time of the period considered. This control variable is called "cooling capacity ratio" hereafter.

Due dynamic effects in ON/OFF regime, the cooling part load factor stays always (a little) below the above defined control variable. The ratio between these two variables is called “capacity part load correction” hereafter.

These definitions give the following expression of the cooling power as function of the full load cooling power (currently called “cooling capacity”) in given surrounding conditions:

$$\dot{Q}_{cooling} = \dot{Q}_{cooling,full} \cdot ratio_{cooling,capacity} \cdot C_{capacity,part,load}$$

$$\dot{W} = \dot{W}_{cooling,full} \cdot C_{power,part,load}$$

with

C_capacity_part_load=CoolingCapacityCorrection
C_power_part_load=CompressorPowerFactorCorection

The two correction factors are correlated to the cooling capacity ratio thanks to the following functions:

FUNCTION CoolingCapacityCorrection:

* full load

IF ratio_cooling_capacity=1 then
CoolingCapacityCorrection=1

endif

*part load

IF ratio_cooling_capacity<1 then

CoolingCapacityCorrection=

0.961006 + 0.0543636 x ratio_cooling_capacity- 0.0151515*(ratio_cooling_capacity)^2

endif
end

FUNCTION CompressorPowerFactorCorrection:

* full load

IF ratio_cooling_capacity=1 then
CompressorPowerFactorCorection=1

endif

*partload

IF ratio_cooling_capacity<1 then

CompressorPowerFactorCorrection =

(ratio_cooling_capacity$^{0.85}$)*(1.02301-0.0356631 x (ratio_cooling_capacity) + 0.0127928 x (ratio_cooling_capacity)^{2})

endif
end

The empirical model flow chart is presented in **Figure 9**:

