ETSII-UMA

COMPARATIVE STUDY OF ALL AIR AND NON-ZONING SYSTEMS

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WORKING VERSION

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FIRST CHAPTER
Introduction

1.1 Background and Objectives

Energy consumption in buildings is increasingly important. A fair part of annual consumption in electrical energy is due to the popularization of air-conditioning systems. It is estimated that 40% of end energy consumption occurs in buildings and that approximately 50% is attributable to cooling and heating systems. Solid proof of the interest in reducing electrical consumption in buildings is the progress made in the standards currently underway. The European Directive 2002/91/EC on Energy Efficiency for Buildings establishes, among its requirements, the Energy Certificate for Buildings. The procedure for the certificate appears in the Directive. Any of the following alternatives may be taken:

a) Use of a reference computer programme

b) Use of an alternative computer programme that gives results equivalent to those of the reference programme.

This research work uses the second alternative.

In the residential sector and the small and medium services sector, all-air systems are often used with direct expansion machines and a constant flow duct network. This is the basic system. Logically, a centralized system of this nature, with only one thermostat in the area considered to be the most representative, is unable to meet the demand of all the areas in variable conditions. There will be areas, which, even though they have a well designed duct network and having selected the maximum power for the machine, are undercooled in summer or overheated in winter. In other words, the very design of the system prevents Zoning and, as a result, the system consumes more energy than is necessary and fails to meet the conditions of comfort required by its occupants.

When a constant flow system needs to adapt to demands in different zones, with no variation in drive temperature, the most immediate way to do so are the so-called false variable flow systems. With this type of system, the ventilator is of the constant flow type, but the volume reaching the zones is variable, depending on the load. Zoning is applied based on motorized dampers in the respective zones (all-nothing or authentic variable flow boxes), keeping the temperature required in each zone. The excess flow, which has driven the ventilator and which is not required by the zone, is once again by-passed to the battery. In this manner, there is no reduction in energy consumption in the ventilator, as is characteristic of the variable flow systems, but the simultaneity factor in the load is made use of, making it possible to use a lower powered machine.

The advantages of these systems are:

- Lower powered machine required
- Lower load combated
- Greater comfort

The disadvantages are:

- More noise
- Possible problems with air distribution
- Machine has worse COP. By operating with more unfavourable temperatures in the interior unit and combating less load, nominal performance changes in terms of the fraction of time it is operating at and depending on the conditions of incoming air (temperature and moisture) in the direct expansion battery.

There is currently no machine available for calculating the saving that a false variable flow system can give as opposed to a conventional one. The work proposed is an advancement as it involves a multi-zone system with a detailed formulation of the heat transfer methods in buildings. Numerous load/timetable calculation programmes exist that are based on maintaining the temperature in the different rooms at the set values (by the very definition of acclimatization thermal load). Nevertheless, in order to evaluate the false variable flow system, as it does not combat the load in hours when unoccupied, the calculation of loads is invalidated. It is essential to approach thermal balance in the room in conditions of free temperature evolution. Furthermore, the profile condition for an acclimatized area in contact with another whose damper is inactive is that of free temperature evolution and, given the high overall transfer coefficient and the surface area on which it acts, the load in the acclimatized area is altered in terms of the basic calculation in which all the areas are acclimatized. This fact determines the load on the direct expansion battery and, what is equally as important, the conditions of the surroundings for the occupied areas. In the work developed, the system is coupled to the balance method in order to obtain a representative simulation of the real operating conditions.

The company AIRZONE, manufacturer of Zoning systems, proposes evaluating the suitability of a system of this type, from the viewpoint of comfort for the occupiers and for the thermal and electrical consumption of each system. In order to carry out this comparison, the following cases are considered:

- Non-Zoning system and all/nothing machine
- Non-Zoning system and inverter machine
- Zoning system with no power reduction and all/nothing machine
- Zoning system with no power reduction and inverter machine
- Zoning system with power reduction and all/nothing machine
- Zoning system with power reduction and inverter machine

And the following towns and cities:

- Madrid
- Barcelona
- Malaga
- Valencia
- Seville

1.2 Methodology and structure of the report

Experimental attainment of the comparative study proposed is costly and complicated, due to the need to compare different acclimatization systems subject to the same operating (meteorological and usage) conditions over a long period of time, over a year as a minimum, and in different places.

Thermal simulation of buildings offers an economic, reliable alternative thanks to the validation studies developed in this field. Consultation has involved using one of the most advanced simulation programmes on the market, TRNSYS. On this calculation platform, mathematical models have been implemented for both acclimatization systems. These models incorporate up to date concepts in simulation, such as the simultaneous resolution of the building and its acclimatization system.

The second chapter is an introduction to the problem of thermal simulation in acclimatization systems in buildings. An important feature of this problem is the co-existence of physical processes alongside very different time constants. This is, in turn, a source of numerical complications, when dealing with the problem in its physical literality, and a starting point to formulate simplified models, located in a particular time scale describe as stationary the dynamics that lie below. Based on this approach, the third chapter details the models adopted for both acclimatization systems, non-Zoning and Zoning, as well as some models for calculating the electrical consumption in the production equipment depending on if it incorporates an all/nothing machine or an inverter.

This problem is solved in the following chapters.

§4 describes the case under study: constructional characteristics of the house and calculation conditions (meteorology, usage and conditions of the surroundings).

§5 is concerned with sizing the production equipment (heat pump) and nominal flows to the areas for the non-Zoning acclimatization systems, Zoning with no energy reduction and Zoning with energy reduction. In the case of this latter system, a smaller sized machine is installed that takes advantage of the foreseeable lack of simultaneity in the demands in the respective areas.

§6 presents the results of the simulations. In the first part of this chapter, a description is given of the general behaviour of the models and some interesting effects are commented. The second part describes the characteristics of the acclimatization systems as far as thermal comfort and electrical consumption are concerned.

SECOND CHAPTER
Thermal simulation in buildings

2.1 Introduction

The thermohygrometric state of the air contained in a receptacle is affected by heat flows and mass that tend to modify it and that originate in internal gains, imposed heat flows (e.g., solar radiation) and flows arising spontaneously driven by differences in temperature, pressure and concentration. To maintain pre-set interior (dry and/or radiating) temperature conditions, air quality and, occasionally, relative moisture, HVAC equipment and their controllers are fitted, under instructions from the user of the installation, to meet their comfort requirements. Several strongly linked dynamics can be identified in this problem: structure of the building, conditioning systems (HVAC), measuring and control systems, occupants and environmental conditions. The conjunction of all these determines the interior and operational conditions of the equipment (Figure 2-1).

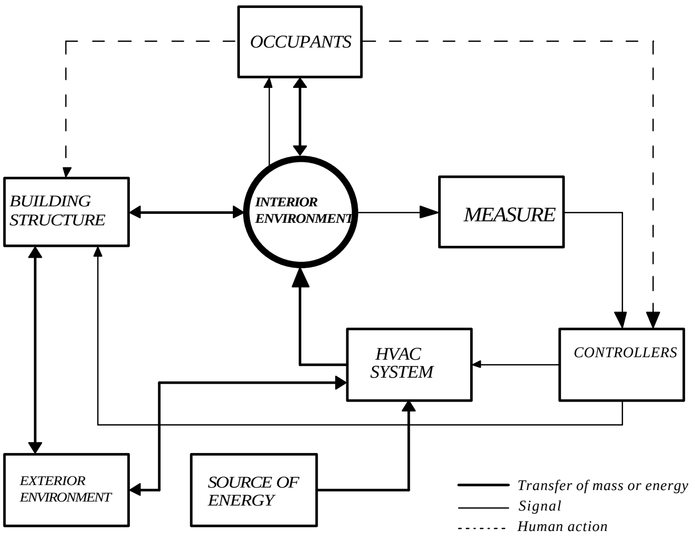


Figure 2-1: Sub-systems and their interactions (adapted from Hensen).

The physical processes involved cover a very wide range of time scales, as shown in Figure 2-2 for typical cases. The fastest phenomenon is the spread of radiation (solar gains, long wave exchanges) at the speed of light, with characteristics times being in the order of microseconds. This is followed by the spread of pressure waves, linked to the movement of fluids (air driven by a network of ducts, infiltrations and movements of air between the areas in the building, flow of coolant inside an evaporator or condenser, etc.), at the speed of sound and with characteristic times in the order of milliseconds.

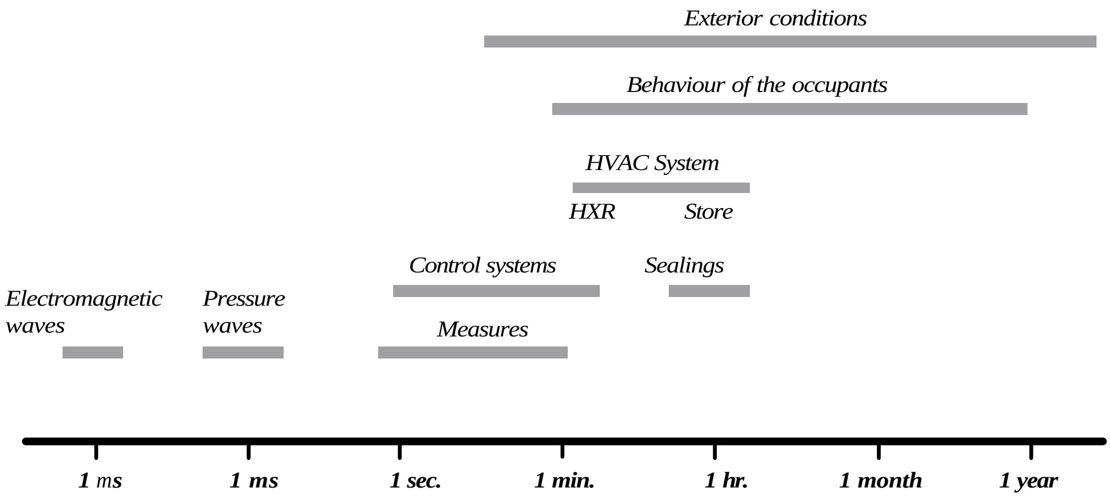


Figure 2-2: Typical values of time constants for different phenomena and subsystems (the time axis is not to scale)

The measuring systems sample the variables noted in period of time in the order of seconds to minutes, a range in which the control systems also act, with the proviso that the sampling time must be less than the time for decision-making. Depending on the HVAC system used, its time constant will range from a few minutes (from 2 to 5 minutes in the case of batteries), tenths of minutes (drying wheel) to a few or several hours for the accumulation systems (tanks or rock beds) or radiating floors. The time constant for the common exterior seals in buildings usually lies at around an hour. The occupants' preferences and their number vary depending on the season of the year and may also vary by a matter of minutes. It is the meteorological variants that cover the widest range: annual and interannual (seasons and climatic changes in the mid and long term), daily (day-night cycles, changes of time, etc.), timetable (changes of time) and other factors (occasional clouds, sudden changes in wind speed and direction, etc.).

Both the behaviour of the occupants and of the meteorological variables show a high degree of stochasticity and are usually described by specifying their values as input data. As regards occupant behaviour, it is common to define an occupancy profile as detailed as possible, either based on a statistical model or on measured or intuited data. As far as external conditions are concerned, current practice involves using a model year for the area concerned, obtained from statistically representative observations or generated synthetically from average values. Perhaps in the future, once sufficient measurements and greater computational capacity are available, it will be common to make simulations covering several *real*, consecutive years (this issue is covered in more detail in §5.2).

In order to study phenomena linked to a particular time scale, it is essential to formulate a model that takes into account the dynamic effects to be shown to that scale and higher, while any dynamic whose characteristic time is of a lower order than that of interest, it could be considered that it develops according to a series of stationary states. As a general rule, the more time goes by (lower resolution), the more the problem loses its differential nature and more process are described with algebraic (linear or non-linear) equations.

The first step to be taken when designing a mathematical model for the system in Figure 2-1 involves defining its purpose, then with the help of the diagram in Figure 2-2, deciding on the degree of detail in the representation of each subsystem. For instance, if the aim is to make a detailed evaluation of control strategies for HVAC equipment, the problem will be set out in the window [*seconds, minutes*], it being compulsory to consider the dynamics of the sensors, of the control equipment, of the HVAC systems and of the seals. The degree of detail in the models for these elements will, at all times, be conditioned by the information available, which is usually limited.

Typically, the purpose of thermal simulation for buildings has been to forecast the evolution over time of the interior conditions and electrical consumption in which HVAC equipment incur to maintain them. Experience has shown that this problem can be satisfactorily solved around a *timetable*[1]window, which limits the dynamic parts of the model for the building to ducting in the seals and, although not always, to the capacity to store heat and steam in the air found in the areas concerned. At this level, most of the HVAC systems are modelled on an ongoing system, and likewise all the controllers. As will be explained in §2.5, the latter may lead to problems of numerical convergence. Two factors work in favour of this level of modelling: their appreciably lower computational cost and the time cadence of the meteorological data that is normally available, a factor that limits solving the models in shorter periods of time. In view of the purpose of this work and taking into account the limited information available on the acclimatization equipment, this second *timetable* approach has been selected.

As far as modelling techniques and problem simulation are concerned, their nature is ideal for a modular approach, and in fact, this is implicit in the comments above. This involves dividing up the entire system, which is physically a continuous whole, into several appropriately interlinked subsystems. Once each subsystem has been modelled separately, there is freedom in the way in which they can be connected, which makes it possible to construct any system. Having reached this point, it is essential to define two aspects:

- WHICH ARE THE SUBSYSTEMS?: they may be those listed in Figure 2-2 or others of a greater entity. For instance, the building can be assembled based on seals, windows, air masses, etc., and the equipment can be based on the condenser, compressor, valves, frequency regulator, absorber, etc. Alternatively, the building can be wrapped up in a single subsystem and the acclimatization system in another. In reality, this is a matter of a hierarchy: the building can be modelled by assembling its individual components and encapsulating the whole in a single object that will subsequently be handled at a higher level, where the relationship between the building and the acclimatization system will be taken into consideration. The same applies to the equipment or, alternatively, a table with the manufacturer’s data can be used. This object orientated approach makes it possible to combine models with varying complexity in the same simulation, depending on the information available for each subsystem.

[1] Occasionally, lower time passages are used in order to ensure the numeric stability of the simulation or to detail usage profiles with less intervals.

- NUMERICAL RESOLUTION OF THE SYSTEM: once the system is assembled on its component subsystems, a numerical method should be defined to solve the interchanges (flows of heat and mass and signals). The main setback at this stage is the presence of dominions with highly differing time constants (Figure 2-2). This is a highly complex issue, some aspects of which are covered in §2.5.

This work uses the TRNSYS modular simulator, from the University of Wisconsin-Madison, which gives the platform numerical resolution. It also allows for the use of various subsystem models, the most important of these being the one for the building, termed TYPE56 (§2.3). Paragraph 2.2 provides an introduction, with a brief discussion of the thermal aspects of buildings and the basic transfer mechanisms of mass and energy occurring therein.

At the *timetable* level in which we have set the problem, the HVAC equipment are modelled in a stationary system. §2.4 comments on this, leaving §3 for a detailed description of the equipment model used in the simulations.

Finally, §2.5 discusses the technique adopted for coupling the building and its acclimatization system, a central issue in this work. Practical application of these ideas to the problem in question is dealt with in the following chapter.

2.2 Methods of balance and basic hypotheses

The physics of a building and its installations is complex and varied. We move on to consider some examples. Air-flow within an area arises from the combination of forced and floatation movements, dominating each other in different regions, depending on if it is near a drive or window grid, far from it, etc. The radiating properties of surfaces depend on the direction and length of the wave of the affecting radiation, and some components of air are participatory gases. The flow of coolant inside the heat pump interchangers is biphasic. The ducting system in seals is three-dimensional, especially in areas where different materials join or at corners. The speed and direction of wind is considerably altered by buildings and other adjacent obstacles. An explicit consideration of these phenomena requires detailed, computationally costly models, apart from more highly detailed information than is normally available.

The family of models nearest to the physical literality of the problem are the so-called TNMs (*Thermal Network Methods*), which discretize the building and its conditioning systems in a network of nodes, interlinked to represent the interchange of mass, moment and/or energy, depending on the basic transfer mechanisms (diffusion, convection, radiation …). In brief, the TNMs pose a classic problem of fields that is solved with a technique of finite differences or of finite volumes.

