Multiple model approach and experimental validation of a residential air to air heat pump.

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Abstract :
The aim of this work is the realization of a design tool, which is a multiple model software called « CODYRUN », suitable for professionnals and usable by researchers. The original aspect of this software is that the designer has at his disposal a wide panel of choices between different heat transfer models. More precisely, it consists in a multizone software integrating both natural ventilation and moisture tranfers.
Generally, HVAC systems and especially domestic air conditioners, are considered only in either a very simplified way, or in an over elaborated one. The originality of our work is the multimodel approach in the modellisation of air conditionning systems (in this case, we have studied the modellisation of a split-system) and the integration into the calculation code CODYRUN. Three models will be presented, characterized by different complexities of modellisation and different time lapses. The aim is to compare an hourly time lapse model to a reduced time lapse model enabling the consideration of the regulation and the start-up of the system.
An experimental validation was achieved in order to compare the accuracy of each defined model. The data acquisition system and the test cell, located at the University of Reunion island, Indian Ocean are described.
The results found show that short time step dynamic models are more precise energy consumption predictors than hourly time step models.

1. INTRODUCTION

The main problem concerning Demand Side Management in islands such as Reunion Island is that the means of electrical production is restricted. The power supply demand is not great enough to justify a nuclear power plant and thus the means of production is more expensive. The average production cost per kWh is greater than the selling price (the selling price being the same as in mainland France). Under these conditions, the air conditioning of buildings is a growing problem because air conditioning systems consume a large amount of energy and generate investments in the power supply. In addition, the buildings in general are of unadapted thermal design, which generates high cooling loads. The large population increase in the tropical islands, the rise in the living standards and the decreasing cost of air-conditioning units constitutes a real energetic problem.

The modelling and simulation of HVAC systems, and the interaction with the building is also necessary for the evaluation of a means for reducing energy consumptions.

Our studies focus on the modelling of a residential air-to-air split-system heat-pump, which is the most commonly used air conditioning system found in tropical islands. The problem in heat-pump modelling is the definition of the accuracy level of the model. Generally, HVAC systems and especially domestic air conditioners, are only considered in either a very simplified way, or in an elaborated one. The question is : To what degree of complexity can the heat-pump be modelled in view to further

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integration into thermal and airflow simulation software. According to Irving [15], there are an important number of simulation software, all of varying degrees of complexity available to the industry. These programs range from very simple steady-state heating or cooling loads used on programmable calculators, to very detailed simulation programs which require powerful computers. It is obvious that no single program can satisfy the many needs of the engineer at every stage of his design.

Another point to be considered is the time step of the calculation programs. It is often taken to be one hour, which is too long when taking into account the HVAC systems control. In fact their time constants are much shorter than the building time constants. It is apparent that optimal cooling control strategies require a dynamic model for predicting the performance characteristics of the overall system [10]. Lastly, the cooling coil of the indoor unit of the split-system is one of the key components in the air conditioning system. The main difficulties are found in wet and/or transient regimes and the performance of the whole system (and thus the accuracy of the model) depends on a precise model of the cooling coil .

2. CODYRUN AND THE INTEGRATED EQUATIONS FOR HVAC SYSTEMS

2.1 An overview of **CODYRUN** *software :*

CODYRUN is a thermal multizone software integrating both natural ventilation and moisture transfers. Its main characteristic is that it is a multiple model structure, which enables the choice of a wide range of models of heat transfer and meteorological data reconstitution. More information can be obtained about this software in [3] concerning multiple model aspect, building description in [5], thermal model constitution in [4] and preliminary software validation in [11]. This software is developed on PC micro computer and has the advantage of the Microsoft WINDOWS front-end.

When concerning the calculation, the main parts include the airflow model (pressure model covering wind, thermal buoyancy and large openings), the thermal model and the moisture transfer model. For the study presented in this paper, the last two models have been modified by keeping the same multiple model concept, allowing different levels of air conditioning system modellisation.

2.2 Summary of thermal and moisture transfer models:

By considering usual assumptions as mono-dimensional heat conduction in walls, well mixed air volumes, and linearized superficial exchanges, nodal analysis (or lumped capacities analysis) leads to an electrical network. To simplify our discussion, it is supposed that the heat conduction is treated with a model constituted of a thermal resistance and 2 capacitors, called the "*R2C*" model (leading to no internal nodes in walls). The thermal model of the building is obtained by a set of equations 1 to 4, defining the thermal balance of inside (*Tsi*) and outside nodes (*Tse*) according to the boundary conditions, the thermo-convective balance equation of dry-bulb air nodes (*Tai*) and the radiative balance equation of the inside mean radiant temperature nodes (*Trm*).

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$$C_{si}\ \frac{dT_{si}}{dt} = h_{ci}\ (T_{ai} - T_{si}) + h_{ri}\ (T_{rm} - T_{si}) + K(T_{se} - T_{si}) + \varphi_{swi} \qquad (1)$$

$$C_{se}\ \frac{dT_{se}}{dt} = h_{ce}\ (T_{ae} - T_{se}) + h_{re}\ (T_{ae} - T_{se}) + K(T_{si} - T_{se}) + \varphi_{swe} \qquad (2)$$

$$C_{ai}\ \frac{dT_{ai}}{dt} = \sum_{j=1}^{Nw} h_{ci}\ (T_{ai} - T_{si\,(j)} + c\ \dot{Q}(T_{ae} - T_{ai})\ + \dot{Q}_{sens} \qquad (3)$$

$$0 = \sum_{j=1}^{Nw} h_{ri}\ A_j\,(T_{si}(j) - T_{rm}) \qquad (4)$$

The separate calculation of moisture balance (induced by air motion between indoor and outdoor zones, latent storage in walls and room furniture not accounted for*)*, for a zone of specific humidity r_s, leads to a linear system of equations :

$$C\ \frac{dw}{dt} = \dot{m}_{in}\ l_v\ w_{in} - \dot{m}_{out}\ l_v\ w + \dot{Q}_{lat} \qquad (5)$$

$\dot{Q}_{sens}$ and $\dot{Q}_{lat}$ respectively are the sensible and latent capacities interacting in the zone, comprising internal loads (such as lighting, occupants) and HVAC systems.

2.3 Integration of the air cooling system :

At each time step, depending on indoor and outdoor conditions, the HVAC model calculates $\dot{Q}_{sens}$ and $\dot{Q}_{lat}$ for the considered zones, values used in equations *(3)* and *(5)* . The main difficulty when integrating HVAC system models to CODYRUN is that all the models used (except for the pressure airflow models) are linear models. Consequently, in order to reduce calculation time a linear model has to be used.