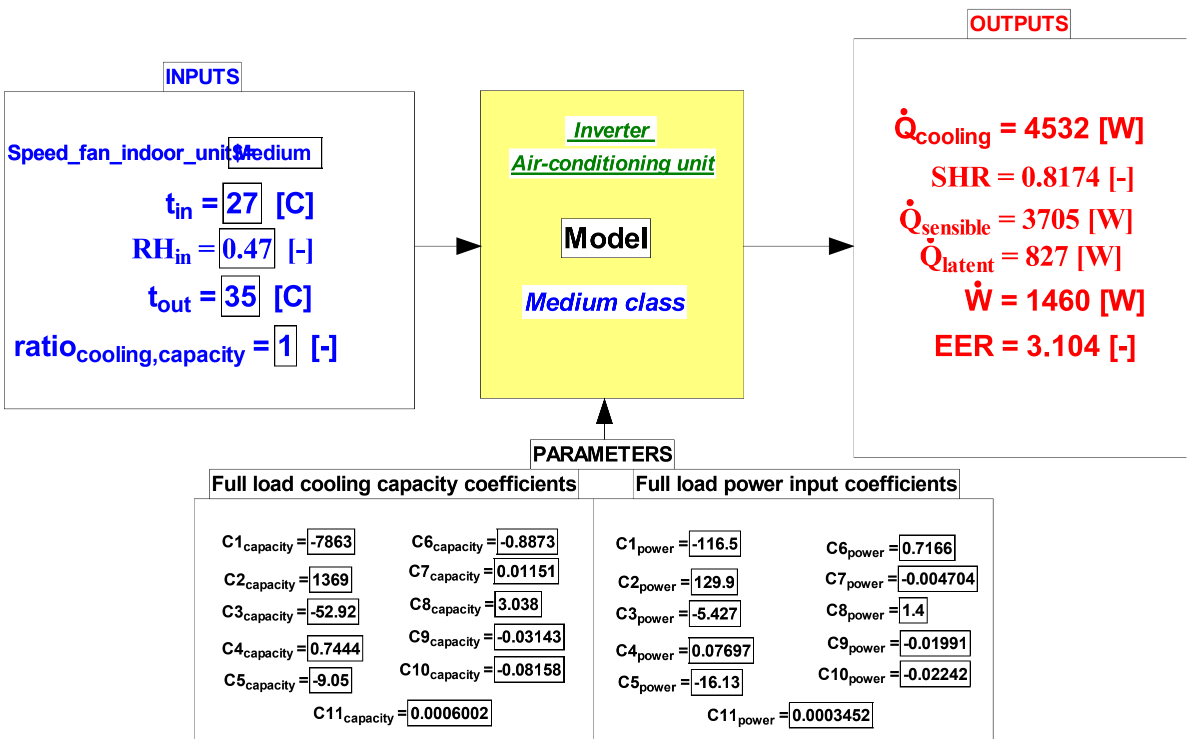

Figure 9: Empirical model flow chart (nominal cooling mode)

4.2 Simulation results

The simulation results are shown in **Figure 10** to **12**.

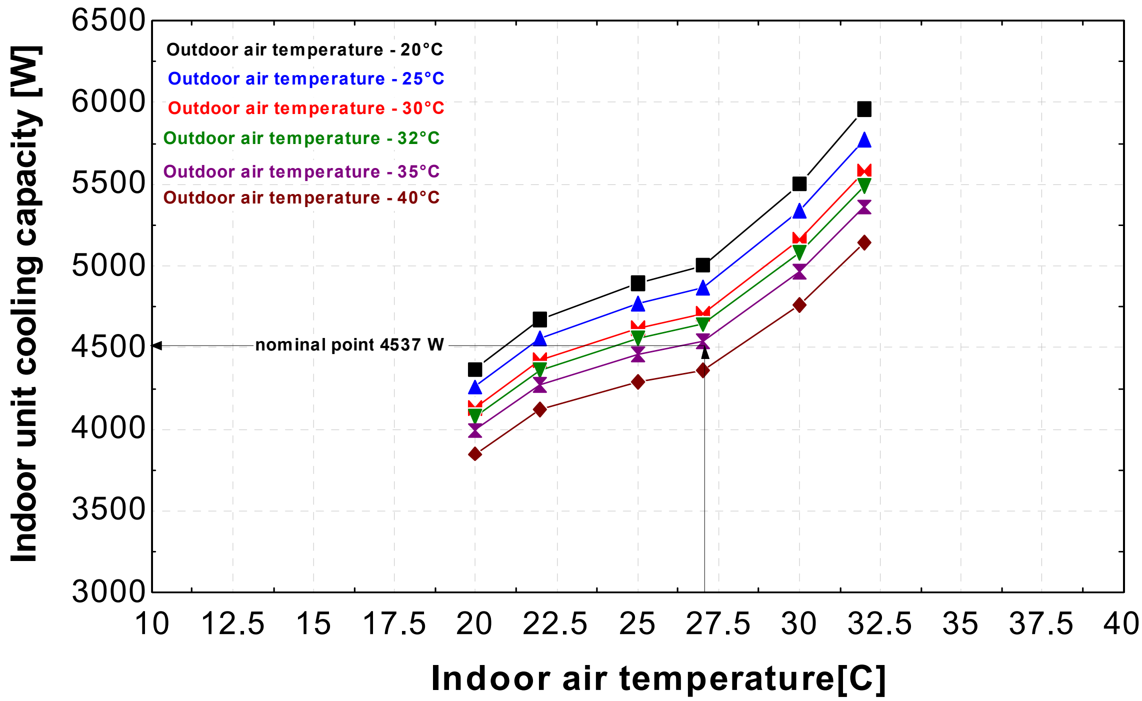

Figure 10: Cooling capacity as function of indoor and outdoor air temperatures (medium fan speed; RH_in= 47%)