The majority of the balance methods currently implemented share, as far as the building model is concerned, hypotheses widely accepted that make it possible to capture significant aspects, the complexity of the model being in line with it purpose:

1. **Node model**: the air in each area is considered to be at a uniform temperature, i.e., perfectly mixed due to its own movement.

2. **Isotherm surface areas and one-dimensional piping in walls**: inside the seals, it is assumed that the temperature gradient running in the normal direction is far more marked and can be calculated uncoupled from the rest. Two- or three-dimensional effects are not modelled in detail, such as thermal bridges or, for example, local heat on a surface in the region where a solar spot falls.

3. **Dividing the spectrum into two bands: shortwave and longwave**: shortwave radiation (0.1 µm to 100 µm) derive from high temperature sources such as the Sun or incandescent lights, whereas longwave radiation (2.5 µm to 100 µm) mostly interchange between low temperature objects (radiators, building faces, etc.).

4. **Optically transparent air**: the air in an area does not participate in the interchange of radiation (one model assumes that the air is completely opaque to longwave radiation, but this does not appear to be a good approximation).

5. **Grey surface areas by bands**: the radiant properties of surface areas on seals (absorption factor and emissivity) are constant within each band of the spectrum and regardless of temperature.

6. **Diffuse emission and reflection**: the radiation emitted and reflected by a surface area is distributed equally in all directions.

7. **Glass is semi-transparent to shortwave radiation and perfectly opaque to longwave**

Each model can incorporate additional hypotheses. The range of operative conditions in buildings being quite narrow, let us say from -20ºC to -80ºC at temperature [Hagentoff] and with pressures around atmospheric, it is also normal to assume that the thermophysical properties of substances (density, thermal conductivity, etc.) are constant, which linearizes the constituent laws (Fourier, Fick, Darcy, etc.) and allows for approaches such as the transfer functions method to solve the heat diffusion in massive seals [Mitalas], §2.3.1.

Following these hypotheses, Figure 2-3 shows in diagram form the heat interchanges in any exterior seal, in the interior of an area, in the distribution network of a multi-zone system and in its production equipment. The thick lines represent shortwave gains, mostly deriving from the Sun, whereas the broken lines represent internal gains (radiators, electronic equipment) and heat interchanges by long radiation between surfaces. The continuous lines stand for mass flows, whereas the broken and dotted lines are control signals. In §2.3, a mathematical model is outlined for these interchanges and energy balances.

In each area, it is necessary to solve three parallel balances:

- Sensitive (air temperature
- Latent (absolute air moisture)
- Radiated

Latent balance affects the sensitive balance due to the effect of heat given off or absorbed in the phase changes occurring, for example, in the water absorption or desorption in the seals, whose thermophysical properties are also modified with amount of water that they contain. Normally, these crossover effects may be considered to be secondary and thus solve the sensitive and latent balances decoupled from one another.

Radiated balance should be approached in the control volume of each seal surface in order to determine, in conjunction with the other flows affecting the same, the temperature of said surface.

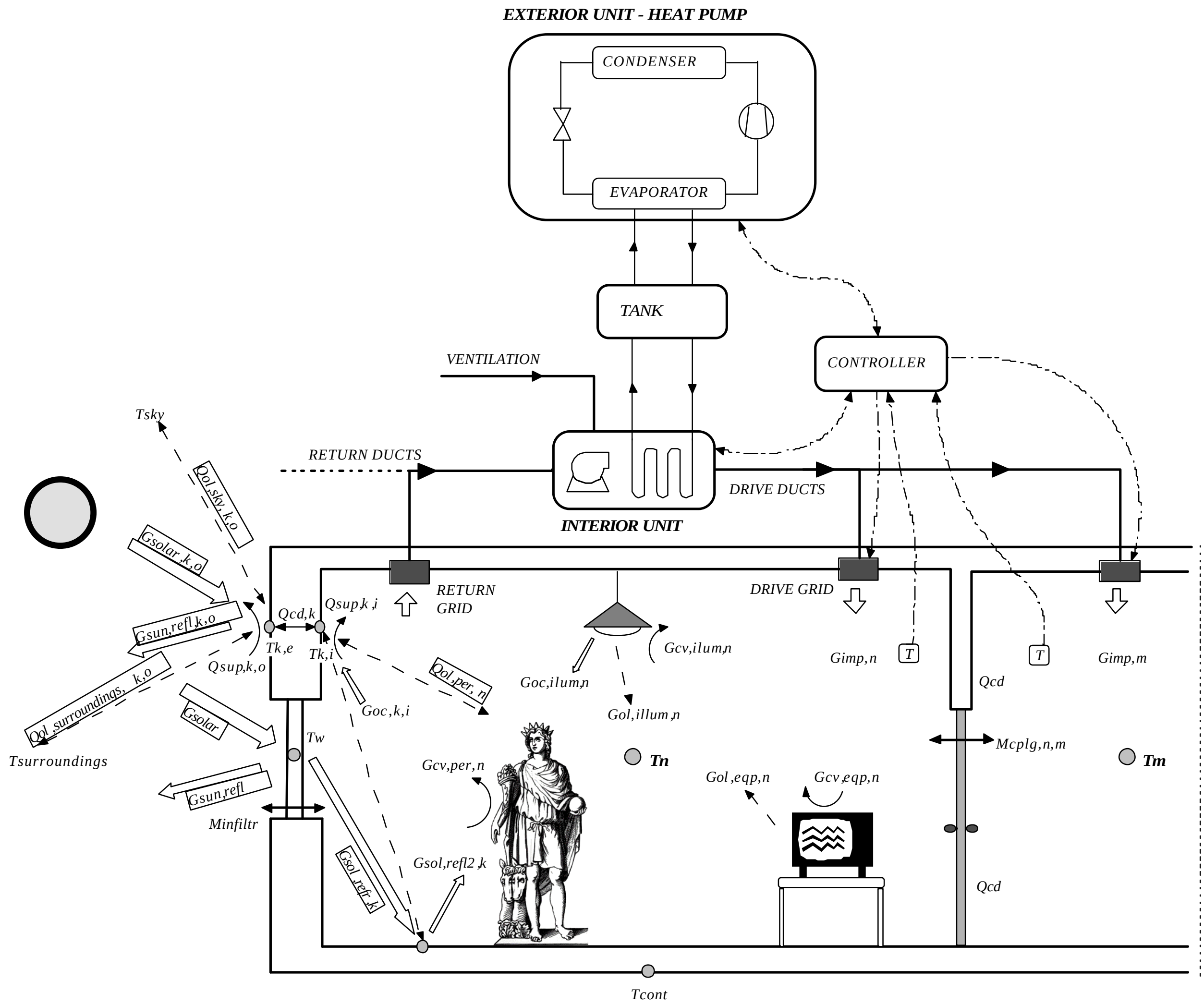


Figure 2-3: General diagram of energy interchanges in a building served by a multi-zone system with air-air heat pump.

2.3 Type 56 multi-zone building model by TRNSYS

The standard library included in the TRNSYS simulator used in this work has a full, validated model for a multi-zone building, termed TYPE56, implementing a non-geometric balance method. This is explained in general terms in this section.

As illustrated in Figure 2-4, to solve the temperature in each area, a system is needed with three balance equations per seal (exterior surface, piping in the seal, interior surface) and one equation for the air. Also, it is necessary to incorporate the transfer mechanisms (coupling between balances) and determine the value of the flows imposed.

Figure 2-4: terms in the sensitive balance of an area, adapted from [Pendersen]

In the centre of the problem is the air contained in the areas. So let us consider any area "*n*". The net convective flow affecting the air that it contains is (Figure 2-5)

$$Q_n = \sum Q_{sup,k,i} + Q_{gcv,n} + Q_{imp,n} + Q_{vent,n} + Q_{\inf,n} + Q_{\mathrm{cplg,m,n}} \quad , \qquad \textbf{(2.1)}$$

o, in terms of temperatures,

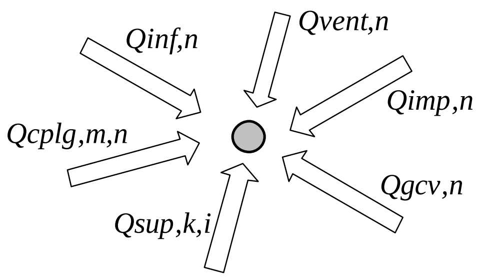


Figure 2-5: Convective heat flows to the node "*n*"

$$Q_n = \sum_{z=NSo}^{NSf} U_{z,i} A_{z,i} \left(T_{z,i} - T_n\right) + Q_{gcv,n} + \dot{m}_{imp,n} \left(r\, c_p\right)_{imp,n} \left(T_{imp,n} - T_n\right) + \dot{m}_{vent,n} \left(r\, c_p\right)_{vent,n} \left(T_{vent,n} - T_n\right) + \dot{m}_{\inf,n} \left(r\, c_p\right)_{\inf,n} \left(T_{amb} - T_n\right) + \dot{m}_{cplg,m,n} \left(r\, c_p\right)_{cplg,m,n} \left(T_m - T_n\right)$$

(2.2)

where

$\boldsymbol{A_{z,i}}$	Area of the interior surface of the seal *z*,
$\boldsymbol{m_{cplg,m,n}}$	Interchanged mass flow from the area *m* to the area *n*,
$\boldsymbol{m_{imp,n}}$	Mass flow driven by the HVAC equipment to area *n*,
$\boldsymbol{m_{inf,n}}$	Mass flow infiltrated into area *n*,
$\boldsymbol{m_{vent,n}}$	Mass flow of ventilation to area *n*,
$\boldsymbol{NS_o}$	Value of *z* for the first seal in the area *n*,
$\boldsymbol{NS_f}$	Value of *z* for the last seal in the area *n*,
$\boldsymbol{Q_{cplg,m,n}}$	Convective gain by infiltrated air flow from area *m* to area *n*,
$\boldsymbol{Q_{gcv,n}}$	Convective gains from internal sources (people, lighting, radiators, equipment, etc.) in area *n*,
$\boldsymbol{Q_{imp,n}}$	Convective gain by injected air flow via conditioning equipment,

$Q_{inf,n}$	Convective gain by air filtered from the exterior,
Q_n	Net convective heat flow to the node *n*,
$Q_{sup,k,i}$	Convective interchange between the interior side (*i*) of the seal *k* and the air in the are in contact with this side,
$Q_{vent,n}$	Convective gain by ventilation in the area *n*,
t	Time
T_{amb}	Room temperature
$T_{imp,n}$	Drive temperature to the area *n*,
T_m	Air temperature in the area *m*,
T_n	Air temperature in the area *n*,
$T_{vent,n}$	Ventilation temperature in the area *n*,
$T_{z,i}$	Temperature in the interior side of the seal *z*,
$U_{z,i}$	Overall transfer coefficient between the interior side of the seal *z* and the air,
z	Index that enumerates the seals in the building,
$(?c_p)_k$	Capacity of the mass flow *k = imp, vent, inf, cplg*,

If the result of the flows Q_n is not nil, the internal energy of the air will change. So the balance equation for the node "*n*" is

$$r c_v V_n \frac{dT_n}{dt} = Q_n = \sum Q_{sup,k,i} + Q_{gcv,n} + Q_{imp,n} + Q_{vent,n} + Q_{\inf,n} + Q_{cplg,m,n} \quad . \qquad \textbf{(2.3)}$$

For close the sensitive problem, the heat flows in the equation above should be expressed in terms of independent variables (temperature and air moisture, surface temperature of the seals) and of the surrounding conditions (atmospheric conditions, conditions of use, etc.).

The flows of infiltrated air (*inf*) and air from adjacent areas (*cplg*) concern one of the dominions alluded to in Figure 2-2, termed aeraulic. The aeraulic problem entails stirring the movements of air inside the building, which are provoked by differences of pressure induced by the wind and the HVAC equipment (forced movements) and by differences in temperature (self-induced movements). Solving this problem is complicated, so that normally a constant or correlated infiltration is fitted, in a simple manner, the term of coupling is cancelled. The terms of ventilation (*vent*) and equipment (*imp*) are dealt with in §2.5. Convective gains ($Q_{gcv,n}$) and internal radiations are tabulated in terms of the activity carried out by the people, in terms of the type of apparatus they derive from, etc.

The greatest difficulty arises in calculating the heat flow by convection with the surfaces in the room (*sup, k, i*). Calculating this requires the surface temperatures of the seals and windows (dotted block in Figure 2-4). Figure 2-6 shows the heat flows in a generic seal "*k*", with extreme temperatures $T_{k,i}$ (interior surface) and $T_{k,o}$ (exterior surface).

The balance equation for the interior surface is

$$Q_{cd,k} + Q_{oc,k,i} - Q_{\sup,k,i} - Q_{ol,k,i} = 0 \quad , \qquad \textbf{(2.4)}$$

where

***Q**$_{cd,k}$* Heat flow by conduction via seal,
***Q**$_{sup,k,i}$* Heat flor by convection with the air,
***Q**$_{ol,k,i}$* Net longwave radiating flow,
***Q**$_{oc,k,i}$* Radiating shortwave gain.

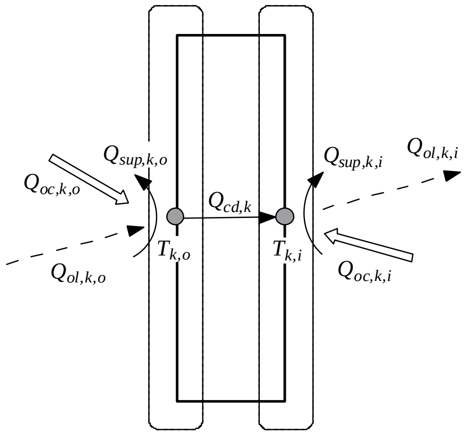


Figure 2-6: Control volume on the interior and exterior surfaces of the wall "*k*"

For the exterior surface, an identical equation to the above can be made, with the exception that here, the exterior conditions will intervene (atmospheric temperatures, outside and surrounding temperatures and incidental solar radiation).

The following subsections briefly comment on the treatment given in TYPE56 of the balance equation terms (2.4).

2.3.1 Piping in massive seals ($Q_{k,i}$,$Q_{k,o}$)

As is known, the overall transfer coefficient "*U*" is the simplest measurement of the thermal behaviour of a one-dimensional solid: it provides, under stationary conditions, the heat flow per unit of surface when the thermal hop between the end surfaces is 1 K. Nevertheless, two reasons limit its applicability in general:

1. The piping system in force in massive seals is usually transitional as it is continually subject to variable excitations over time.

2. The method of the global transfer coefficient does not preserve spatial provision for the layers of material. Different walls can have the same value of "*U*" but have very different dynamic behaviours.

This stationary behaviour only gives satisfactory results when applied to common glazing or opaque seals with low thermal mass. For massive seals and, in general, for heterogenous seals, it is necessary to solve the heat diffusion equation in the seal. Nevertheless, this problem is computationally costly in the context of an annual simulation due to the fact that numerical methods (Euler, Crack-Nicolson …) present a critical time passage and calculate the internal distribution of temperatures in the seal.

Various authors [Stephenson] have developed alternative methods to solve this under the central hypothesis of linearity in the diffusion equation (constant thermophysical properties). The most used are the response factors method and the transfer functions method, both deriving from the general theory of linear systems. The response factors relate the output of a linear system with its current and past inputs. Transfer functions, used by TYPE 56, add past outputs to this relationship, which considerably reduces computational costs. Heat flows on the end surfaces of the wall "*k*" are expressed in terms of the thermal history of the wall[2] as follows:

$$q_{k,i}(t) = \sum_{z=0}^{Nbk} b_k(z) T_{k,o}(t - zd) - \sum_{z=0}^{Nck} c_k(z) T_{k,i}(t - zd) - \sum_{z=1}^{Ndk} d_k(z) q_{k,i}(t - zd) \quad , \qquad \textbf{(2.5)}$$

$$q_{k,o}(t) = \sum_{z=0}^{Nak} a_k(z) T_{k,o}(t - zd) - \sum_{z=0}^{Nbk} b_k(z) T_{k,i}(t - zd) - \sum_{z=1}^{Ndk} d_k(z) q_{k,o}(t - zd) \quad , \qquad \textbf{(2.6)}$$

where d is the simulation time passage and a_k, b_k, c_k and d_k are the coefficients of the transfer function for the wall that depend on the thermal properties and on the arrangement of constituent layers. The advantage of this method lies in the fact that it is only necessary to calculate these coefficients once for each wall. Once they are known, the series (2.5) and (2.6) correctly approximate the solution of the heat diffusion equation. The number of coefficients per wall depends on how massive they are. This ranges from k = 2 for walls with a lot of inertia to k = 1 for walls defined with no thermal mass (the series are reduced to the current term).