3. MODELLING REVIEW

When concerning heat-pump simulation, it appears that the different models can be broadly classified as steady and dynamic state simulations. Weslby [25] reviewed various mathematical models of mechanical vapour-compression heat-pumps from 1975 to 1988, with particular emphasis on the bases and end-use.

It appears that detailed steady state models are often used to study variations in system or component configuration in order to identify optimal values of parameters that dictate the performance of the system. They are very helpful to optimize the design of air conditioning equipment. The component approach is often used by reseachers because of the strong coupling between them. [2]. In general, authors have developed a numerical model to determine the steady-state performance of heat-pumps for specific source and sink conditions and specific components.The model is based on simple models of the components of the heat pump. When considering the coupling with a thermal simulation software, Loveday [16] developed a heat-pump simulation program in order to relate the hourly thermal building load to the corresponding electricity input function of the heat pump COP. The hourly heating energy to be supplied to the building is obtained from ESP [9]. The heat-pump model is a steady-state model which relates heat-pump COP to external and ambient temperature, and takes into account motor/compressor inefficiencies, thermal/mechanical losses through heat transport, and defrosting performance.

These steady-state models are generally too detailed for our needs. Other investigators have worked on steady-state models based on experimental data and manufacturers data [1], [12], [24]. With the knowledge of different set points and different experimental conditions, a model based on polynomial laws relating the cooling capacity or the power consumption to the outdoor and indoor temperature can be elaborated.

Transient perturbations and dynamic regimes can be simulated to analyse the start-up process, the defrost-cycle and rapid varying operating conditions. The aim of these models is to study the variation of a working fluid mass and thermodynamic state distributions within the system from known initial conditions, where the system operating conditions are changed. Some reseachers have developed precise dynamic models [8], [19], [17], [23]. These models range from lumped parameters [19], [23] to mathematical models based on a multi-node or distribution approach [17]. These models tend to be complex, require powerful computers, and do not readily yield physical insight into the variables affecting the system performance. However, they do allow the user to find detailed information (temperature, pressure, etc.) in the system during the transient start-up process.

Other investigators [18] have suggested a two-time constant model to monitor the physical phenomena responsible for start-up losses. The first time constant monitors capacity delay in relation to the mass of the heat exchanger. The second time constant monitors a very high, but rapidly decreasing, initial capacity that is produced by the time lag for the refrigerant to be pumped from the evaporator through out the system. This model is very accurate during the start-up process.

Other investigators have hypothized that during start-up, the system capacity could be modelled as a first-order system (single-time constant) [7], [13], [20], [21], [22]. Experimental data from Murphy [20] shows good correlation between a single time constant model and data from a heat-pump. We also verified the hypothesis ourselves and found a time constant of 2 minutes. Murphy found time constants for cooling mode operations ranging from 0.32 to 0.47 minutes for a heat-pump and an air-conditioner respectively. O’Neal [22] found a time constant of more than 2 minutes for a heat-pump operating in cooling mode. The differences in time constants, from 0.4 to 2 minutes may be significant if the fractionnal ontimes are low (close to 10%). In this case, the startup conditions are most important for the evaluation of the energy consumption of the system because it never reaches its nominal state. Another important assumption made by Goldsmith [13] and O'Neal [21] was the constant power at start-up. This assumption appears to be reasonable. Whilst there is an initial surge in power during the few seconds after start-up, the power overall however, is relatively constant during the start-up period.

When considering the coupling effects of cooling and dehumidifying on a cooling coil, several authors have proposed different models of a coil in steady-state and/or transient conditions. Hirsch [14] considers that the moisture condensation on cooling coils is simulated by charactarizing the coils by their bypass factors and solving the bypass relation with the system moisture balance in steady-state conditions. Xin Ding [26] has worked on different models of a cooling coil in steady-state and transient conditions, and in dry or wet regimes.These models are based on the NTU method and on the determination of the efficiency of the coil in wet and dry regimes. The dynamic model of the coil is taken to be a first order model.

4. MODEL DEVELOPMENT

The above shows that complex models are needed for the study of the performance of the system alone. When studying the coupling of the system with the building, the models used are much simpler as a means to reduce the calculation time.This is why the three models used by CODYRUN are linear models, based on manufacturers data. The HVAC system studied is a R22 residential air-to-air split-system heat pump that is commercially available, characteristics shown in table1.

This split-system operates only in a cooling mode, with a nominal cooling capacity of 3,3 kW. Both indoor and outdoor heat exchangers are of tube-and-plate-fin construction. The compressor is a hermetic rotative compressor and the expansion device is a capillary tube.

Nominal values of the split-system *:*

- total cooling capacity : 3.3 kW
- sensible capacity: 2.52 kW
- electric power : 1.25 kW
- COP : 2.6

CARRIER Indoor unit : 42 HWA - Outdoor unit : 38 SDF012
Indoor dry bulb temperature : 27°C

		By Pass factor = 0.04		Air flow rate : 110 l/s		
	Wet bulb temp (°C)	16	18	19	20	22
Outdoor ttemp (°C)	Indoor air specific enthalpy (kJ/kg)	45	55	58	61.2	65
21	Total capacity	3.46	3.65	3.72	3.78	3.91
	Sensible capacity	3.26	2.88	2.67	2.45	2.04
	Power (kW)	0.99	1.00	1.00	1.01	1.02
25	Total capacity	3.34	3.54	3.64	3.72	3.84
	Sensible capacity	3.20	2.84	2.65	2.44	2.03
	Power (kW)	1.06	1.07	1.07	1.08	1.09
30	Total capacity	3.16	3.37	3.48	3.59	3.79
	Sensible capacity	3.10	2.78	2.59	2.41	2.03
	Power (kW)	1.14	1.15	1.16	1.17	1.19
35	Total capacity	2.95	3.18	***3.30***	3.40	3.63
	Sensible capacity	2.95	2.70	***2.52***	2.34	1.98
	Power (kW)kW	1.23	1.25	***1.25***	1.27	1.28
40	Total capacity	2.67	2.77	2.95	3.18	3.44
	Sensible capacity	2.67	2.52	2.38	2.25	1.91
	Power (kW)	1.33	1.34	1.35	1.36	1.38
45	Total capacity	2.49	2.56	2.69	2.93	3.3
	Sensible capacity	2.49	2.43	2.28	2.16	1.86
	Power (kW)	1.40	1.40	1.41	1.42	1.45

Table 1 : Manufacturers data for a residential air-to-air split-system heat pump- Source : CARRIER Corp. Professionnal catalogue for residential units, 1995-1996. (Nominal values : dark italics)

4.1 *Model n°0*

This model is based on the assumption that the supply of the cooling load is obtained by an ideal control loop (no delay between the sollicitations and the time response of the system). The available outputs are considered to be the hourly cooling consumptions which do not integrate the real characteristics of the HVAC system. We assume that sensible and latent cooling rates are independant factors. The real physical phenomena is different in that cooling and dehumidifying occur at the same time.