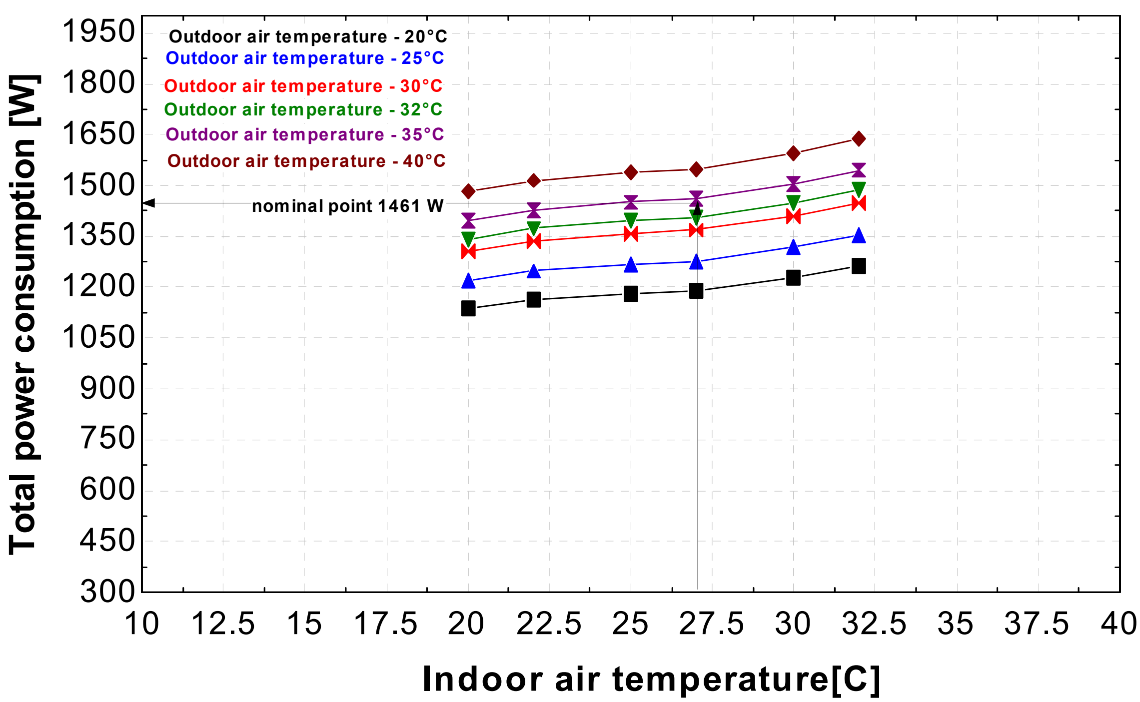

Figure 11: Total power consumption as function of indoor and outdoor air temperatures (medium fan speed; RH_in= 47%; full load)