2.3.2 Solar radiation and internal sources radiation at high temperature *($Q_{oc,k,i}$,$Q_{oc,k,o}$)*

Solar radiation falling on exterior surfaces *($Q_{oc,k,i}$,$Q_{oc,k,o}$)* is calculated, as a minimum, based on the total radiation on horizontal surface using widely acknowledged models [Duffie]. Part of this radiation will penetrate the room through the windows and will, in a different way, fall on the interior surfaces depending on the geometry of the enclosure and on the sun's position.

By saying that the TYPE56 implements a non-geometrical balance method we refer to the fact that it does not consider the geometry of the room. So in order to distribute radiation gains and long-wave interchanges, a criteria must be articulated based on the information available: absorption factors and the area of interior surfaces. On this point, differentiation will be made between primary direct solar radiation and the primary radiation reflected. The former, coming from a solar spot, is distributed according to coefficients known as *GEOSURF(k)*, defined by the user for each interior surface k, i. The other two components are distributed among all the surfaces in the room in terms of their absorption factor product * area,

[2] [Seem] strictly derive these expressions formulating and solving the heat piping problem in the area of states. To do so, it measures the application of a diagram of differences or finite elements for the spatial dimensions, giving rise to a system of concentrated parameters that approximates the original.

$$f_{c,k} = \frac{a_{k,i} A_{k,i}}{\sum_{z}^{walls} \left(1 - r_{diff,z,i}\right) A_{z,i}} \quad , \qquad \textbf{(2.7)}$$

where $a_{k,i}$ is the absorption factor on shortwave of the seal k,i and $?_{dif,z,i}$ the reflectivity of the interior side of the seal z.

The total short-wave radiation absorbed by the surface k, I can thus be written as

$$QSIAB(k) = QIBAB(k) + QIDAD(k) = GEOSURF(k) \cdot a_{k,i} \cdot QS_{prim} + f_{c,k} \cdot \left(QS_{reflec} + QS_{diff}\right)$$

The total radiation from internal sources is distributed in line with the relationship of areas. The fraction of internal radiation gains falling on the surface k, I

$$f_{l,k} = \frac{A_{k,i}}{\sum_{z=1}^{walls} A_{z,i}} \quad . \qquad \textbf{(2.9)}$$

The form factors between interior surfaces, necessary for calculating radiation interchange, are also approximated by (2.9).

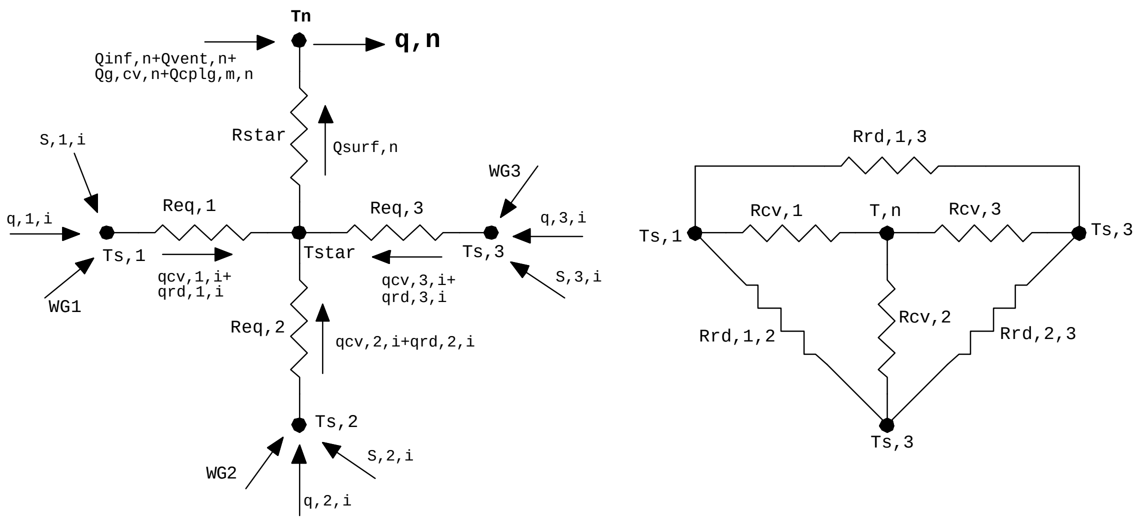
2.3.3 Convection $(Q_{sup,ki}, Q_{sup,k,o})$

Convective exchange between the interior surface of the seal "*k*" and the air in zone "*n*" is given by Newton's Law for convection where $A_{k,i}$ is the interior surface area of the seal, $h_{k,i}$ the coefficient of film on said surface and $T_{k,i}$ is its temperature. Convective exchange is usually a combination of natural and forced, depending, therefore, on the orientation of the surface, of its roughness and on the interior flow pattern. The exact calculation of the average coefficient of film for each surface requires computational fluid mechanic calculations. To avoid this complication, it is common practice to use a constant coefficient of film or a simple experimental correlation. For the exterior side, a similar equation to (2.10) can be proposed.

On the inner surface of the seals, convection is solved along with the long-term exchange, as explained in the following section.

2.3.4 Radiating exchange between walls and exterior convection $(Q_{ol,k,i}\ ;\ Q_{cv,k,i})$

These mechanisms mutually couple the balances in all the surfaces of the room and are treated jointly by the TYPE56 using the star network method [Seem]. Figure 2-7 (b) shows the electrical diagram for the real exchanges in an enclosure with three surfaces radiating between each other (*Rrd*) and convecting with the node in their zone (*Rcv*). The presence of direct transfer paths between surface nodes, as a result of long-wave radiation, means a substantial mathematical complication in terms of obtaining a system of equations able to relate exterior (known) excitations with the temperature of nodes or interior nodes (*room transfer function*).



(a) (b)

Figure 2-6: Star network (a) and electrical diagram of convective-radiating exchanges (b).

The star network equal to the real network is shown in Figure 2-7 (a). Here, a definition is given of a virtual STAR node to which, from each surface and via the corresponding equivalent resistance, the combined convective-radiating flows arrive. This network facilitates a direct path between the exterior and the interior: a balance on the exterior surface (equation 2.4) ? conduction in the seals (equations 2.6 and 2.7) ? a balance on the interior surface (equation 2.4) ? star network ? air node.

The resistances on the real network 2-7 (b) are calculated in terms of the interior convection coefficients ($h_{k,i}$), long-wave emissions (e_i), the areas of the surfaces (A_i) and the shape factors (F_{ij}), such as

$$R_{cv,i} = \frac{1}{A_{k,i} h_{k,i}}, \qquad \textbf{(2.11)}$$

$$R_{rd,i,j} = \frac{1}{4 s e_i A_i \bar{T}^3 F_{ij}}, \qquad \textbf{(2.12)}$$

where s is the Stefan-Boltzmann constant and $\bar{T}$ the approximate average temperature of the surfaces in the zone, a term from the linearization of the radiating exchanges and which is usually taken to be a constant.

Transformation of the real network into the star network is not one-to-one since the problem is overdetermined, except in the case of an enclosure with two surfaces, when the change is reduced to the known star-triangle equivalence. In [Seem], a procedure is proposed that adds some conditions to the equivalence properties between both networks.

The terms in the equation (2.4) are treated in this manner. By assembling the entire problem, which for reasons of brevity we shall not detail more elements, a system of linear equations can be arrived at for a multi-zone building

$$X \cdot \vec{T} = \vec{Z}, \qquad \textbf{(2.13)}$$

where $\vec{T}$ is a vector formed by the temperatures of the air in the zones and the temperatures of the interior surfaces of the building seals, X a matrix that depends on all the parameters of the problem (coefficients of the transfer functions, convective and radiation resistances) and the air flows (ventilation, infiltration, coupling and equipment), and Z a vector that depends on exterior excitations (atmosphere) and on exchanged air flows.

By solving the system of equations (2.13), for each passage of time, we obtain the interior conditions in all the zones, including the temperatures of the seals, with which the operational temperature may be obtained. For a full formulation, see the following references [TRNSYS manual], [Thesis Eduardo], [Pedersen] and [Seem].

2.4 Modelling HVAC equipment

Following the modular approach put forward in §2.1, we studied the building model and now turn to the equipment model. Modelling primary HVAC systems (compression, absorption or adsorption machines, solar systems, cogeneration, etc.) and secondary systems (fluid networks, radiating floors, batteries, etc.) this is inherently more complicated than modelling for buildings, and this is due to various reasons [Gough].

1. HVAC systems are built with a wide variety of components;
2. These components can be connected in an infinite number of ways, creating a complex network of fluids and monitoring;
3. Many of the physical process involved are notably non-linear, such as the condensation and evaporation processes, the mass pressure-flow characteristics of the components in the fluid distribution networks, the all-nothing control actions with hysteresis, etc.;
4. Almost all the equipment have far lower response times than the building equipment (Figure 2-2), which complicates the numerical resolution methods in the joint problem.

As commented in §2.1, due to reasons of computational cost and the information available, for annual simulations it is common to consider only the dynamics of the seals in the building and possibly the air dynamics in the zones. In this context, the HVAC equipment are modelled on a permanent system.

Whereas in the case of secondary systems it is normal to use models based on physical considerations, as occurs for the building (§2.2, §2.3), the far greater complexity of the primary systems means that ongoing test data, provided by the manufacturer, are used, either in table form or in functional form, typically, polynomes adjusted by regression analysis. When this information is not available, it is possible to turn to experimental identification procedures.

The solutions adopted for the problem in point are explained in detail in the following chapter.

2.5 Joint solution for the building and HVAC equipment

One critical aspect for the problem dealt with is the need to solve the building and its HVAC systems together, in order to determine the evolution of the interior conditions in the non-controlled zones of the non-Zoning system or to be able to make a correct calculation of the thermal demand in the active zones of the Zoning system when some of the adjacent zone(s) is/are not being acclimatized, since this fact influences heat exchange though the separation partitions. This section describes this problem and the basis of the solution adopted which, in the following chapter, is applied to the particular case in point.

Generally, the problem of acclimatization involves three basic elements that have already been introduced: the building, the secondary HVAC system and the primary HVAC system. Even with detailed models for all of them (§2.3, §2.4), the question now is how to assemble the whole (Figure 2-8).

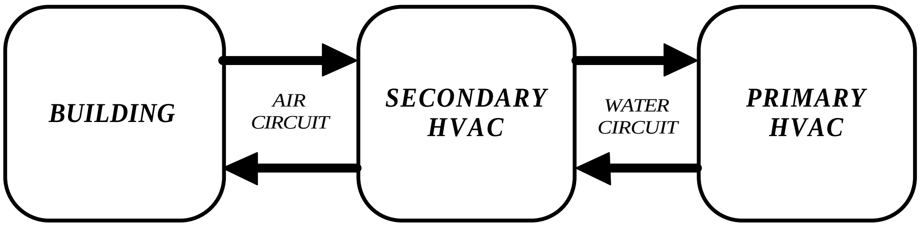


Figure 2-8: Diagram showing the acclimatization problem

For the sake of simplicity and without losing a general outlook, let us assume that it is a uni-zone building. The air temperature will evolve according to the equation (2.2), where by grouping together terms, we obtain:

$$r c_v V_n \frac{dT_n}{dt} = Q_{int,n} + Q_{imp,n} \qquad (2.14)$$

where $Q_{imp,n}$ is the flow of heat injected into the zone by the secondary system and $Q_{int,n}$ is the sum of the rest of the exchanged flows.

The action of the equipment should tend to maintain comfortable conditions in the controlled zone so that $Q_{imp,n}$ depends on the temperature in the zone by means of a control law. The control system samples the conditions in the zone every few seconds and takes decisions, the effects of which are noticed later in the air. The time constant for the dynamic processes occurring in the air is obtained by adimensionalizing (2.14),

$$t = \frac{r c_v V_n}{\left|\dot{Q}_{imp,n} + \dot{Q}_{int,n}\right|} \qquad (2.15)$$

and its value is not constant as the denominator changes.

Although the transitories in the HVAC system are faster than t, if the temperature sensor is fitted in the return, the system has to wait a time t to determine the result of an action. In any case, this is a problem with various and different dynamics: control, air and seals (Figure 2-2). To solve it, ideally a numerical method is used to deal with each dynamic separately, with a time passage of its own that is able to adapt. Interactions between dominions exist and need to be considered, but on a nested basis (e.g., seals may be solved each hour but with the average air temperature resulting form calculations for a lesser passage of time).

The previous approach entails two setbacks,

- High computational cost
- TRNSYS uses a numerical method with a constant passage of time.

As commented in §2.2, the interest in our simulation does not lie in the detail on control, but on the equipment's performance. For this reason, we are in a *timetable* window, with time passages from one hour to fifteen minutes, which solves the two difficulties noted above. This time passage, however, does not give a solution to simulate the detail of the action of the control systems and equipment. The solution adopted involves assuming that, at the end of a period of time in the order of the time passage of the simulation, which is far greater than (2.15), the equipment will be able to take the final temperature in the zone to its pre-set value, or as near as possible, if for any reason, it lacked sufficient power to be able to do so.

This involves an *average* in time that establishes a final state for the air in the controlled zone, regardless of how the equipment control system manages to achieve it. Thus, the detail of the equipment dynamic is lost to lower scales of time and is replaced by its desired effect, which, formally, is a condition directly applicable to the thermal balance model for the controlled zone, which may then be solved with a long time passage (Figure 2-9).

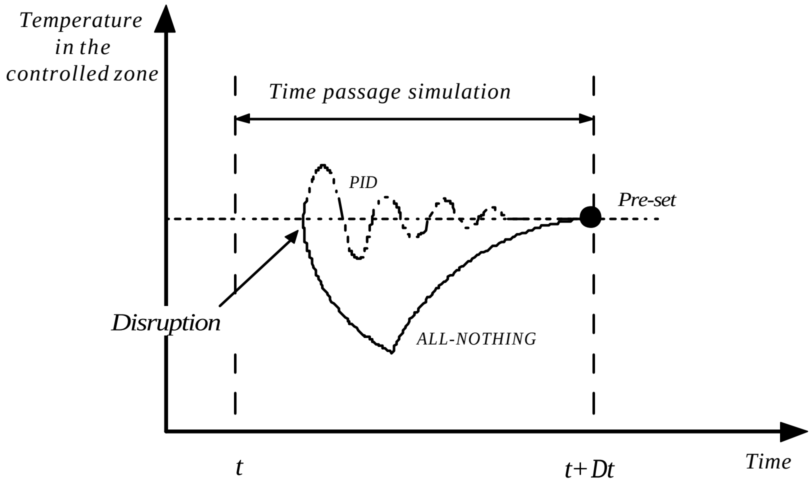


Figure 2-9: The passage of time does not make it possible to make a detailed simulation of control systems, so that it is imposed that the final temperature of the time passage is equal to that of the pre-set temperature or to the nearest achievable value at full power. It is foreseeable that, on average, oscillations in electrical consumption caused by control will tend to cancel each other out.

Finally, it is noted that when, in this document, temperature is referred to in a time passage (§6), we refer to the average value between the initial and end temperatures of the passage, and not to the final value. The final value is only used as a condition to adjust the power that the equipment injects into the zone.

THIRD CHAPTER
Systems modelling

3.1 Introduction

Based on ideas explained in the previous chapter, we move on to describe the models implemented in TRNSYS for each acclimatization system: non-Zoning (§3.2) and Zoning (§3.3). These models solve the thermal dynamics of the problem, calculating for each time passage the temperature in the zones and the thermal energy required by the primary equipment, the amount for which is limited to the full load value provided by the manufacturer in the equipment catalogue. To transform the thermal demand into electrical consumption, the approximated models detailed in §3.4 are applied for all-nothing machines with an inverter.

3.2 Non-Zoning system

3.2.1 Description of the system

This system comprises an air-air heat pump, with interior units (ventilator and direct expansion battery) and exterior units (rest of the components), and a network of ducts to distribute the treated air to the various rooms (Figure 3.1). Machine control is based on a single temperature, typically that found in the main room of the house (living room).