The HVAC system must reach a set temperature and a set humidity. It allows the user to determine the hourly sensible and cooling load of the building, but does not take into account the characteristics of the system.
In order to estimate the consumption of electricity, the system is modelled by the cooling Coefficient of Performance (COP), which is the ratio between the cooling load and the electric power in nominal conditions. The COP is taken to be constant and in our case is 2.6.

The COP is also an initial comparison quality criteria for heat pump systems . It enables the estimation of the electrical consumption of the system during the simulation sequence.

This type of model is generally used by engineers to calculate the dimensions of the air conditioning appliances and to estimate the exploitation costs of these installations.

4.2 *Model n °1:*

This model aims to cover the regulation and start-up parameters of the system. For this purpose**,** the time step was reduced to one minute in order to take into account the on-off cycling and the control of the system.

Steady -state conditions :

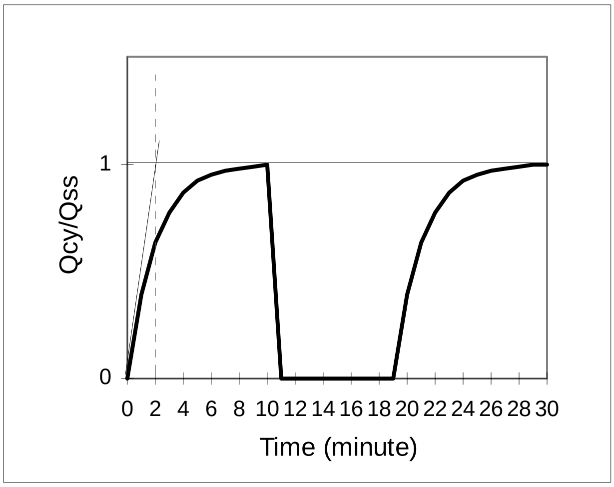
The capacity rate is the nominal capacity of the system operating in steady state and standard conditions. (27°C dry bulb temperature, 19°C wet bulb temperature, 35°C outdoor temperature). The system is characterised in the steady state conditions by the *SHF* (sensible heat factor) which is the ratio between the sensible cooling rate and the total cooling rate. The *SHF* enables the determination of the sensible and latent capacities which will be implemented at each time step in the solving equations of CODYRUN.

In our case, (see Table 1), $\dot{Q}tot = 3.3$ kW
$SHF = 0.76$

Dynamic conditions :

As seen in the modelling review above, the system capacity is modelled as a first order system, in order to account for the start-up of the split-system [21], [22].
Figure 1 shows the ideal capacity during start-up and shutdown of a residential air-conditioner.

The instantaneous cooling capacity, $\dot{Q}cyc$ is given by :



$$\dot{Q}cyc = \dot{Q}ss(1 - e^{-t/\tau}) \quad (6)$$

where

$\dot{Q}ss$: steady state capacity in nominal conditions.

τ : time constant for the system (experimental data)

t : time after start-up

Fig 1 : Dynamic behaviour during start-up conditions.

The time constant for the system was determined experimentally in our test cell (see experimental set up section below). The measurement of the start-up period of the split system enabled the determination of a time constant of 2 minutes.

System control :

The on/off controller with dead zone is the most commonly used control system in small size residential units. The experiment showed a controller dead zone of ± 0.5°C from the set temperature.

Electric power :

The electric power of the system is taken to be constant, even under start-up conditions [13], [21], [22].

In our case, the value in nominal conditions was taken. P_e = 1.25 kW

4.3 Model n°2 :

The performances of an air-to-air heat-pump (total cooling capacity, sensible and latent capacities, electric power) are strongly influenced by parameters such as the indoor dry bulb air temperature, the indoor relative humidity and the outdoor air temperature. In model 1, the performances of the system are taken to be constant, but figure 2 shows that they are dependent on these external inputs. The aim of this model is to take into account these parameters in steady-state conditions.

Steady-state conditions :

The method is based upon manufacturers data shown in Table 1. This data was measured under steady-state conditions, where the indoor dry-bulb temperature was kept constant. In addition to this data the indoor air specific enthalpy was considered. Table 1 enables the plotting of the total and sensible cooling capacities function of the indoor air specific enthalpy (see figure 2).

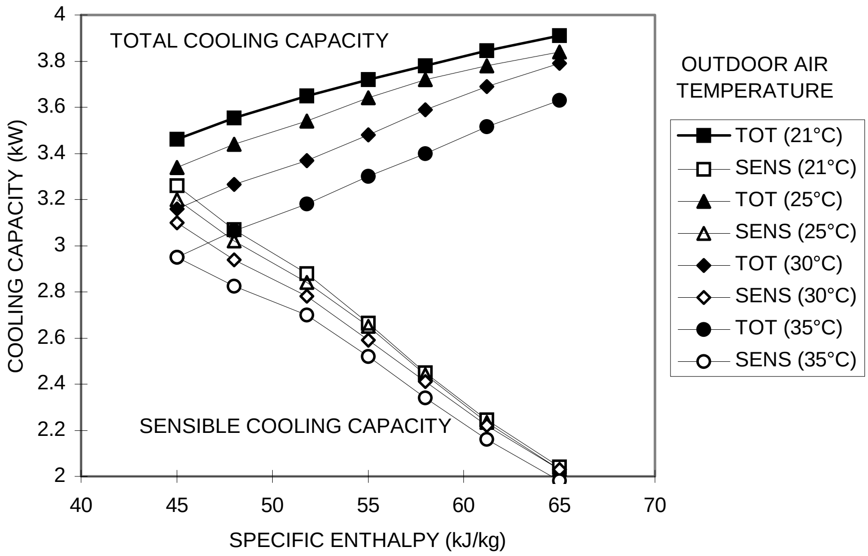


Fig 2 : Evolution of total cooling capacity and sensible cooling capacity function of the indoor air specific enthalpy, for different outdoor air temperatures. Indoor air temperature = 27°C

As these experimental total and sensible capacity curves are nearly linear, it can be assumed that, under a fixed outdoor temperature, the sensible and total capacities follow linear laws. These equations, determined from manufacturers data are written as follows:

$$\dot{Q}tot(27°C) = a_0 + a_1 \cdot h_{ent} \qquad (7)$$

$$\dot{Q}sens(27°C) = b_0 + b_1 \cdot h_{ent} \qquad (8)$$

where : h_{ent} is the entering air specific enthalpy at the evaporator.
a_0, a_1, b_0, b_1 are the correlation coefficients of $\dot{Q}tot$ and $\dot{Q}sens$ respectively.

a_0, a_1, b_0, b_1 depend on the outdoor air temperature. We determined the values of these coefficients for the different outdoor temperatures of 21°C, 25°C, 30°C and 35°C. The values of 40°C and 45°C were not taken into account as on the one hand this is outside the operating range of the split-system and on the other, the evolution of the capacities tends to become non linear.