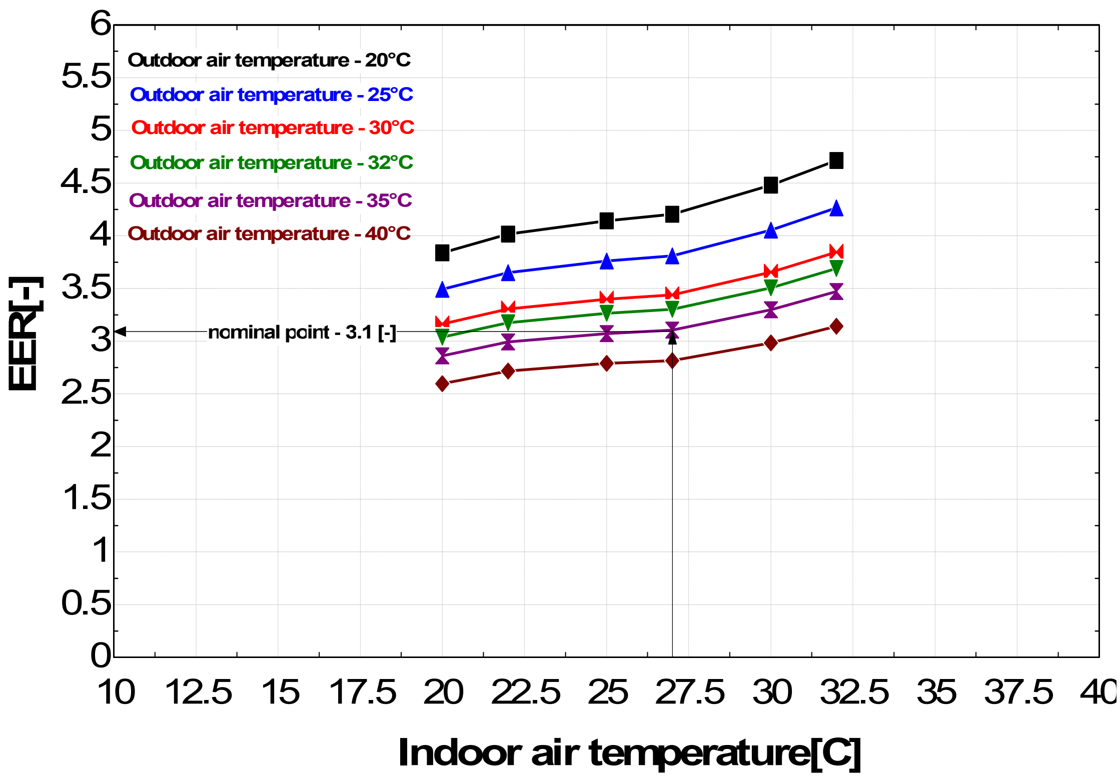

Figure 12: RAC EER as function of indoor and outdoor air temperatures (medium fan speed; RH_in= 47%; full load)

5. Simplified modelling in heating mode

5.1 Modelling

The simplest modelling consists in expressing the heating nominal capacity and the corresponding electrical consumption as polynomial functions of two independent variables:

1) The outdoor air temperature (the effect of outdoor humidity can be, neglected in fair approximation);
2) The indoor air temperature.

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

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

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

W_dot_heating_full=W_dot_heating_full_nom*C_speed_power
Q_dot_heating_full=Q_dot_heating_full_nom*C_speed_capacity

Fan speed correction factors

Again here, a set of constant values can be identified for the fan speed correction factors:

C_speed_capacity=SpeedFanCdCorectionFactor(Speed_fan_indoor_unit$)

with:

```
FUNCTION SpeedFanCdCorectionFactor(Speed_fan_indoor_unit$)
        IF  (Speed_fan_indoor_unit$='High')  then
        SpeedFanCdCorectionFactor=1.019
    endif
        IF (Speed_fan_indoor_unit$='Medium') then
        SpeedFanCdCorectionFactor=1
    endif
        IF (Speed_fan_indoor_unit$='Low') then
        SpeedFanCdCorectionFactor=0.994
    endif
        IF(Speed_fan_indoor_unit$='Silent Operation') then
        SpeedFanCdCorectionFactor=0.983
    endif
end
```

C_speed_power=SpeedFanCompressorFactor(Speed_fan_indoor_unit$)

```
FUNCTION SpeedFanCompressorFactor(Speed_fan_indoor_unit$)
        IF  (Speed_fan_indoor_unit$='High')  then
        SpeedFanCompressorFactor=1.016
    endif
        IF (Speed_fan_indoor_unit$='Medium') then
        SpeedFanCompressorFactor=1
    endif
        IF (Speed_fan_indoor_unit$='Low') then
        SpeedFanCompressorFactor=0.964
    endif
        IF(Speed_fan_indoor_unit$='Silent Operation') then
        SpeedFanCompressorFactor=0.953
    endif
  end
```

Part load correction factor

Again in part load, we suppose that the RAC is working in ON-OFF, with the same causes of performances degradations as in cooling mode, except that the auxiliary power consumption is here increased (30 W for the crank case heater).

Figure13 gives the system performance degradation function of load factor.
The same types of correlations as in cooling mode are used to reproduce this degradation.