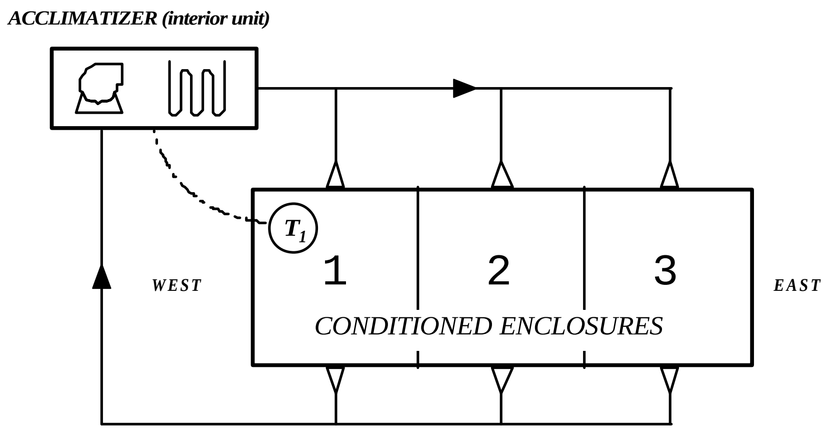


Figure 3-1: Uni-zone system serving a multi-zone building. Equipment control depends on enclosure “1”

Based on the measurement taken by the sensor and on user specifications (instructions and heat or cold mode), the control system decides, at each moment, if the machine should function or not. If it is equipment with a frequency variator in the compressor engine (*inverter*), it modulates the cooling flow to cover the part load more gently and efficiently that would an all-nothing equipment.

Figure 3-2 shows the operation regions according to the controlled temperature: a comfort band (let us say, from 22ºC to 24ºC) within which the equipment is not required to act, a heat mode band (<22ºC) and a cold mode band (>24ºC). When the equipment functions in heat mode, it attempts to maintain the temperature in the zone under control, at 22ºC, when operating in cold mode, at 24ºC.

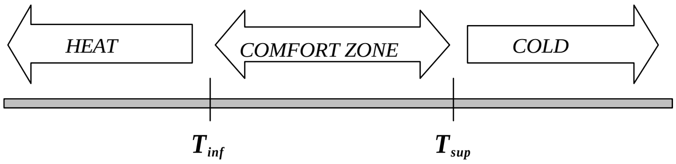


Figure 3-2: Equipment control regions

This system will fulfil its task when all the acclimatized enclosures can be grouped together in a single thermal zone, by presenting similar load and occupancy profiles (unless it can never stop acclimatizing unoccupied enclosures). Nevertheless, this is not normally the situation, so that the uni-zone system is incapable of meeting the different needs in each zone at the same time, either undercooling and/or overheating some of them. As illustrated in Figure 3-1, by way of an example, this circumstance could be due to the different orientation of the exterior seals (east-west) which, in an extreme case, would lead to simultaneous thermal inversion, i.e., the need for cold on the sunny side and for heat on the opposite side. These difficulties can be cleared up with the following comparison: the non-Zoning system is equivalent to fitting a switch in the living room that controls the lighting for the entire house, according to the level of lighting in the living room.

The lower initial total cost of this system has meant that it is commonly installed in houses, despite its higher operational cost and poorer performance, as demonstrated in §6.

3.2.2 Hypothesis

To the hypotheses listed in §2.3 for the building model, the model proposed for the non-Zoning system adds the following:

1. The network of ducts is well insulated and sufficiently short as to be able to make the load in the equipment negligible through heat exchange between return and drive currents.
2. The network of ducts is assumed to be perfectly balanced. During installation, the relevant elements will have been adjusted so that each zone receives its design mass flow.
3. Users do not alter the thermostat setting to achieve comfortable conditions in other zones other than the controlled zone (living room). Any criteria for change in setting for this reason would be quite arbitrary.
4. The windows remain closed all the time. A constant infiltration value is taken into account (§4.1.2).
5. No movements of air occur between the zones in the house, except via the equipment.
6. The kitchen does not return and stops being acclimatized when it is unoccupied (extractor hood turned off).
7. Occasionally, differences in temperature occurring naturally between different zones in the house may be sufficient, only by starting up the ventilator, to give comfortable conditions in the living room. It is assumed that the control system is capable of detecting these situations.

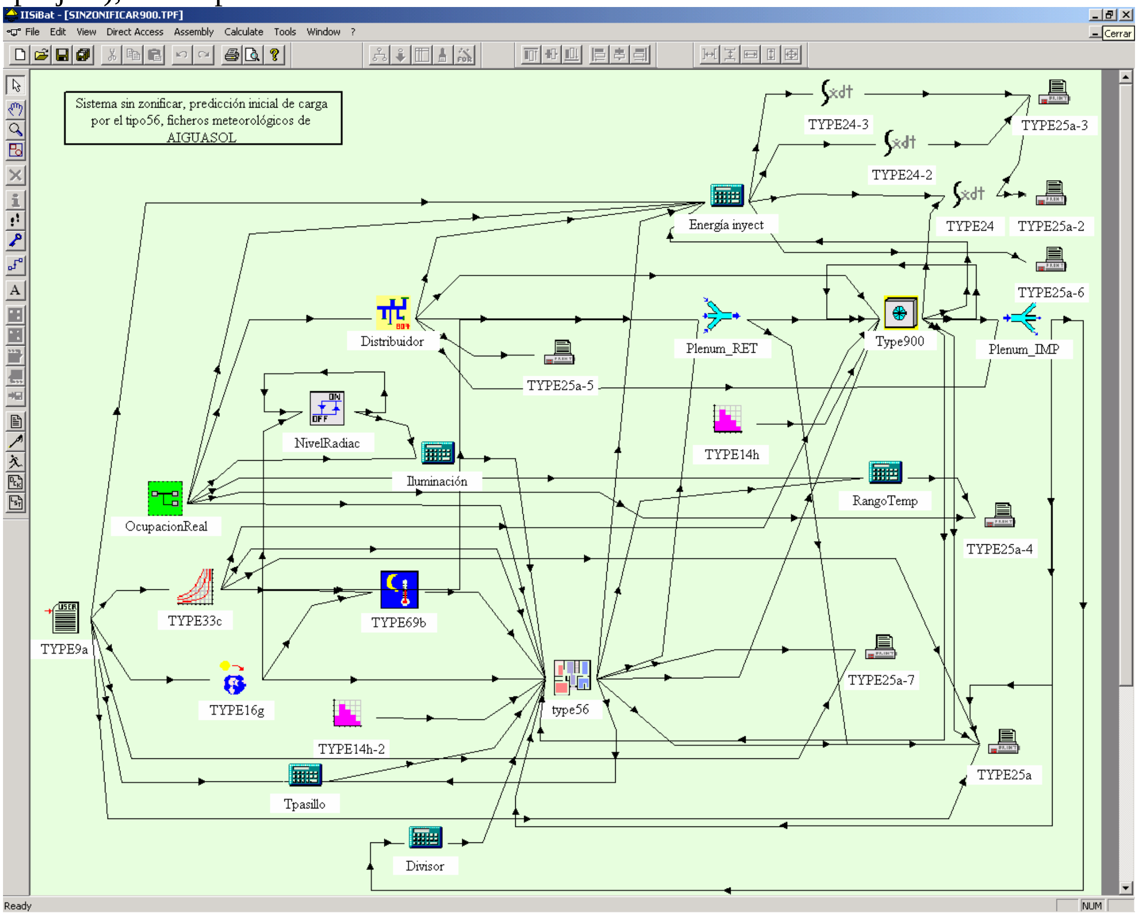
Hypotheses referring to the equipment are explained in §3.4, as they are also applicable to both acclimatization systems.

3.2.3 Implementation of the model in TRNSYS

This system is designed to maintain the controlled zone (zone 1 in Figure 3-1) within the region of comfort (Figure 3-2). The logic of the model involves three consecutive stages that are executed to solve the problem at each passage of time.

1. FIRST PHASE, where it is determined if the action taken by the equipment is necessary so that the controlled zone is in comfortable conditions. If this is the case, a first estimate is made of the power that the equipment would required and if it is in heating or cooling mode.
2. SECOND PHASE: only executed when the equipment has to operate. Finally, the thermal power injected into the building is adjusted, i.e., the balance of energy is closed in the air circuit (building-equipment).
3. THIRD PHASE: electrical consumption is calculated (§3.4).

Figure 3-3 shows the assembly in TRNSYS for the entire system. Figure 3-4 illustrates the main components in the air circuit: building (TYPE56), equipment and control (TYPE808, programmed in this project), return plenum and network of ducts.



Non-Zoning system, initial load forecast by type 56, meteorological files by AIGUASOL

Radiation Level
Real Occupancy
Lighting
Temp. Range
Divisor
PassagewayT

Figure 3-3: Graphic illustration for the non-Zoning system. The most important components are the Type808 (primary and control equipment), Type56 (building), the plenums, the distributor (air distribution network) and the lighting controller (Radiation Level, see §4.1.3).

Following these stages, in the first repeat of any time passage, the model determines if the equipment should or should not operate to maintain conditions of comfort. To do so, it cancels the drive from the equipment and calculates the resulting temperatures in each zone of the house.

Taking into account the final controlled temperature value (T_{f1}) three situations are possible. Figure 3-5:

- COMFORT, $T_{f,1}(t+\ ?t) \in [T_{inf}, T_{sup}]$: the action of the machine in the time passage is not necessary
- HEAT, $T_{f,1}(t+\ ?t) < T_{inf}$: the machine should operate in heat mode
- COLD, $T_{f,1}(t+\ ?t) > T_{sup}$: the machine should operation in cold mode

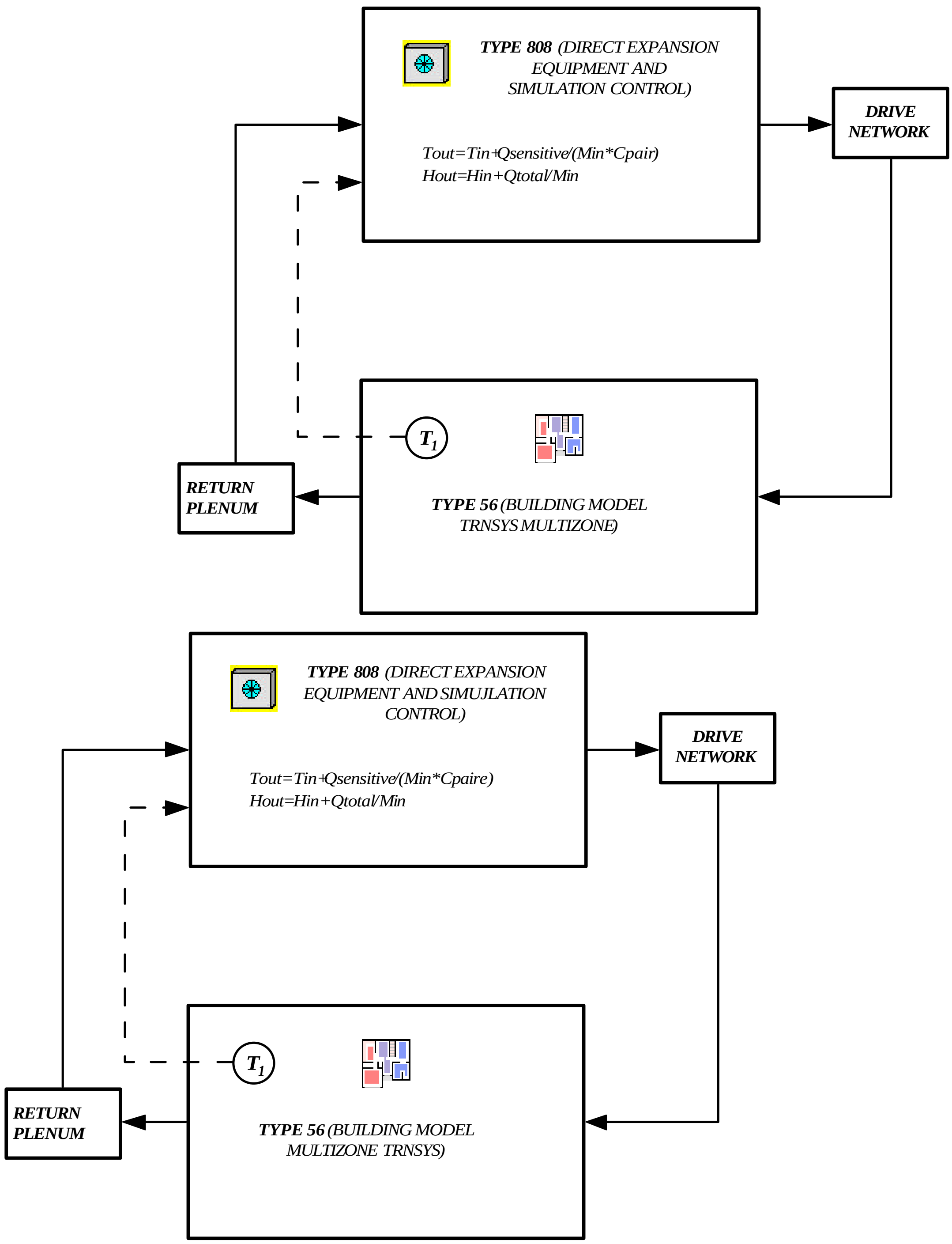


Figure 3-4: Air circuit coupling the building with the HVAC equipment

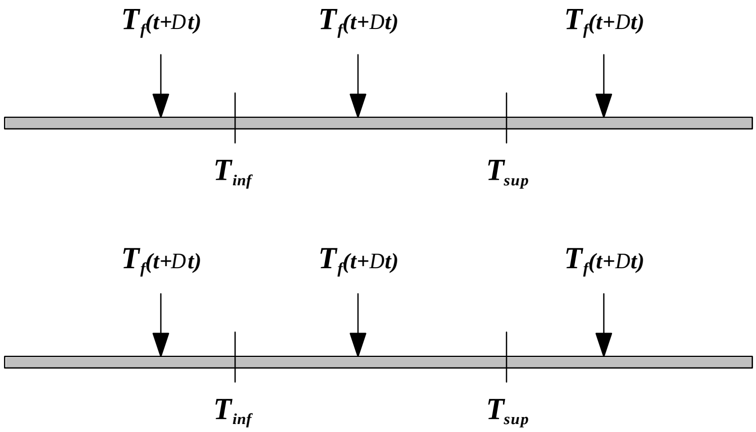


Figure 3-5: Possible locations for the final temperature of the controlled zone with the equipment turned off

If the balance temperature of the controlled zone falls into the region of comfort, the problem is solved and the time passage is taken as completed. If this is not the case, the potential need is repeatedly adjusted to take T_{f1} to the nearest point in the region of comfort: in heat mode, the intention is to reach T_{inf} (22ºC) and in cool mode to T_{sup} (24ºC). Formally, in the equation (2.14), the desired value of the controlled temperature is imposed, let us assume that T_{inf}, and the power necessary to condition the zone is cleared

$$T_{f1} = T_{inf} = cte \Rightarrow \frac{dT_1}{dt} = 0 \Rightarrow Q_{imp,1}\left(T_{f1} = T_{inf}\right) = -Q_{int,n}\left(T_{f1} = T_{inf}\right). \qquad \textbf{(3.1)}$$

The total potential given by the equipment is calculated by scaling the previous value with the mass flows into the controlled zone (m_1) and the total carried by the ventilator (m_{total}) and adding an estimate of the load due to ventilation

$$Q_{eqp,total} = Q_{imp,1} \cdot \frac{m_{total}}{m_1} + Q^*_{vent} \ . \qquad \textbf{(3.2)}$$

So far, what has been done is to calculate the loads for the living room, equation (3.1), maintaining the condition that the other zones do not receive flow from the equipment (the resulting temperature in these zones is calculated). Obviously, this situation is not real, so that (3.2) is still not a solution.

If the equipment works, in the second phase, we then repeatedly solve the air circuit (Figure 3-4), i.e., adjust the sensitive power that the equipment should give to the return current so that the living room temperature reaches the desired value. At this phase, the air circuit closes, considering the return from all the zones and the subsequent drive to the same.