When plotting the a_0, a_1, b_0, b_1 coefficients as a function of exterior temperature, it is seen that they also follow linear laws. Thus, they are given by :

$$a_1 = c_3 + c_4 \,.\, Text \qquad (10)$$
$$b_0 = c_5 + c_6 \,.\, Text \qquad (11)$$
$$b_1 = c_7 + c_8 \,.\, Text \qquad (12)$$

where Text is the outdoor air temperature.

c_1, c_2, c_3, c_4, c_5, c_6, c_7, c_8 are the calculated correlation coefficients from the linear equations (9), (10), (11), (12).

Through these correlations, the total capacity and the sensible capacity are given by the following equations for an indoor air temperature of 27°C :

$$\dot{Q}tot(27°C) = c_1 + c_2 \,.\, Text + (c_3 + c_4 \,.\, Text).h_{ent} \qquad (13)$$
$$\dot{Q}sens(27°C) = c_5 + c_6 \,.\, Text + (c_7 + c_8 \,.\, Text).h_{ent} \qquad (14)$$

The manufacturers data enables the estimation of the total and sensible cooling capacities as a function of the inside specific enthalpy and the outdoor air temperature for an inside set temperature of 27°C. To determine the cooling capacities at any operating point, the coil bypass factor concept [6], [14] can be introduced.

The coil bypass factor (BF) model characterizes the air exiting the coil as a mixture of two major streams : the air which isuninfluenced by the coil and the air which leaves at coil surface temperature (apparatus dew point). The coil bypass factor is the fraction of air which is unaffected by the coil, given by :

$$BF = \frac{T_{out} - T_{adp}}{T_{ent} - T_{adp}} = \frac{h_{out} - h_{adp}}{h_{ent} - h_{adp}} = \frac{w_{out} - w_{adp}}{w_{ent} - w_{adp}} \qquad (15)$$

with T_{adp} : apparatus dew point temperature (temperature of the coil surface)
h_{adp} : specific enthalpy at the coil surface.

The coil bypass factor is a function of both physical and operational parameters of the coil : the coil exchange surface *A* (when *A* is increasing, *BF* is decreasing), and the coil air velocity *v* (when *v* is increasing, *BF* is increasing). In our model it is assumed that these parameters are constant (air fan at top speed), so the bypass factor is also considered to be constant.

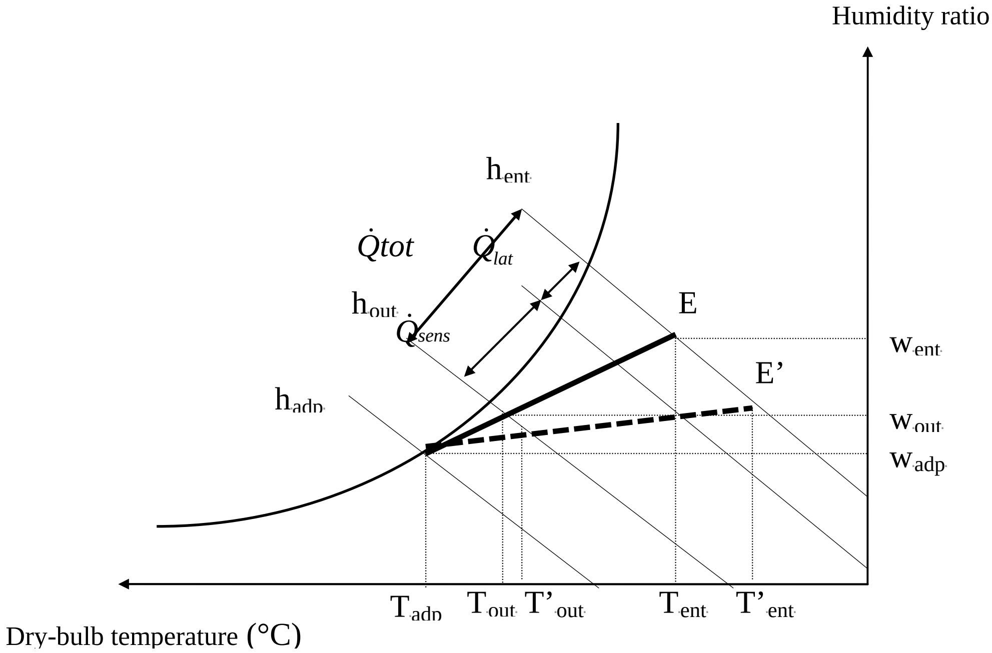


Fig 3 : Cooling coil performance and determination of cooling capacities

Figure 3 shows an illustration of the problem. One important assumption was made when developing this model : the total cooling capacity rate is considered to be constant for two operating points E and E' which have the same entering specific enthalpy which gives :

$$\dot{Q}tot(E) = \dot{Q}tot(E') = \dot{m}.(h_{out} - h_{ent}) \qquad (16)$$

where $\dot{m}$ is the mass flow rate of the air through the coil (kg.s^{-1})

The apparatus dew point is in consequence the same for the two entering points :

$$h_{adp}(E)=h_{adp}(E')$$

The sensible and latent capacities change between the two points, but the total capacity remains constant. Considering the point E', the equations (13) and (14) give the value for the same specific enthalpy of the total capacity and the sensible capacity for a drybulb temperature of 27°C.

Thus :

$$\dot{Q}sens = \dot{m} \cdot (h_{out} - h_B) = \dot{m} \cdot c_{pm} \cdot (T_{out} - T_E) = \dot{m} \cdot c_{pm}(1 - BF)(T_{adp} - T_E)$$

$$\dot{Q}sens' = \dot{m} \cdot (h'_{out} - h_{E'}) = \dot{m} \cdot c_{pm} \cdot (T'_{out} - T_{E'}) = \dot{m} \cdot c_{pm}(1 - BF)(T_{adp} - T_{E'})$$

$$\dot{Q}sens' = \dot{m} \cdot c_{pm}(1 - BF)(T_{adp} - T_E) + \dot{m} \cdot c_{pm}(1 - BF)(T_E - T_{E'})$$

then $$Qsens' = \dot{Q}sens + \frac{\dot{V}}{v} c_{pm}(1 - BF)(27 - T_{E'}) \qquad (17)$$

where $\dot{Q}sens'$ is the sensible capacity of the heat-pump for the E' entering point.