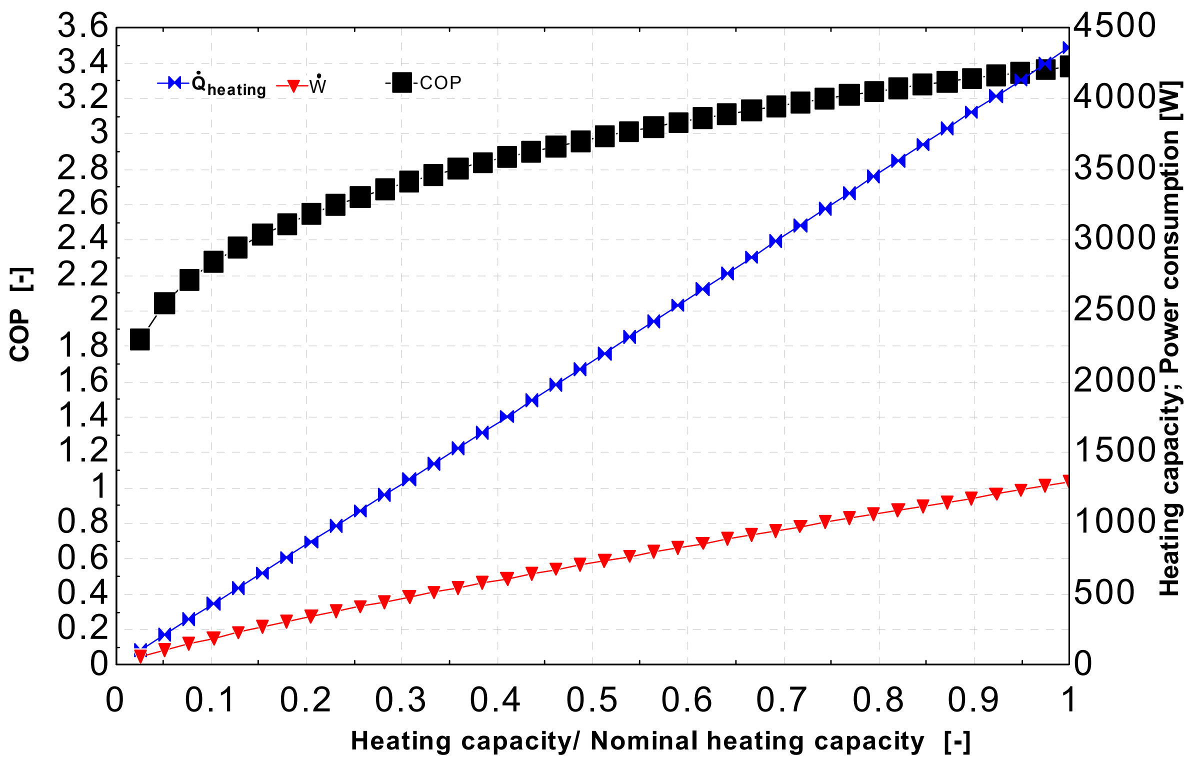

Figure 13: Part load system COP

Again here, the control variable is supposed to be the ON time fraction called hereafter "heating capacity ratio". The same two correction factors are used as in cooling mode and both factors are again correlated to the control variable:

FUNCTION HeatingCapacityCorrection:

* full load

IF ratio_heating_capacity=1 then
HeatingCapacityCorrection=1

endif

*part load

IF ratio_heating_capacity<1 then

HeatingCapacityCorrection=

0.961006 + 0.0543636 x ratio_heating_capacity- 0.0151515*(ratio_heating_capacity)^2

endif
end

FUNCTION CompressorPowerFactor:

* full load

IF ratio_heating_capacity=1 then
CompressorPowerFactorCorection=1

endif

*partload

IF ratio_heating_capacity<1 then

CompressorPowerFactorCorection =

(ratio_heating_capacity$^{0.85}$)*(1.02301-0.0356631 x (ratio_heating_capacity) + 0.0127928 x (ratio_heating_capacity)^2)

endif
end

The empirical model flow chart is presented in **Figure 14**:

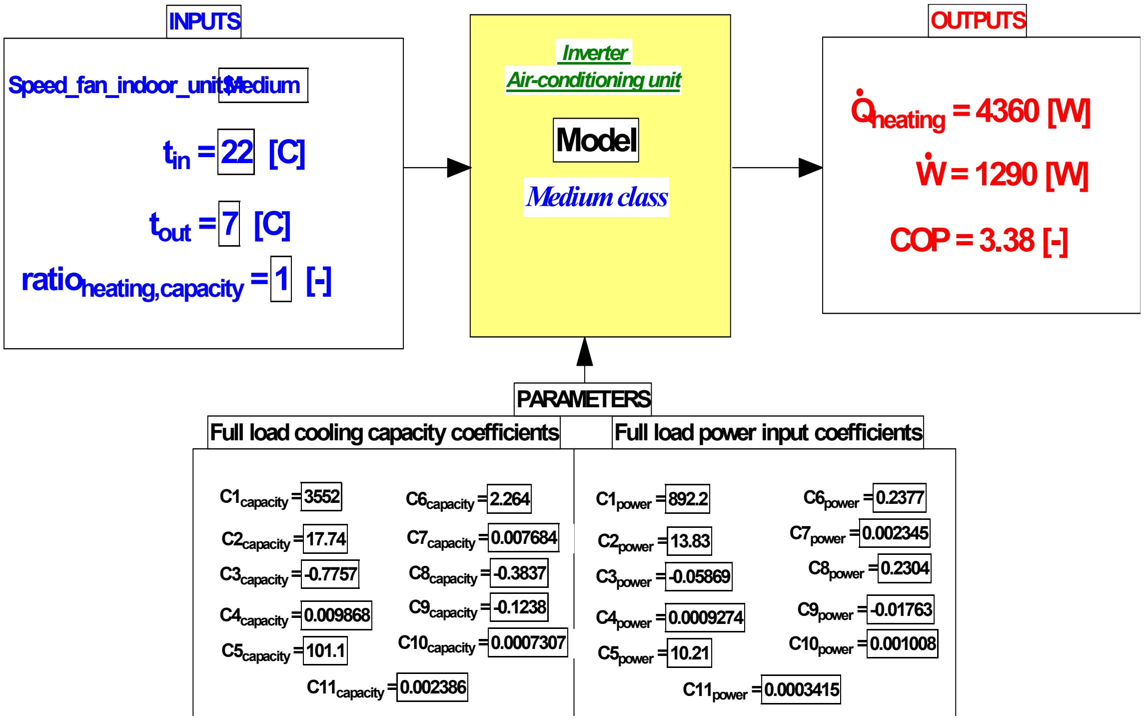

Figure 14: Empirical model flow chart (nominal heating mode)

4.2 Simulation results

The simulation results are shown in **Figure 15** to **17**.