Whereas the first phase solves the problem when the machine is stopped, the second does so when it is operating. A power estimate is obtained from the first phase, and this serves as a seed for repeating in the second phase, and the indication of if the machine should operate in cooling or heating mode. Although this decision on the mode is normally correct, occasionally, it is necessary to check on it due to the fact that it was obtained by considering only the load in the living room:

- It may be the case that the living room alone demands heat/cold but when mixing returns and driving, the situation changes and demands cold / heat since the other zones are excessively cold/hot
- A frequent case in spring and autumn is that, in the second phase, the adjusted sensitive power is zero. This means that only by starting up the ventilator can the difference in temperature between zones acclimatize the living room. As shown in §3.2.2, it is assumed that the control system can detect this situation.

Checks are run to ensure that the total power given by the machine (that is governed by the living room temperature control) does not exceed the full load value. If this occurs, the power given to the maximum possible value is cut short and the resulting temperatures are calculated in the zones.

Once the second phase is completed, the sensitive and latent powers given by the machine and the resulting temperature in each zone will have been obtained. In the third phase, calculation is made of the electrical consumption of the equipment by the models in §3.4

3.3 Zoning system with false variable flow

3.3.1 Description of the system

In order to control the conditions in each enclosure separately, all-nothing motorized terminal dampers are installed in the drive branches. Control is performed by a central unit that receives instructions from users and the temperature in the zones, it then decides actions on the dampers, if the machine should operate or not and in which mode it should do so (heat or cold). This control system does not replace the machine's internal control.

Once a function mode (heat/cold) has been set for the entire house, the objective in each enclosure is to maintain its temperature within the band of comfort (Figure 3-2). This is achieved by opening and closing its drive damper.

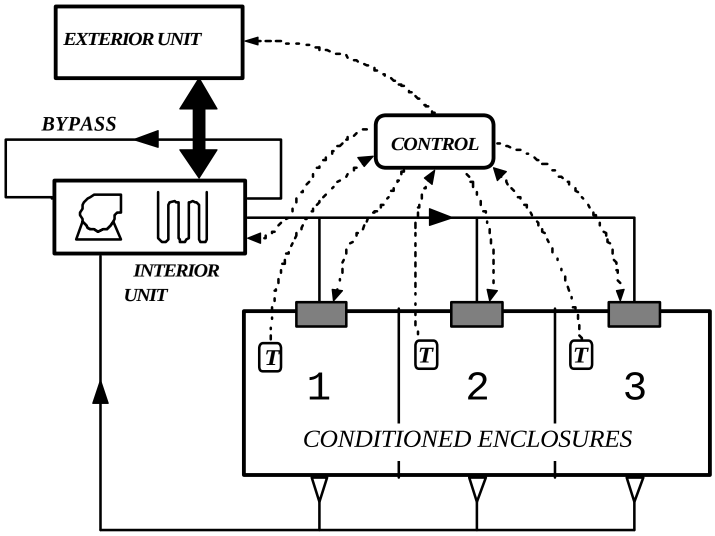


Figure 3-6: Multi-zone system with variable false flow

This damper opening and closing process changes the total driven mass flow, with the result that, ideally, it would be necessary to have a variable speed ventilator. Alternatively, the constant speed ventilator is maintained and a gravity activated by-pass is added that recirculates part of the excess flow in the network, dumping it in the return plenum.

The disadvantages to this system are:

- Greater initial cost than for the Zoning system
- Air recirculation worsens the equipment's COP (in heat mode, it increases the incoming temperature and the opposite occurs in cold mode)

The advantages to this system are:

- It maintains the conditions of comfort in all the enclosures (except in cases of simultaneous thermal inversion), introducing only the energy required)
- It can stop acclimatizing unoccupied enclosures
- Thanks to the lack of simultaneity in the demand from different enclosures, a lower powered machined than the one used in the Zoning system can be installed
- Different instructions can be set for each enclosure, depending on user preferences.

3.3.2 Calculation hypothesis

To the hypotheses listed in §2.3 for the building model, the model proposed for the non-Zoning system adds the following:

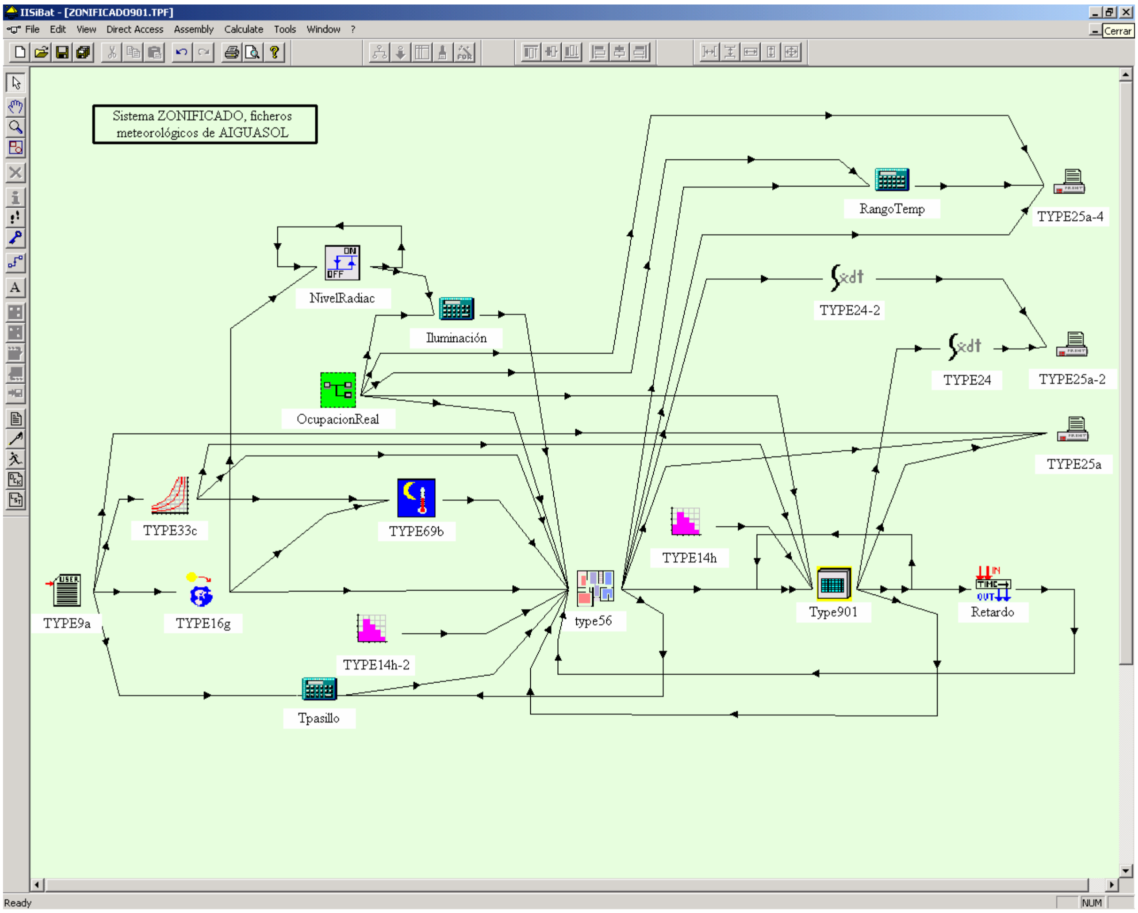
1. The network of ducts is well insulated and is short enough for the load in the equipment through heat exchange between the return and drive currents to be negligible;
2. Certain imbalances are allowed in the network of ducts when the dampers close as a result of the unsuitability of the by-pass;
3. The windows remain closed all the time. A constant infiltration value is taken into account (§4.1.2);
4. No air movements occur between the different areas in the house;
5. The non-acclimatized zones do not return. It is not foreseeable that any considerable differences in pressure will appear that may lead to the contrary, as there is no overpressure originating the drive flow into a zone;
6. The kitchen does not return.

The hypotheses on the equipment are explained in §3.4, as they equally apply to both acclimatization systems.

3.3.3 Implementing the model in TRNSYS

This system is designed to maintain conditions of comfort in all the occupied zones and not to acclimatize the unoccupied zones. This simplifies the relation between the building and the equipment model as regards the previous system (§3.2.3), as there is no return from non-controlled zones.

Figure 3-7 shows the assembly in TRNSYS for this system.



ZONING system, meteorological files from AIGUASOL

Temp. range

Level of Radiation

Lighting

Real occupancy

PassagewayT

Figure 3-7: Assembly in TRNSYS of the multi-zone false variable flow system

In the first repetitions of each time passage, the model sets the final temperature of the occupied zones in the set value (T_{inf} or T_{sup}, depending on the season) and leaves the temperatures for the other zones floating. Thus, the sensitive for the occupied zones and the resulting temperature for the unoccupied zones are calculated (this is important because it influences in the demand value for the occupied zones).

The next phase is to ascertain if, at full load, the equipment is able to meet the sum of the sensitive and ventilation demand. To do so, a mix is made of the return currents from each occupied zone[1], exterior ventilation air and air recirculated by the by-pass, to calculate the temperature and relative humidity of the current entering the battery in the interior unit. Referring to the manufacturer’s table (§3.4) with these conditions and with those for the interior unit, the machine’s maximum sensitive contribution is interpoled. If this is less, temperatures for the acclimatized zones are re-calculated until the sensitive power required is equal to that of full load.

Finally, the electrical consumption for the equipment is determined by the models commented in the following section.

3.4 Modelling the direct expansion equipment

The equipment has been incorporated in the TRNSYS assemblies as a new component: TYPE900 for the non-Zoning system (Figure 3-3) and TYPE901 for the Zoning system (Figure 3-7). Both components are special as they jointly control the *solver* in TRNSYS, the different phases of the calculation process explained in sections §3.2.3 and §3.3.3, apart from performing the thermal and electrical calculations explained below.

The information available on the equipment is found in the manufacturer’s catalogue: table of total capacity, sensitivity and electrical consumption at full load in terms of the sources as well as some data on performance at part load. The tables are calculated for a narrow range of values,

- HEATING: $T_{db,evaporator}$=-15ºC at 10ºC (6 values) ; $T_{db,condenser}$=16ºC at 24ºC (6 values),
- COOLING: $T_{db,condenser}$=25ºC at 40ºC (4 values); $T_{db,evaporator}$= 22ºC at 32ºC (6 values); $T_{wb,evaporator}$=16ºC at 24ºC (6 values),

which means a limitation for simulations, especially for those in a Zoning system, which experiences more extreme conditions at the entrance to the interior battery as a result of the by-pass. In cooling mode, the lack of information is evident because the relative humidity shown in the points appearing on the table are always in the region of 50%.

[1] In proportions related to the fraction of the time passage, which, on average, each damper will be driving.

For heat mode, the exterior temperature range has been broadened to 25ºC by regression analysis. Excepting this aspect, when operational conditions go out of range, the nearest higher value conditions in the table are taken. For points within the range in the table, a two-dimensional interpolation technique is applied for heating, and a three-dimensional technique for cooling, which provides the following continuous functions, descriptive of the technical characteristics of the direct expansion equipment at full load:

HEAT MODE

$$Q_{total,calor} = Q^{PC}_{total,calor} \cdot f_1\left(T_{wb,amb}, T_{db,ret}\right), \qquad \textbf{(3.3)}$$

$$COP_{calor} = COP^{PC}_{calor} \cdot f_2\left(T_{wb,amb}, T_{db,ret}\right), \qquad \textbf{(3.4)}$$

COLD MODE

$$Q_{total,frio} = Q^{PC}_{total,frío} \cdot f_3\left(T_{db,amb}, T_{wb,ret}\right), \qquad \textbf{(3.5)}$$

$$FCS = FCS^{PC} \cdot f_4\left(T_{db,amb}, T_{wb,ret}, T_{db,ret}\right), \qquad \textbf{(3.6)}$$

$$COP_{frio} = COP^{PC}_{frío} \cdot f_5\left(T_{db,amb}, T_{wb,ret}\right), \qquad \textbf{(3.7)}$$

where

PC	Full load
wb	Humid bulb
db	Dry bulb
amb	(Exterior) atmosphere
ret	Conditions at entrance to the interior battery (in the Zoning system, after mixing with the current from the by-pass)
COP	Coefficient of performance
FCS	Sensitive heat factor (sensitive heat / total heat)

As explained in §3.2.3 and §3.3.3, we know that the programme gradually adjusts the sensitive contribution to the return current until it achieves the conditions of comfort in the controlled zones or until it reaches the limit value for the sensitive load (full load). Once the sensitive contribution value is set in an iteration, the drive conditions to the room are calculated with the following expressions

HEAT MODE

$$T_{imp} = T_{ret} + \frac{Q_{s,heat}}{\dot{m}_{ret} c_{p,air}} \quad , \qquad \textbf{(3.8)}$$

$$W_{imp} = W_{ret} \quad , \qquad \textbf{(3.9)}$$

COLD MODE

$$T_{imp} = T_{ret} - \frac{Q_{s,cold}}{\dot{m}_{ret} c_{p,air}} \quad , \qquad \textbf{(3.10)}$$

$$H_{imp} = H_{ret} - \frac{Q_{s,cold}}{FCS \cdot \dot{m}_{ret}} \quad , \qquad \textbf{(3.11)}$$

being $Q_{s,heat}$ or $Q_{s,cold}$ the sensitive power contributed by the equipment, m_{ret} the return mass flow, W the absolute humidity and H the enthalpy.

As far as calculating the electrical power requirement is concerned, one is tempted to write

$$PE = \frac{Q_{total}}{COP^{PC}} \quad , \qquad \textbf{(3.12)}$$

but this is not entirely correct due to the fact that the machine experiences changes in its characteristics at part load. We shall deal with the matter of electrical consumption depending on if it is an all-nothing machine or an inverter.

3.4.1 *Calculating electrical consumption in ALL-NOTHING machines*

These machines cover the part load by turning the compressor on and off (*cycling*).

Let us assume that during a time passage of one hour the machine demands half of the energy that it is able to supply at full power under the operational conditions of the passage. Initially, to meet this demand the machine has to operate at full load for a total of thirty minutes. With calculation passages of one hour, there is insufficient resolution to determine details of how the equipment operates (if it works for 30 minutes in a row, if it works in two 15 minute runs, etc.). But in any case, we know that it goes through start and stop cycles. As a result of these cycles, the equipment suffers a loss of efficiency since part of the electrical energy consumed is channelled into heating or cooling its own mass. Thus, the real operational period is slightly greater than that predicted for thirty minutes and, consequently, electrical consumption also increases.

Let us define the part load ratio as the quotient between the sensitive load required by the equipment during the time passage and the maximum sensitive power (at full load) that the equipment is able to supply

$$plr = \frac{\text{Sensitive load demanded}}{\text{Maximum sensitive power available}} \qquad \textbf{(3.13)}$$

Physically, (3.13) is the fraction of time passage during which the machine has to continue working (half an hour, in the previous example).

Loss of efficiency in the equipment at part load can be characterized by up to third grade polynomic functions.

$$DEG = a + b \cdot plr + c \cdot (plr)^2 + d \cdot (plr)^3 \qquad \textbf{(3.14)}$$

a, b, c and *d* being coefficients obtained experimentally, depending on the type of equipment and its quality. For domestic direct expansion equipment, [Henderson] proposes various correlations, from which the following has been selected:

$$DEG = 0.000352822 + 1.19199 \cdot plr - 0.246716 \cdot (plr)^2 + 0.0546566 \cdot (plr)^3 \ , \qquad \textbf{(3.15)}$$

Correction (3.15) is applied for consumption in the compressor and ventilator in the exterior unit. Thus, the electrical consumption is calculated as:

$$P = plr \cdot P_{vi} + DEG \cdot P^{pc}_{compr+speed} \ , \qquad \textbf{(3.16)}$$

where

P	Electrical power consumed
P_{vi}	Electrical power consumed by the interior ventilator
$Q_{sensitive}$	Sensitive load demanded during time passage
Q^{pc}_{total}	Total thermal power that the equipment is able to supply at full load under conditions of evaporation and condensation during time passage
$P^{pc}_{compr+ve}$	Electrical power consumed at full load by the compressor and exterior ventilator of the equipment

The first addend of (3.16) ponders the electrical power consumed by the ventilator in the interior unit by the fraction of time passage during which it operates. The second term provides the total power consumed by the exterior unit, taking into account the loss of efficiency (*DEG>plr*).

3.4.2 *Calculation of electrical consumption in INVERTER machines*

This type of equipment regulates the rotation speed of the compressor to adjust thermal production at part load. In this way, they achieve a more continuous operation, avoiding loss due to start-ups and stops. Since the mass flow of coolant may not be made arbitrarily small, there is a minimum speed for the compressor below which the equipment operates as a conventional all-nothing system. In this manner, two regions of operation are differentiated between: inverter and all-nothing.