$\dot{Q}sens$ is the sensible capacity of the heat-pump for the E entering point (T_E = 27°C) given by the expression (14).

The latent loads can be determined with the equations (16) and (17):

$$\dot{Q}tot(E) = \dot{Q}tot(E')$$

$$\dot{Q}sens' = \dot{Q}sens + \frac{\dot{V}}{v} c_{pm}(1 - BF)(27 - T_{E'})$$

$$\dot{Q}lat' = \dot{Q}tot - \dot{Q}sens' \qquad (18)$$

Then, the values of $\dot{Q}sens'$ and $\dot{Q}lat'$ are integrated for each time step in the equations (3) and (5).

Dynamic state conditions :

The dynamic effects upon start-up are the same as in model n°1 (single time constant).

System control :

The system control is an on/off controller with a dead zone of $\pm$ 0.5°C from the set temperature.

Electric power :

Table 1 indicates that the electric power is essentially dependant on the outside temperature, the inside conditions have very little influence. The electric power as a function of the exterior temperature as shown in fig.4 appears to be linear. It can therefore be assumed that the electric power is a linear function of the outdoor temperature.

It is also assumed that there is no dynamic effect during the few time steps after start-up.

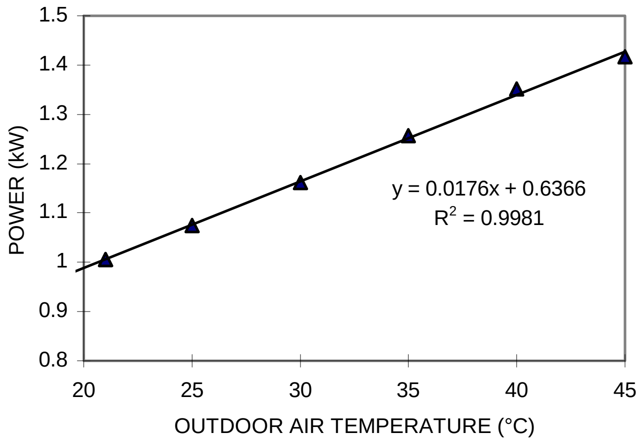


Fig 4 : Electric power of the split-system

5. EXPERIMENTAL SET-UP AND TEST PROCEDURE

A single-speed air-to air heat pump was installed and tested in a test cell located at the University of Reunion Island. The experiments were monitored under natural climatic conditions.

5.1 Test split-system :

The characteristics of the installed unit are shown in table 1. Therefore the manufacturers data, from which the models where established (time constant, dead zone) is the same for the experimental system.

Monitoring included, the temperature of the refrigerant fluid from both sides of each component (indoor and outdoor units, compressor and expansion device), the R22 volume flow rate by means of a litre-meter, high and low pressures, power and energy supply (active, apparant and reactive) by means of an electronic watt-meter. Also the inlet and outlet temperatures and the mass flow rate of the water from the indoor unit were measured.

Thus we can determine the total cooling capacity through the thermodynamic cycle of the R22, and both sensible and latent capacities with the measures of the inlet and outlet temperatures of the indoor unit, and the measured mass flow rate of the water.

5.2 Test cell

The size of the test-cell is 3.0 x 3.0 x 2.30m (see fig. 5).
The test cell is made of sandwich panels composed by two 7 mm layers of ciment-fibre boards with 6 cm of polyurethane foam between them. There is a 5 cm layer of styrofoam between the floor panel and the concrete. On the roof, there is an extra sandwich panel, made of aluminium sheets and polyurethane, approximatly 5 cm thick. The inner floor is made of 5cm concrete paving stones .

Fig 5 : Sight of the Test cell and the outdoor unit of the split-system.

Indoor air temperatures are mesured by type T (copper-constantan) thermocouples shielded against radiation, at three different levels (0.30m, 1.20m, 1.80m). External and internal temperatures are measured with type T thermocouple. The sensors are glued to the walls and painted the same colour.

The indoor relative humidity is also recorded by to a temperature and relative humidity probe.

5.3 Weather data acquisition :

A weather station situated near to the test cell (see fig.5)was used to measure the outdoor air temperature, the outdoor relative humidity, the wind speed and the direction and the global and diffuse horizontal solar radiations. This data, which constituted a meteorological was used to compare measurements and modellisation results.

5.4 Data acquisition system and computer processing.

All the data is measured and collected by dataloggers that are controlled and programmed by a PC computer via RS232 links. The time step of the data acquisition can be modified. For instance, a time step of one hour was used for the validation of the test cell without a HVAC system, but the time step was reduced to one minute when experimenting this split system.

5.5 Experimental procedure

The experiments were conducted at different stages. Firstly a comparison between the simulation and the measurements on the test cell without air-conditioning system was carried out in order to optimize the modellisation of the cell. Figure 6 shows a passive period of two weeks. The measured and simulated air temperatures are similar, with deviations smaller than 1°C.

Secondly, experiments were carried out in order to attain accurate information about the behaviour of the split-system unit (determination of the time constant for the model n°1 and 2, values of the dead zone band of the controller, etc...). Through these preliminary tests a time constant of 2 minutes and a dead zone band whithin a range of $\pm$ 0.5°C from the set temperature was determined.

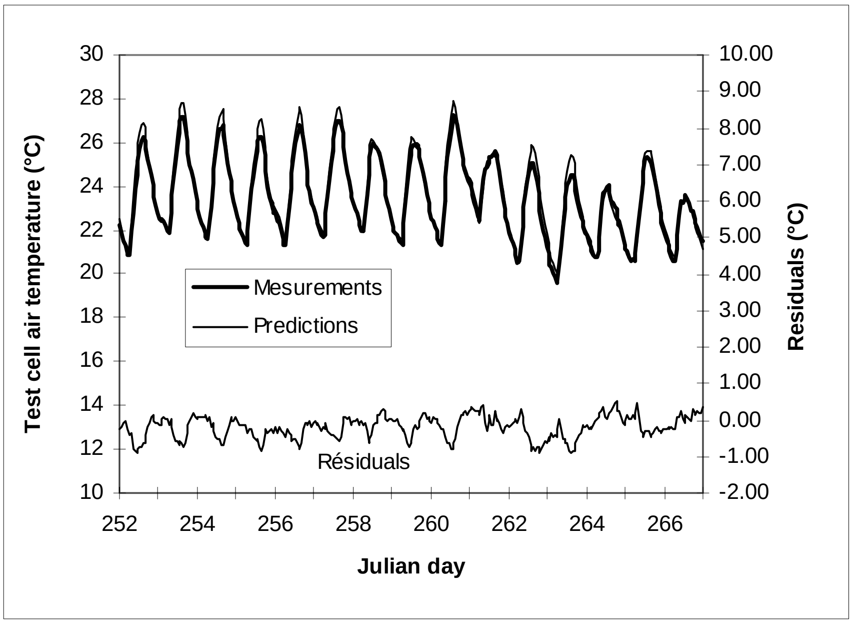


Fig 6 : Comparison between measurements and predictions - Passive period

Thirdly, experiments were carried out with the split-system in the operating mode for different indoor drybulb set temperatures. The time step of the experiments was taken to be 1 minute. The test periods generally lasted for seven days . The first results are detailed below.