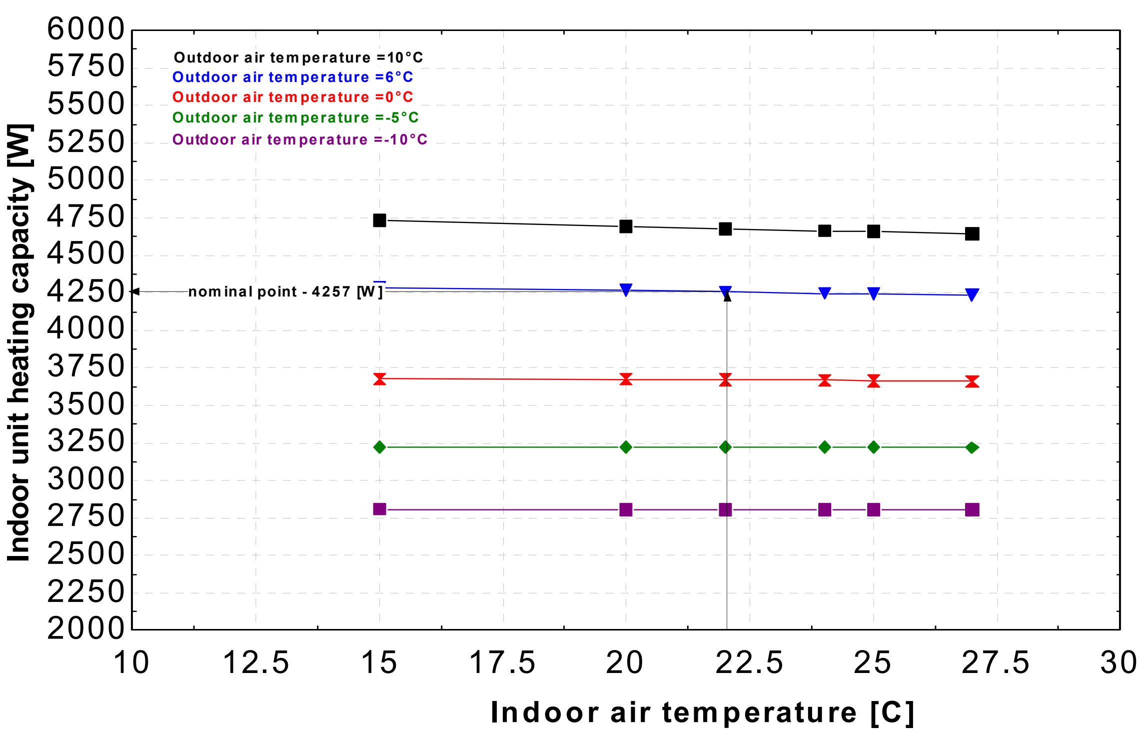

Figure 15: Heating capacity as function of indoor and outdoor air temperatures (medium fan speed)