Reduction in coolant flow makes the machine exchangers oversized, which leads to a very important improvement in the COP at part load (Figure 3-8). In the all-nothing zone, there is a progressive loss, equation (3.14), although based on the transition point COP, which is greater than the full load COP.

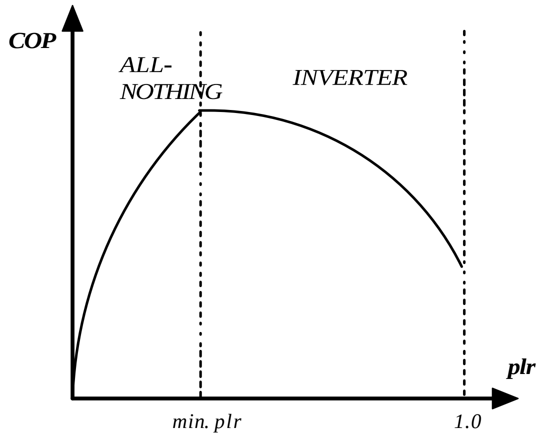


Figure 3-8: Evolution of the COP at part load

Little, scarcely verified information is available on the modelling for these machines. All references found are very recent [Bettanini], [Chino], [Gottfried], which supports the need for research in this area. With the lack of experimental measurements to validate any model, the following two, described below, have been tested. Stating that the first model forecasts modest electrical consumption rates (§6) while the second, which assumes fairly restrictive hypotheses, leads to electrical consumptions higher than those for the all-nothing model §3.4.1, which is a debatable point. This situation can only be solved by an experimental study.

3.4.2.1 *Inverter-1 model*

The manufacturer’s catalogue characterizes part load under nominal conditions (exterior 35ºC; interior 19ºC dry and 27ºC humid) by curves, such as in Figure 3.9 [Daikin]. The upper curve represents the thermal power given and the lower represents total electrical consumption.

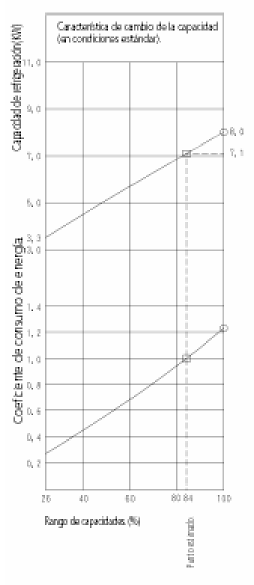


Figure 3-9: Performance at part load in the inverter zone

Characteristic of change of capacity

(In standard conditions)

Coefficient of energy consumption

Capacity range

How these functions vary beyond the nominal condition is unknown. The model assumes that they do not change. The calculation procedure is as follows:

1. With the condensation and evaporation temperatures, thermal and electrical powers at full load are determined (table in the catalogue).
2. With these powers, the part load curves are escalated, i.e., a change of variable is performed so that the maximum number of curves 3-9 is the full load value (thus obtaining the curves for the non-nominal conditions).
3. 3-9 is accessed with the sensitive power demanded. Based on the thermal capacity curve, the range of capacities is calculated (abscissa in Figure 3-9) and, with this value, it is read on the electrical consumption curve.

The machine operates in all-nothing zone when the sensitive power demanded is less than the minimum on the capacity curve. In this case, model §3.4.1 is applied.

3.4.2.1 *Inverter-2 model*

This does not use the part load curves in the catalogue, nor does it consider improvement in the COP at part load. The abscissa of curves 3-9 is interpreted as the part load factor.

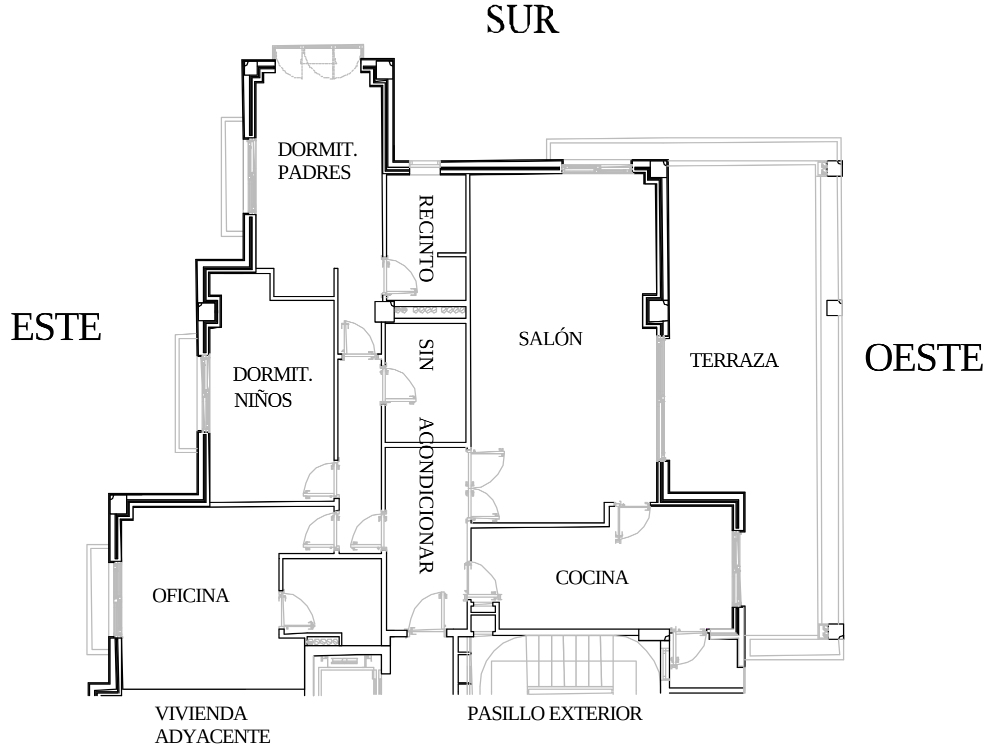
The following is performed for each zone:

1. INVERTER ZONE: here, correction (3.14) is not applied as there is no loss due to starts and stops. The full load COP is used and it is assumed that the ventilator operates during the time passage;
2. ALL-NOTHING ZONE: calculated with model §3.4.1.

FOURTH CHAPTER
Model house and operational conditions

4.1 Definition of the model house in TRNSYS

This is a single-family house (see Figure 4-1 for ground floor plan), designed to be representative of the flats in which ducted acclimatization systems are installed. It is located in the intermediate floor of a building, with offset terraces, the exterior walls being orientated east-south-west.



SOUTH
EAST
WEST
CHILDREN'S ROOM
OFFICE
ADJACENT HOUSE
LIVING ROOM
ENCLOSURE
NON-CONDITIONED
LIVING ROOM
KITCHEN
EXTERIOR PASSAGEWAY
TERRACE

Figure 4-1: Ground floor plan of the house

In order to model the house, TYPE56 is used in TRNSYS, which, as explained in the second part of this document, uses a non-geometrical balance method for multi-zone buildings.

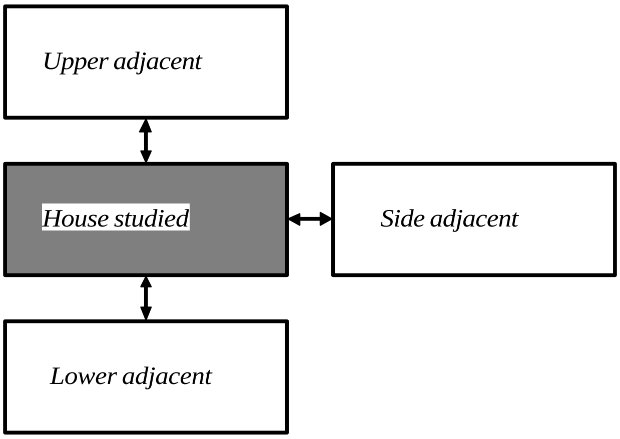
4.1.1 Zoning

The first phase in defining a building in the TYPE56 entails deciding on the number of thermal zones in which it is to be divided. In our case, to size the duct network supplying each acclimatized room and to be able to reproduce the evolution of the thermohygronometric conditions in each zone when explicitly simulating the conditioning equipment, we divide the house into five acclimatized zones.

- Living room
- Kitchen
- Office
- Parents' bedroom
- Children's bedroom

Two non-acclimatized zones have been added to these (in free temperature evolution, §2.5) in order to obtain good estimates for the temperature in:

- the central part of the house, including two bathrooms, passageway and hall
- the adjacent houses: above, below and on the side (Figure 4-2).



The interior conditions in the upper and lower adjacent houses exert a considerable influence on the load value in the house studied, due to the extension of the contact area via upper and lower forging. To determine the temperature in these enclosures, three possible set-ups have been explored:

- same temperature on both sides of the forging, i.e., acclimatized adjacent houses with the same usage model as the house studied (*identical temperature profile conditio*n)
- adjacent houses with an average room temperature and the temperature of the house studied (*average temperature profile condition*)
- Adjacent houses modelled as a non-acclimatized zone, with no direct solar gain (blinds down) and unoccupied (*zone model*).

As shown in Figures 4-3 to 4-6, the third option leads to intermediate sensitive load values between the first, not viable due to variability and unpredictability of use of the houses, and the second option, which forecasts fairly high loads, especially for heating.

In the summer months, the third model forecasts cooling loads slightly higher than in the second, since the intense average temperature fluctuations between the setting and room temperature (2nd model) give an average temperature for the adjacent flats that is lower than the pre-set temperature for the zone model, which has a much gentler evolution due to the inertia of the building.

Finally, the third option was taken. This assumes a compromise between the first very optimistic option as it assumes that the adjacent houses are acclimatized, and the second, which is excessively conservative.

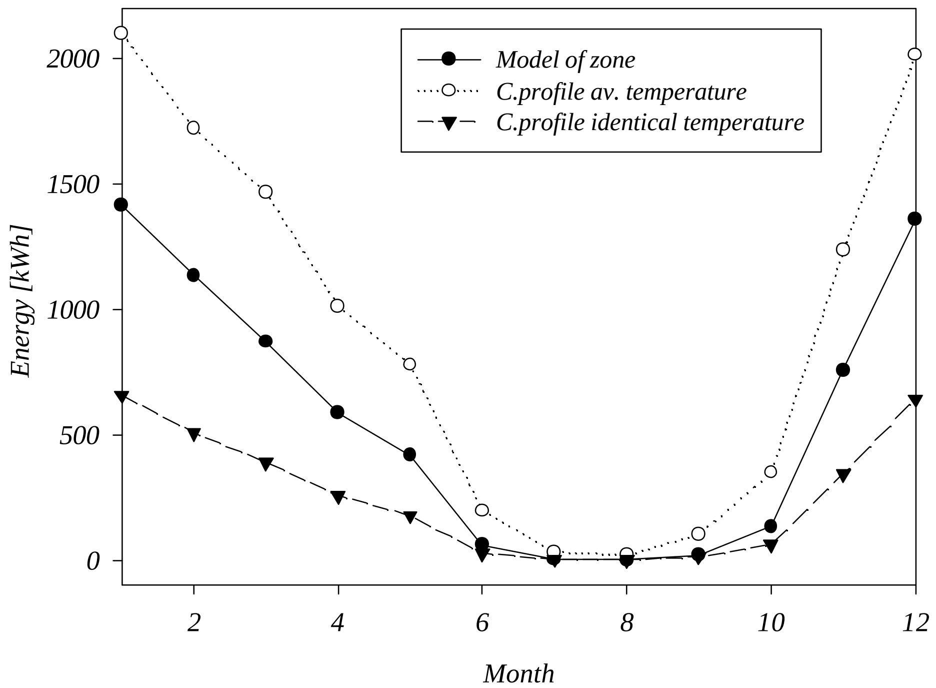


***Figure 4-3**: Monthly heat energy demand for each possible model of adjacent housing in Malaga (in a typical year).*

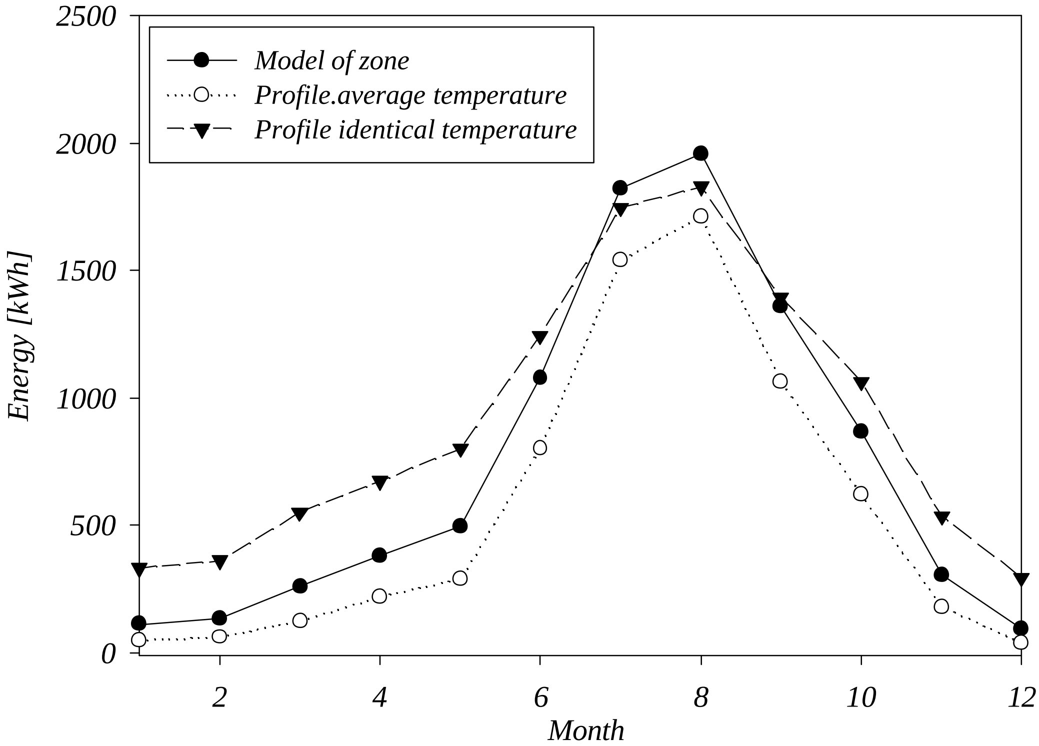


Figure 4-4: *Monthly cooling energy demand for each possible model of adjacent housing in Malaga (in a typical year).*

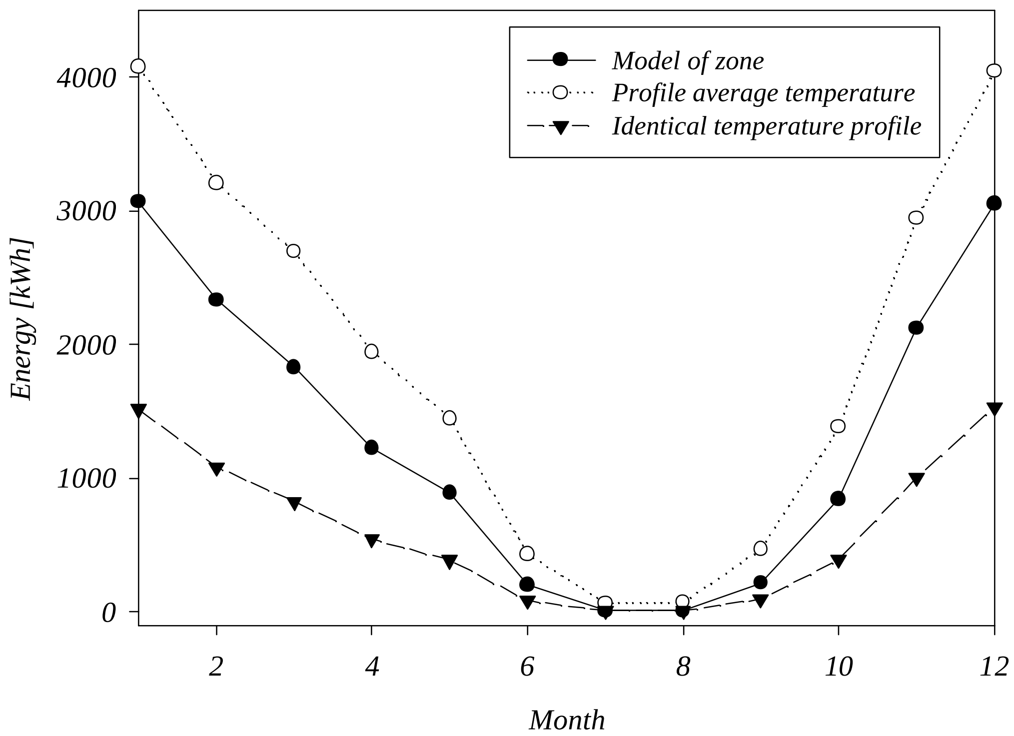


Figure 4-5: *Monthly energy demand for each possible model of adjacent housing in Madrid (in a typical year).*