6. RESULTS

The internal setpoint was 23°C for the whole period of the measurements. The air renewal was ignored. There were no internal loads inside the test cell.

The main problem arising from these simulations is that the air non-uniform mixing is not accounted for. CODYRUN assumes that the air volume in the cell is homogeneous. Therefore all the convective cooling capacity is transmitted directly to the air node in the model. The experiment shows that there is a delay between the transmission of the cooling capacity and the decrease in the room temperature. To compensate this phenomena, the air volume of the cell was increased in the model. The best results were obtained for a model volume, five times bigger than the actual volume of the cell. The results presented below take into account this correction.

Figures 7 to 10 present measured data and simulation results of models 1 and 2 for a time step of one minute. The presented outputs in figures 7, 8, 9,10 were the indoor air temperature, the electric power, the sensible cooling capacity and the total cooling capacity respectively. The selected day was chosen from a test period of seven days. (Oct. 20-27, 1996) This day had the greatest solar radiation.

The model n°0 requires a time step of one hour. In order to compare the model n°0 to the simulations of models n°1 and 2 and to the measurements, results for a time step of one minute were averaged over one hour. Thus, Figures 11 to 13 present the measured and simulated results for a time step of one hour. The presented outputs are the average indoor air temperature, the total cooling capacity and the electric power.

Visual inspection and comparison of the various graphs gives the following observations.

Short time step (one minute):

The comparison of the measured and simulated indoor air temperatures show great similarity. The time of the first start-up time of the split system is earlier in the models (approximatively around 7h30 a.m) than in the experiment (around 8h00). This is due to a 0.5°C gap between the model and the measured temperatures during the passive period. Yet, the number of on/off cycles, the daily variations of the indoor air temperature and the last shut down (around 17h30) are well modellised.

The same trends can be seen when considering the predicted power demand. Model n°2 appears to be more accurate than model n°1.

The predictions for the sensible cooling capacity (see figure 9) show correct general tendancies when compared to the measured values. On the contrary, there is a large difference between the measured values and the two models for the total cooling capacity (see figure 10). This difference is due to the fact that the time delay of one on-off cycle is shorter than the measured one. The modelled capacity does not have enough time to reach the steady-state value. Also due to the non homogenity of the air and the low fractional on-time, it is very difficult to modellise correctly the cooling capacity, A higher fractional on-time would enable the system to reach a permanent regime, and thus probably give a more precise model of the cooling capacity provided by the system.

	Number of cycles	***On time (min)***	***Total time (min)***	***Fractional on time***	***Daily n° of cycles***
Measurements	10	26	163	16%	30
Model n°1	10	30	149	20%	33
Model n°2	10	30	149	20%	32

Table 2 : Results of the models with reduced time laps

The fractionnal on time was calculated for each model by determinating the ontime period and the total time for 10 cycles (seeTable 2).

Table 2 shows that the number of cycles in one day is correctly simulated for the two models. The fractional on-time is very low. The system never reaches the steady state regime.

One hour time step :

Indoor air temperature predictions exibit the same trends as the air temperature. This is quite obvious as these values are the average of the one-minute-time-step values. The maximum gap is around 0.5°C.

The results given by figures 12 and 13 are interesting. The predictions of total cooling capacities for models 1 and 2 are larger than the ones obtained for the model 0, but remain below the experimental results. These discrepancies can be attributed to the simulation of the controller which is taken into consideration in models n°1 and 2 (and supposed to be ideal in model n°0).

The cooling capacity remains under estimated (see fig. 12), but as seen above the system never reaches a permanent regime and the modellisation of a dynamic regime is difficult due to the air volume mixing conditions which are not taken into consideration in the model.

The same trends are shown in figure 13 for the predictions of electric power demand. Model 2 appears to be the more accurate, with an accuracy of 5%. On the contrary, model n°0 underestimates considerably the power demand (-60%). We can also see the limits of hourly simulations when predicting system consumptions such as heat-pump systems with specific control strategies, and low fractional ontimes.

	Daily Energy Consumption (kWh)	***Difference vs measurements***	Mean COP
Measurements	1.68		1.6
Model n°0	0.68	- 60%	2.6
Model n°1	2.09	+ 25%	1.22
Model n°2	1.76	+ 5%	1.48

Table 3 : Results for an hour time lap

The average COP (over a day) is also low in relation to the COP under nominal regime (2,6). The influence of fractional ontime on the effective COP of the system is therefore of prime importance. The model n°2 gives is very precise regarding the energy consumption and the average COP value, compared to the experimental COP.

7. CONCLUSION

Three modelling levels of air-to-air residential heat-pumps have been defined and integrated in a thermal building simulation software. The first model is to supply the demand of hourly cooling loads and electric demand with an ideal control loop (no delay between the sollicitations and the time response of the system) . The second model takes into account the on/off cycle and the control processing. The third model is based on a linear model determined from the manufacturers data. Dynamic effects are taken into consideration in the last two models through a single time constant model. The time step is reduced to one minute. The ouputs are the total, sensible and latent cooling capacities at each time step. These models are linear, is is the existing code and therefore do not increase the calculation time.

The first comparisons between the measurements and simulations show that a dynamic simulation with shorter time steps than one hour give better results for the estimation of the electric consumption.

The system used for the experimental measurements is in fact oversized regarding the size of the cell. Therefore the system never reaches a permanent regime and has a low fractional ontime (16 %). Thus, the reduced time lap models give the best results as the start-up regime is the over-riding factor in relation to the permanent state. Model n°2 gives the best results regarding the energy consumption and the average COP over a day. In contrast, the hourly model under estimates the daily energy consumption by 60 %.

Netherveless, improvments have to be made concerning the estimation of the total cooling capacity. We have to face the physical problem such as non homogeneity of the air inside the test cell and air infiltrations, what implies a time-gap between measures and models. Future research will focus on the introduction of a time delay due to these physical phenomenon.

Comparisons between the different models and experimentations over higher fractional ontime systems will be carried out. These simulations should reach better results for the model n°0 when the system reaches its permanent state and enable the estimation of the real COP as a function of the fractional ontime.