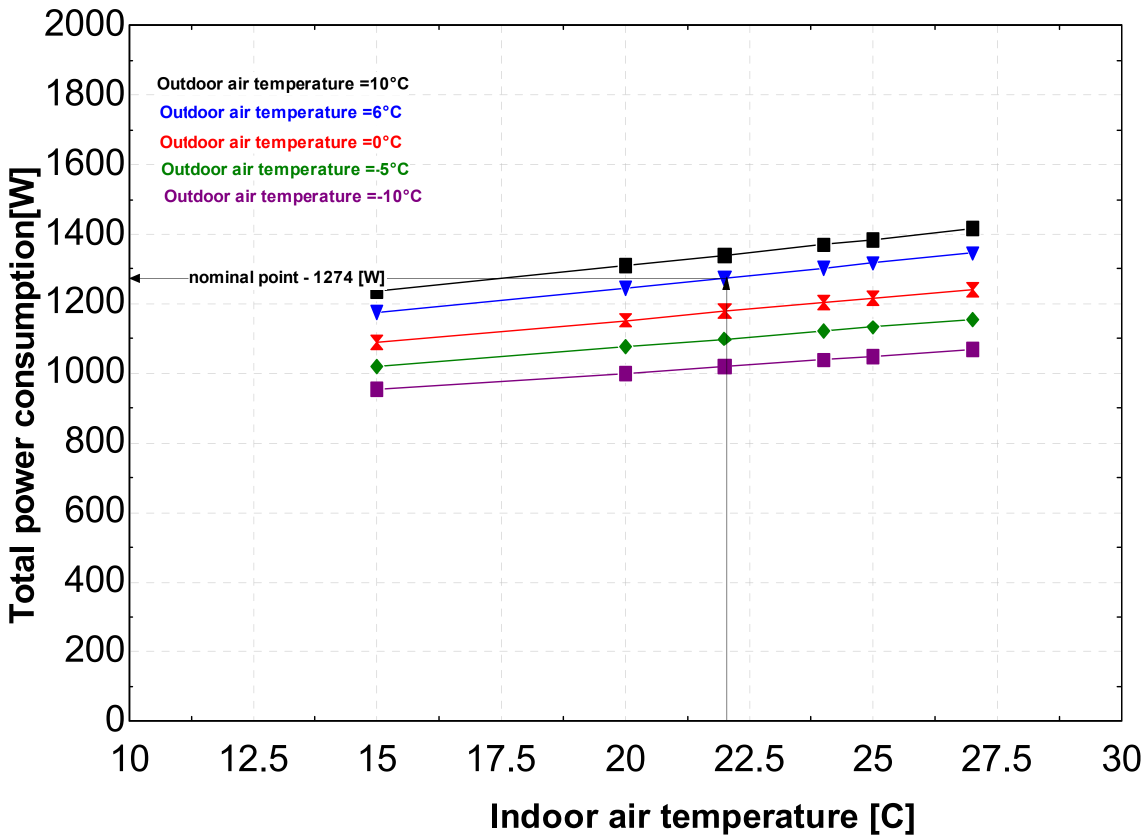

Figure 16: Total power consumption as function of indoor and outdoor air temperatures (medium fan speed; full load)

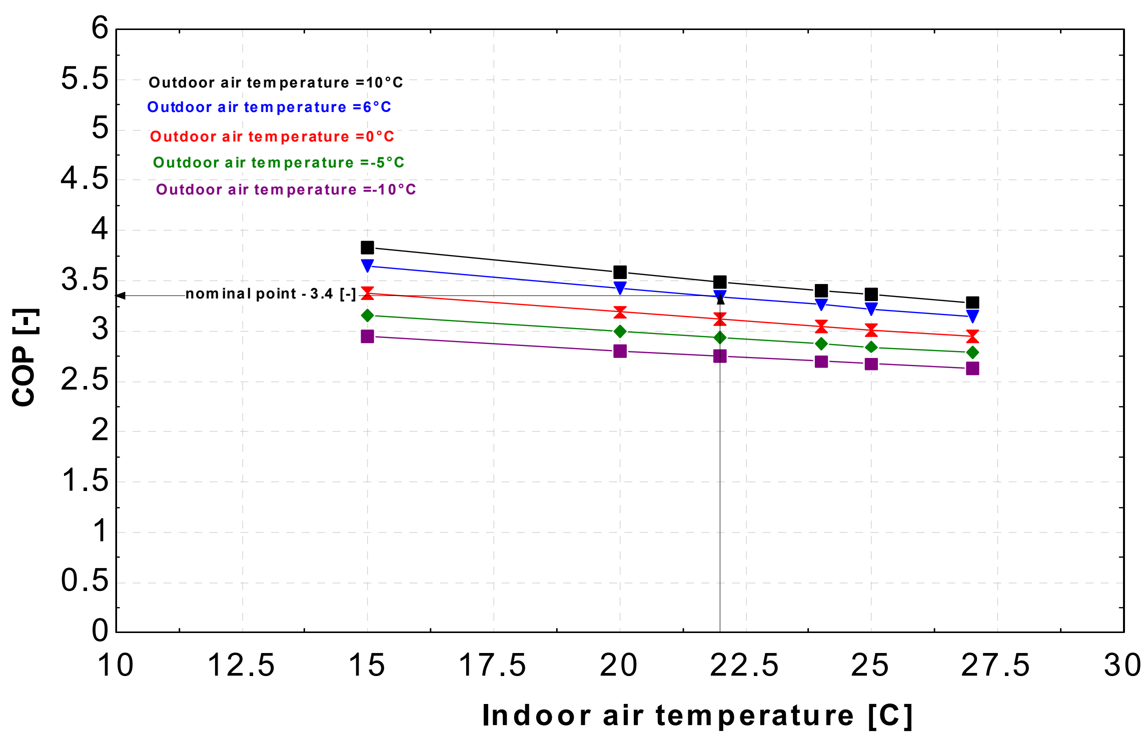

Figure 17: RAC COP as function of indoor and outdoor air temperatures (medium fan speed; full load)

6. References

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