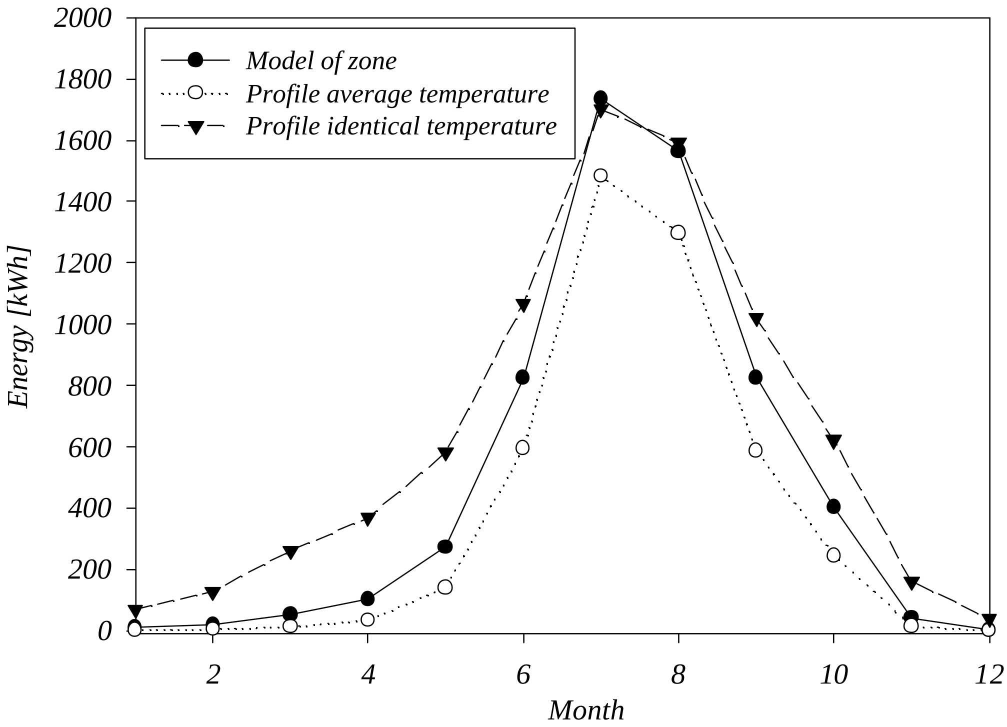


Figure 4-6: *Monthly cooling energy demand for each possible model of adjacent housing in Madrid (in a typical year).*

4.1.2. *Physical description of the zones – opaque and semi-transparent seals*

In order to define a zone in TYPE 56, it is sufficient to determine the equivalent thermal capacity of air that it contains and the area and the thermal and optical performances of each of the seals confining it, as well as the *relation* that these have with their surroundings, i.e., if it is an exterior seal, a separating seal between adjacent zones, with a profile condition imposed on it or inside the zone. As explained in §2.3, the model overlooks further details to do with the geometry of the enclosure as it is an approximate way of sharing out radiation and radiation interchanges.

The thermal properties of a multi-layer wall are determined by its constituent layers and, if possible, by the resistance between consecutive layers. So, in the TYPE56 data structure, firstly, the layers are defined. These are then assembled in order and with their thicknesses to form the walls.

When a layer has little thermal inertia, as in the case of air chambers, instead of defining its conductivity, thermal capacity and density, its equivalent thermal resistance is defined (thickness/conductivity) since ????? it may be considered that the drive system in force in it is stationary.

Table 4-7 shows the properties of the materials used, taken from the reference [Pinazo].

MATERIAL IN THE LAYER	CONDUCTIVITIVITY [kJ/hr m K]	CAPACITY [kJ/kg K]	DENSITY [kg/m³]	RESISTENCE [hr m² K/kJ]
Plaster	1.08	0.92	800	—
Simple air brick	1.764	0.92	1200	—
Air chamber (exterior walls)	—	—	—	0.05
Rockwool	0.1512	0.84	30	—
½ ft. of perforated brick	2.736	1.135	1600	—
Cement mortar	5.04	1.05	2000	—
Concrete arch	5.544	1.05	1168	—
Paving	3.96	1.38	2000	—
Door agglomerate	0.022	2.72	650	—

Table 4-7: Thermal properties of the layers

The walls forming the house have the following structure:

EXTERIOR WALLS

1. Plaster (thickness = 0.015 m)
2. Simple air brick partitioning (thickness = 0.09 m)
3. Air chamber
4. Rockwool (thickness = 0.04 m)
5. ½ ft. perforated brick (thickness = 0.012 m)
6. Rough-coated cement mortar (thickness = 0.015 m)

INTERIOR WALLS

1. Plaster redressing (thickness = 0.015 m)
2. Simple air brick (thickness = 0.07 m)
3. Plaster redressing (thickness = 0.015 m)

FORGING

1. Paving (thickness = 0.05 m)
2. Normal concrete arch (thickness = 0.31 m)
3. Paving (thickness = 0.05 m)

DOORS

1. Agglomerate (thickness = 0.035)

Once the wall composition is known, the next step is to define each zone that borders on it, in the area of each and their respective types (exterior, partitioning …). This task is repetitive and lacking any interest. The details may be consulted in Annex 1 of this document, with the complete definition file on the building.

Defining the windows is somewhat more complex than for the walls. The windows in this house are double glazed, model Climalit Planilux 4/6/4 (4 mm thick glazing with a 6 mm air chamber) with 15% of the surface being taken up by the aluminium frame. Based on the properties appearing in the Glazing Manual and with the Windows 5.1 programme LBNL [Window Manual], all the data need by TYPE56 can be determined to characterize a window (coefficients of adjustment for the thermal bridge, angular dependence on absorption factor, long-term emissivity, etc.). This programme generates a text file with this information (Annex 2), which is later introduced into a windows library where the TYPE56 can read them.

It should be noted that the optical properties of glass, noted in [Glazing Manual] derive from tests for 30º angles of incidence, whereas those required by the TYPE56 are for tests with normal incidence (0º). To convert from one to the other, use the following basic optic relations:

SNELL'S LAW

$$\frac{\sin(q_1)}{\sin(q_2)} = \frac{n_2}{n_1} \qquad \textbf{(4.1)}$$

FRESNEL'S LAW

$$r_{12}(q_1, q_2) = \frac{1}{2}\left[\left(\frac{n_1 \cos q_2 - n_2 \cos q_1}{n_1 \cos q_2 + n_2 \cos q_1}\right)^2 + \left(\frac{n_1 \cos q_1 - n_2 \cos q_2}{n_1 \cos q_1 + n_2 \cos q_2}\right)^2\right], \qquad \textbf{(4.2)}$$

TOTAL ABSORPTION FACTOR OF GLASS SHEET

$$A(q) = \frac{(1 - r(q)) \cdot (1 - t(q))}{1 - r(q)t(q)}, \qquad \textbf{(4.3)}$$

TOTAL REFLECTIVITY

$$R(q) = r(q) \cdot \frac{1 - r(q)}{1 + r(q)} \cdot \frac{1 - r^2(q)}{1 - r^2(q)t^2(q)}, \qquad \textbf{(4.4)}$$

TRANSMISSIVENESS

$$t\left(q\right)=\exp\left(\frac{-k\cdot L}{\cos\left(q\right)}\right)\ , \qquad \textbf{(4.5)}$$

?$_1$ being the angle of incidence and *?*$_2$ the refracted angle.

The problem entails determining the coefficient of extinction of glass based on the data available (A;R;T=1-A-R), which are such for an angle of incidence of 30º. From (4.1) we obtain the angle of the refracted beam (n_1 = 1, n_2 =1.52, ?$_1$ = 30º), with which (4.2) is accessed to calculate the reflectivity of the interphase. From (4.3), the transmissiveness of the glass is cleared for the angle of incidence, a value with which, finally, the product kL of the equation (4.5) is obtained. Once the extinction factor is known, the optical properties can be solved for the new angle of incidence (?$_1$ = 0º) simply by substituting in the previous expressions.

4.1.3 *Conditions of use*

As far as internal gains are concerned, their manifestation is determined by the usage profiles shown in Table 4-8, with the following format for the data cells: Number of people / Television / Computer / Domestic Appliances.

Time \ Zone	LIVING ROOM	KITCHEN	OFFICE	PARENTS' BEDROOM	CHILDREN'S BEDROOM
0:00 - 7:00	0	0	0	2	2
7:00 - 7:45	0	2 / E	0	1	1
7:45 - 17:00	2	2 / E	0	1	0
17:00 - 20:00	2 / T	1 / E	1 / O	1	0
20:00 - 20:30	0	4	0	0	0
20:30 - 23:00	3 / T	0	0	0	1 / O
23:00 - 0:00	0	0	0	2	2

Table 4-8: Usage profiles of the zones.

Lighting control has depended on the radiation value on a horizontal surface, using a controller for the purpose. When its value is less than 250W/ m^2 the lights in the occupied zones turn on (except when people are sleeping). This value provides hours of light turned on and off that are reasonable throughout the year.

Generating power linked to each concept (lighting, equipment, etc.) is taking from the ISO 7730 Standard and can be consulted in the list in Annex 1.

As regards mass flows of infiltration and interchange between zones, this calculation requires solving the aeraulic problem, which is not carried out by this building model and which, in any case, is difficult to solve due to the lack of data. A frequent approximation involves assuming that the first gives a constant amount of renewals on the hour (here set at 0.6) and disregards the second.

4.2 Profile conditions

The profile conditions refer to values imposed on the border of the dominion for the case of the surfaces demarcating the building. In the exterior seals, the issue here is the meteorological variables. In §4.1.1, the first two options to model the adjacent flats were profile conditions, whereas in the third, which was not selected, this is not the case.

In the problem, the air temperature in the exterior passageway where the common stairs for the building are still needs solving (Figure 4-1). This is to be performed in terms of a profile condition calculated as an average between the room temperature and the pondered temperature (depending on the relation of areas of the interior seals) in the areas leading into it,

$$T_{passageway} = \frac{T_{room} + l_1 T_{kitchen} + l_2 T_{\sin cond}}{2} . \qquad \textbf{(4.6)}$$

One critical aspect for the reliability of the simulation results is that of the meteorological conditions used, which typically have a format of a series of time data. These series can be assembled from real measurements [TMY Manual] or be generated from a short series of data using statistical procedures. In Spain, with the exception of some towns and cities, no sufficiently historic meteorological data are available, so that we have generated synthetic series, using the CALENER programme [Technical Code] or have used the database for Spain and Portugal known as "TRNSCLIMA" by Aiguasol [TRNSCLIMA].

For simulations in this research, the TRNSCLIMA database has been applied, which provides three files on each town or city.

- Typical year (TY)
- Extremely hot and sunny year (HI)
- Extremely cold and cloudy year (LO)

Each file includes eight columns and eight thousand six hundred and sixty files, one for each hour in the year. In order, the columns contain the following information:

1. Month
2. Hour
3. Normal direct irradiation
4. Total horizontal irradiation
5. Dry bulb temperature
6. Absolute humidity
7. Wind intensity
8. Wind direction

The extreme years are used to dimension the equipment (for further comment, see §5.1), whereas the typical years are used for the other simulations to calculate typical electrical consumption and interior conditions (§6). Note that the typical year forecasts consumption rates that are very similar to those obtained with the equivalent CALENER meteorological file.

As far as the towns and cities studied are concerned, the following figures give an idea of the meteorological conditions characteristics of the typical year.

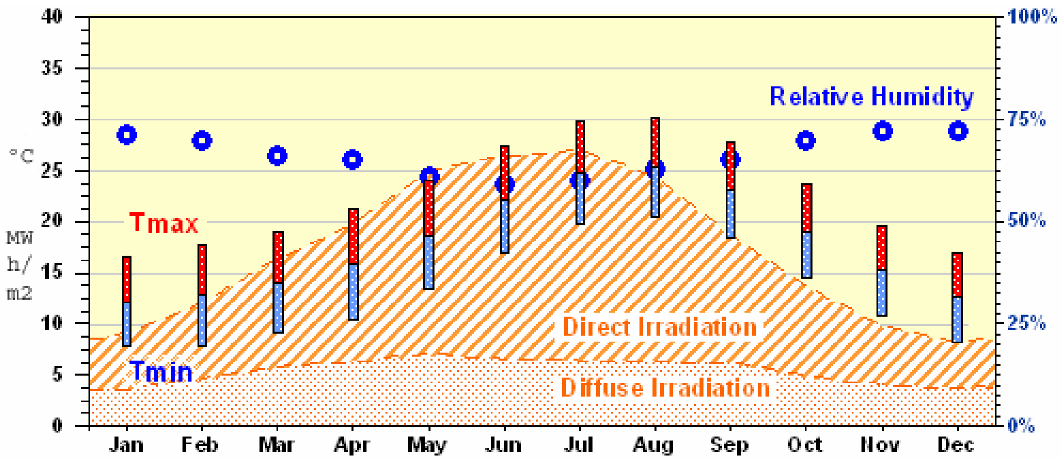


Figure 4-9: Typical meteorological year for Malaga [Aiguasol]

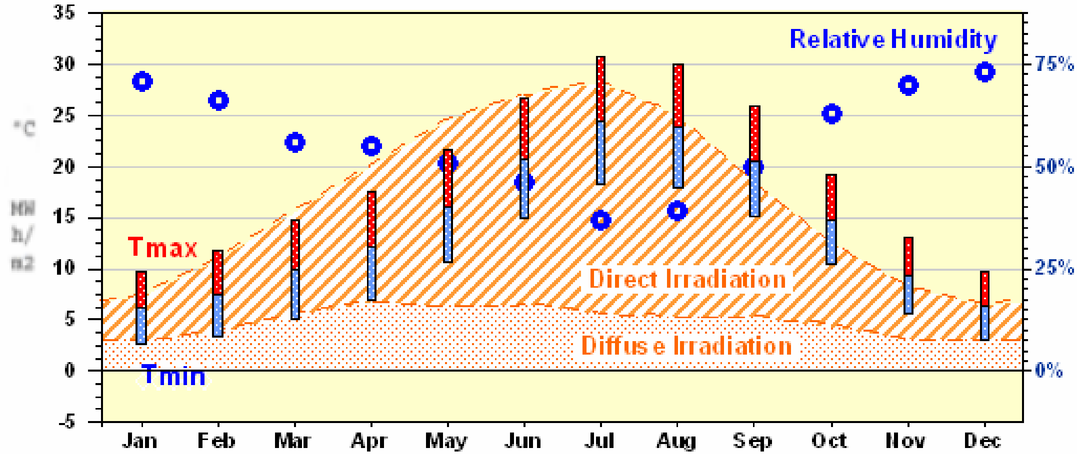


Figure 4-10: Typical meteorological year for Madrid [Aiguasol]

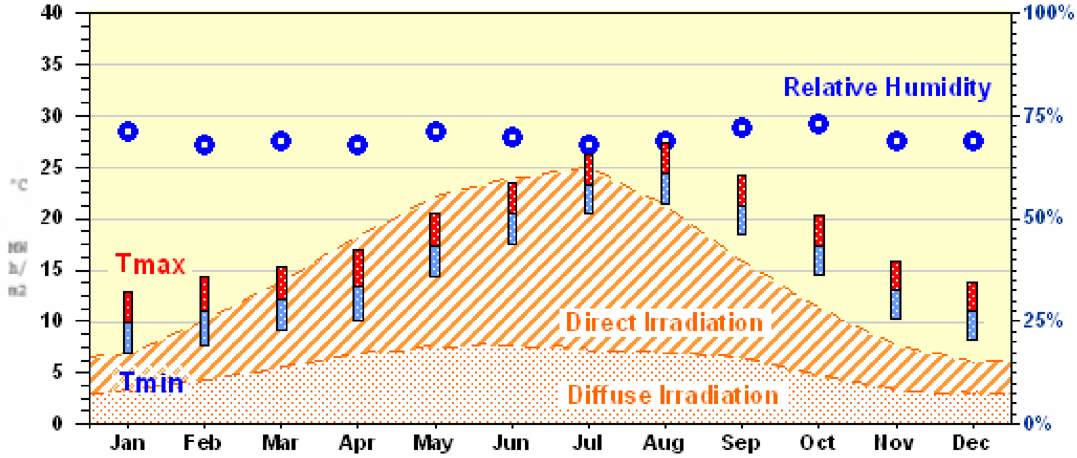


Figure 4-11: Typical meteorological year for Barcelona [Aiguasol]

FIFTH CHAPTER

Dimensioning the systems

5.1 Introduction

The procedure widely used to dimension the production equipment of an acclimatization system involves determining the peak load in the equipment, the sum of the peak load for the building and additional loads only affecting the equipment (ventilation, losses of heat in the distribution network, etc.).