The interest of the multimodel aspect in our calculation code is that the short time lap models could be used in a first simulation to estimate the COP as a function of the fractional ontime. In a second phase, a correction factor could be applied to the hourly model in order to determine the cooling capacity needs and the electricy consumption of a building during a loçng period. This will allow a better accuracy of the hourly model without sacrifying the calculation time.

8. NOMENCLATURE

A	area	***Subscripts***	
BF	by-pass factor		
C	thermal capacity	adp	apparatus dew point
c	specific heat	ae	air outside
COP	coeficient of performance	ai	air inside
c_{pm}	specific heat of mixing of dry air	ce	outside convection
	and vapour	ci	inside convection
φ	radiation flux density	cyc	cycle
h	specific enthalpy	ent	entering
	heat transfer coefficient	in	inside or inlet
K	thermal conductivity	lat	latent
l_v	latent heat	out	outside or outlet
$\dot{m}$	mass flow rate	re	outside radiation
P_e	electric power	ri	inside radiation
$\dot{Q}$	cooling capacity	rm	inside radiant mean
w	specific humidity	se	surface outside
SHF	sensible heat factor	sens	sensible
T	temperature	si	surface inside
t	time	ss	steady state

τ	time constant	swe	short wave outside
$\dot{V}$	air flow rate	swi	short wave inside
v	specific volume	tot	total

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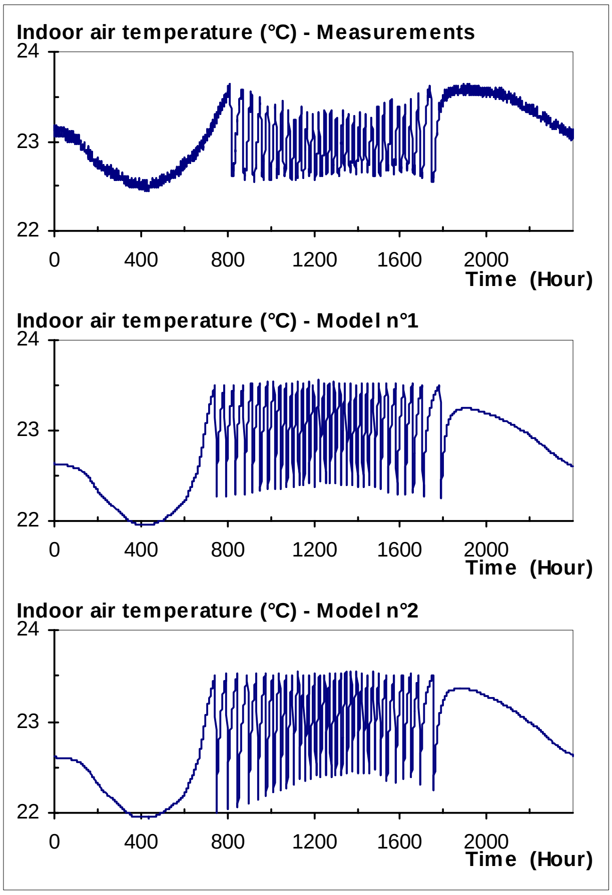

Fig 7 : Indoor air temperature - time step =1 min

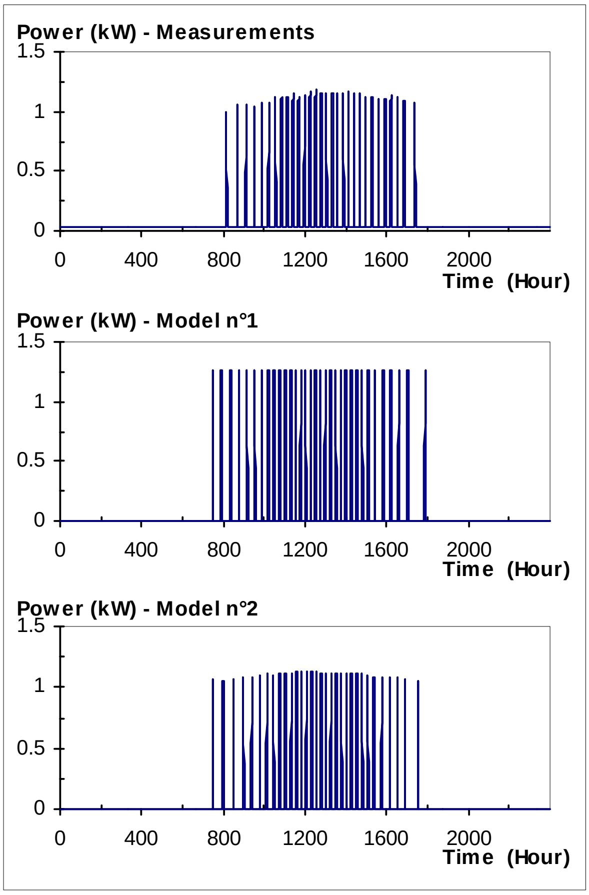

Fig 8 : Electric power - time step = 1 min

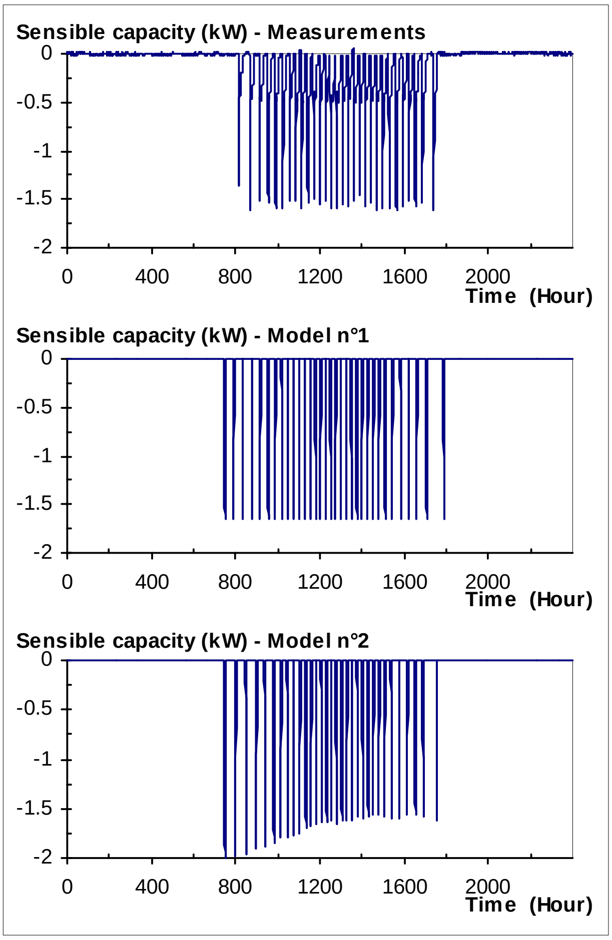

Fig. 9 : Sensible capacity - time step = 1 min

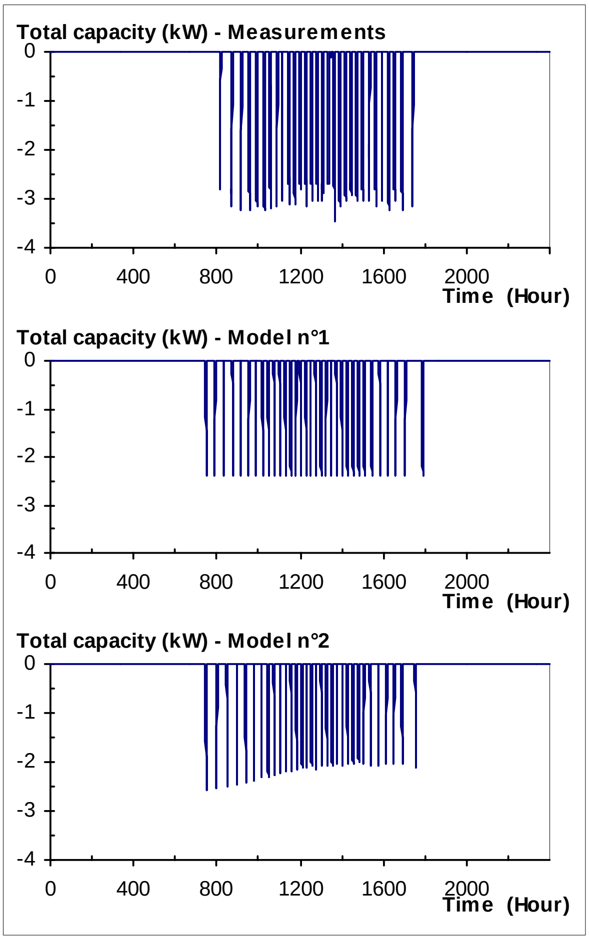

Fig. 10 : Total cooling capacity - time step = 1 min

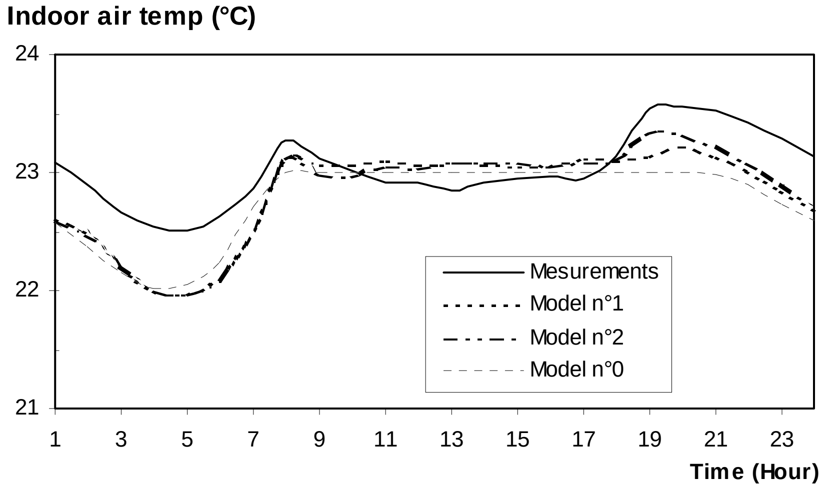

Fig 11 : Indoor air temperature - Comparison of measurements and simulations - time step = 1 hour

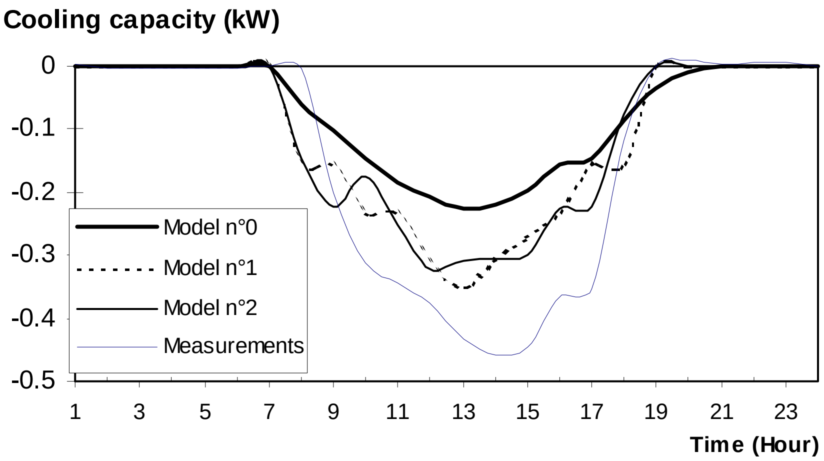

Fig 12 : Total cooling capacity - Comparison of measurements and simulations - time step = 1 hour

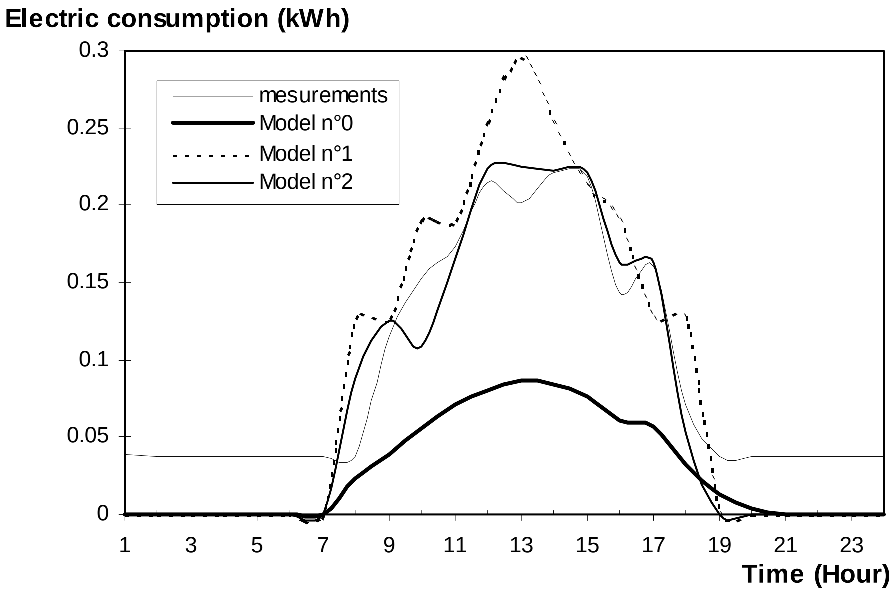

Fig 13 : Electric power - Comparison of measurements and simulations - time step = 1 hour