Calculating loads is useful at the design stage as it idealizes the HVAC systems, both the primary (heat pumps, etc.) and the secondary (acclimatization system, etc.). From the first, it is assumed that they can supply unlimited power, whereas in the case of the second, the compensate, in a precise and instantaneous manner, heat and mass flows affecting the air in each zone which, in this way, maintain their temperature and humidity setting conditions. Thus, the real dynamics of the HVAC systems are replaced by their desired effect (certain conditions for the air) so that the load calculation only requires information on the building. Once the load for each zone is known, depending on which acclimatization system is involved (Zoning, non-Zoning …), the net load demanded by the building for the equipment is calculated. Adding the loads for ventilation, return for lights, etc., the peak power is finally obtained that should counteract the production equipment.

The physical complexity of buildings has led to the appearance of numerous methods for calculating peak load and thermal demand, which are initially manual and, nowadays, computational. The problem lies in the fact that a good estimate of load calls for taking into consideration the thermal inertia of the building’s structure, which delays and attenuates the manifestation of some instantaneous gains such as sensitive load in the air. By way of an example, let us imagine a case in which maximum room temperature and solar gain coming in through the window coincide in time. While the increase in room temperature instantaneously modifies loads through ventilation and infiltration, the thermal wave caused in the mass exterior seals (the building envelope), it will take time to reach the air in the zones. On the other hand, radiation does not directly affect the air, but rather should first heat the surfaces on which it falls to be subsequently convected. Thanks to these dynamic processes, the heat flows that exterior demands induce in the air are not simultaneous (they go out of phase) and their joint instantaneous value is less than would be the case in a stationary situation. The balance methods (§2) implicitly take all these phenomena into consideration, for which reason they are a good tool for dimensioning.

Comments are made on the methodology and calculation conditions used in §5.2. Then the production equipment and nominal mass flows are dimensioned for the different acclimatization systems: non-Zoning (§5.3), Zoning with no equipment power reduction (§5.4) and Zoning with equipment power reduction (§5.5).

5.2 Methodology and calculation conditions

The multi-zone building model TYPE56 in TRNSYS (§2), as with any other balance method, can solve questions of dimensioning and thermal demand calculation on the condition that they use suitable meteorological conditions. The difficulty lies in defining these conditions since the evolution of the climate in a given place is not entirely deterministic nor totally random, but rather it is irregular over time in various scales: hourly, daily, monthly, annually and interannually [Knight]. Also, the variables of interest (dry room temperature, humid temperature, solar radiation, wind speed and direction) have interlaced relations, some being more closely knit than others.

Initially, the problem is solved by having an extensive meteorological register (25 to 30 years of historical data) with which to simulate the building (Diagram 5-1). For each year, a different peak load and thermal demands are obtained, the maximum peak being the one which conditions the size of the equipment (Figure 5-2). For reasons of economy and efficiency, in conventional acclimatization applications, dimensioning is not performed with the absolute peak, but rather with a certain percentile, which in this calculation procedure, will be applied to the load distribution obtained and not to the exterior conditions.

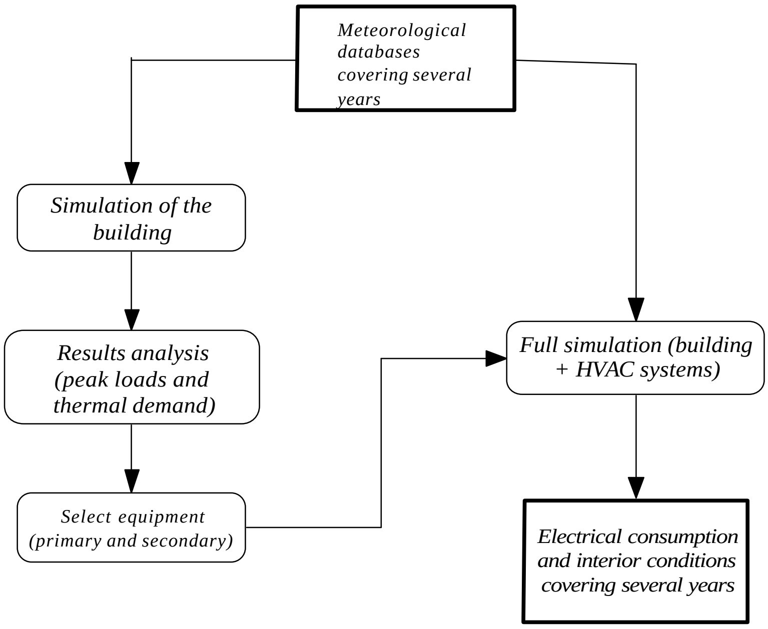


Figure 5-1: Steps for dimensioning and calculating the performance characteristics of an HVAC system when detailed meteorological registers are available. Adapted from [Hui].

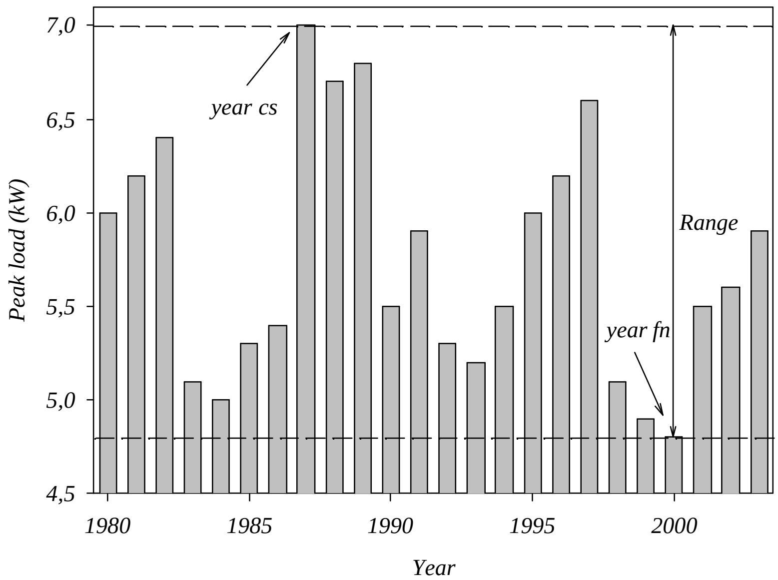


Figure 5-2: Maximum thermal cooling power demanded in each year for a hypothetical case. The "year cs" is the hottest year in the place and sets the peak cooling load. The "year fn" is the coldest and cloudiest year that sets the maximum heating loads (not shown) and minimum cooling loads.

This procedure, although abundant in results, nowadays has two basic drawbacks:

- The detailed registers are only available for certain towns and cities and, occasionally, lack data on radiation.
- The computational cost is extremely high.

These reasons, for the time being, make this type of simulation in general unviable. Alternatively, problems involving dimensioning and calculation of thermal demand can be separated, by using at least two series of data:

- CALCULATING DEMAND: a typical year is calculated with the meteorological conditions deemed to be characteristic or *average* for the place in question over a long period of time (25 to 30 years). Using this year in the simulation, the average thermal demand for a period covering the active life of the equipment will be obtained. We should be aware that the demand obtained in this manner will not necessarily tally with the demand for any particular year.

- DIMENSIONING: an extremely hotter, sunnier year than usual and an extremely colder, cloudier year than usual are defined (Figure 5-2), or alternatively, an extreme day in winter and another in summer. In this second case, the building is set for the extreme day system and is dimensioned with the resulting peak.

The diagram in Figure 5-1 now fragments into two parts, as shown in Diagram 5-3.

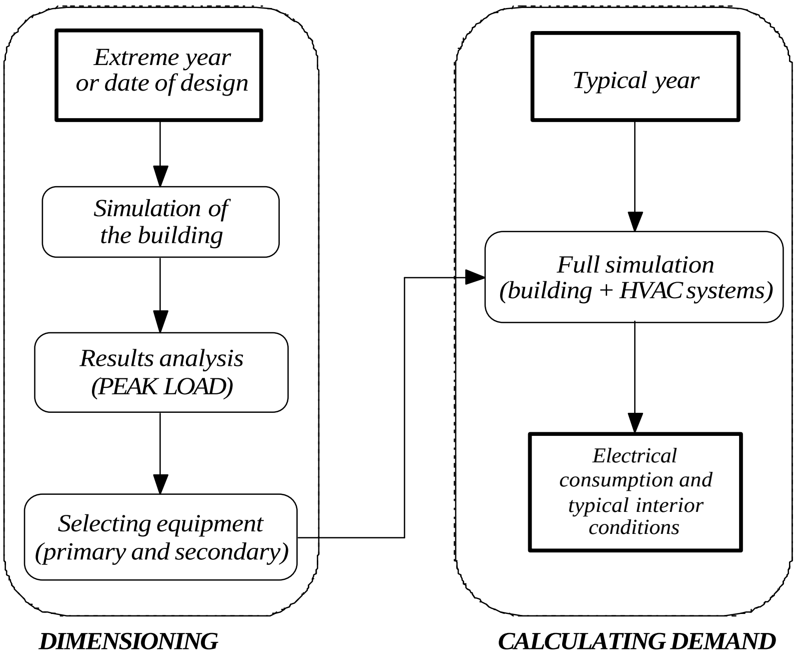


Figure 5-3: Phases in dimensioning and calculating demand

Another use of extreme years is that they make it possible to note thermal heating and cooling demands.

Since simulation methods in Spain are relatively unknown, at present there are no commonly accepted practices for any of the points mentioned above. In countries with more tradition and higher resolution meteorological registers over space and time, such as the United States, typical years are usually assembled from selected months in real year [TMY2 Manual]. In Spain, this is only possible for small provincial capitals. For this reason, options included using synthetic series, i.e., drawn up with statistical methods based on average monthly values, maximum and minimum values, conditioning factors such as if the place lies on the coast or inland, etc.. This information can be used to generate time series for all the variables of interest (dry temperature, humidity, total horizontal radiation and normal direct radiation, wind speed and direction). Several databases generated in this manner are available on Spain, such as the [Calener], [Meteonorm], [Aiguasol] or [AMT-A]. As noted in §4.2, in this research, the TRNSCLIMA database has been used, both for dimensioning with the extreme years, §5.3, §5.4, §5.5) and for calculating thermal demand and electrical consumption (§6) with the typical year.

The system was set at 22ºC with 50% HR for September to April, and at 24ºC with 50% HR for the rest of the year. It is important that, at times of change of setting the transition should not be brusque in order to avoid false load peaks. It is sufficient to interpolate the setting temperature inter-linearly for a few hours around the point where the change occurs.

Figure 5-4 shows an assembly in TRNSYS for calculating loads, providing a brief explanation of the components involved. Simulation time passage has been set at 15 minutes to be able to record the changes experienced in the occupancy profile shown in Table 4-8. Two simulations are performed for each town or city, one for each extreme year. A file is obtained from this with the sensitive and latent powers demanded by each acclimatized zone and with air temperatures in the non-acclimatized zones (interior part of the house and adjacent flats, §4.1.1) for each time passage. With this information, the equipment can now be dimensioned, taking into account the particular characteristics of the acclimatization system.

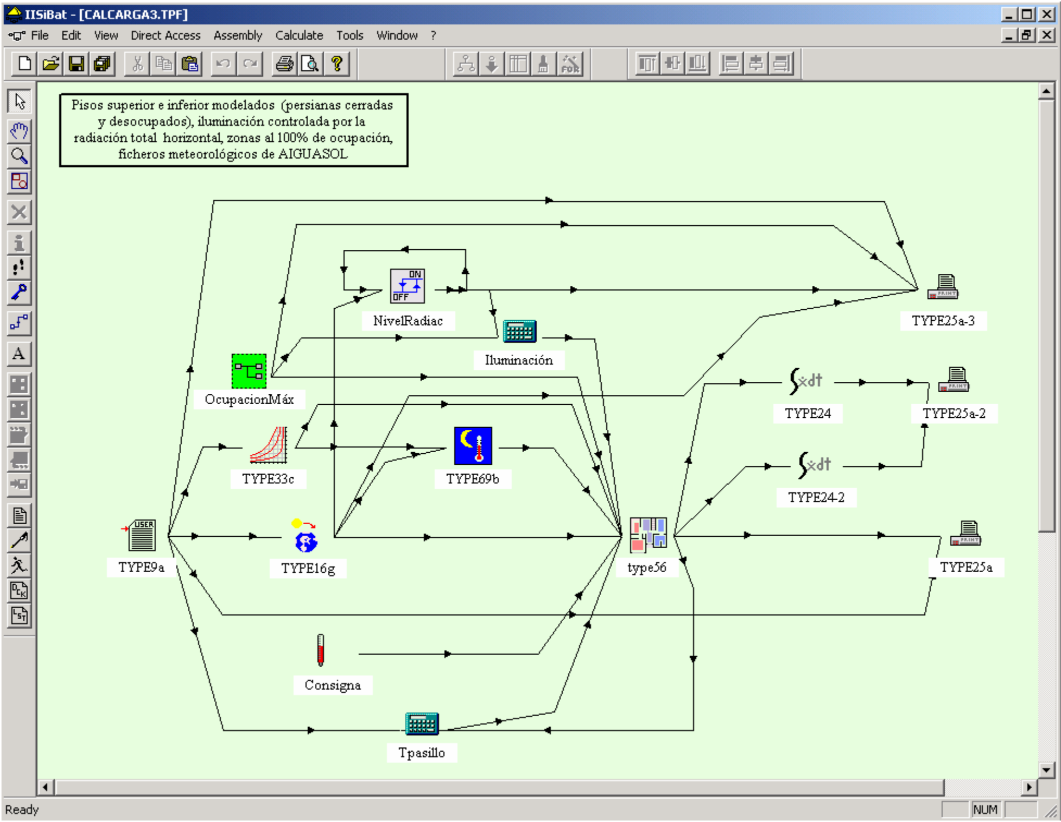


Figure 5-4: Assembly for calculating loads in a house. The building lies in the centre (TYPE56), to the right, the components that process the profile conditions and use of the same, and to the left, the components that process and write the outlets to text files.

5.3 Non-Zoning system

Due to the fact that the distribution network does not allow for dealing with the needs of each zone separately, in order to ensure being able to cover the peak load in all zones, the nominal power of the machine (*PNM*) should be taken to be equal to or higher than the sum of the peak loads in the zones, even though they are not simultaneous

$$PNM = max(PNM_{COLD}, PNM_{HEAT}) = max\left(\sum_{i=1}^{N\ zones} max(Q^{i}_{heat}(t)),\ \sum_{i=1}^{N\ zones} max(Q^{i}_{cold}(t)) \right)$$

(5.1)

Where *Nzones* is the number of zones, $Q_i^{heat}(t)$ the time series of heat loads for the zone *i* and $Q_i^{cold}(t)$ the time series for cooling loads for the zone i. In each zone, the maximum possible internal loads should be taken into consideration.

Generally, the maximum simultaneous load will not be equal to the sum of the peaks, so that the machine will always be working at part load; although this is not the only reason for making the mistake of overdimensioning the same. Let us assume that the sensitive maximum cold load for the controlled zone (k) occurs at moment t. As the mass flow driven to the zones is a constant, coverage of this point automatically imposes the drive temperature value required by the machine.

$$T_{imp}(t) = T_{set} - \frac{Q_{cold}^{k}(t)}{\dot{m}_k c_p}, \quad \textbf{(5.2)}$$

and, as a result, the power at which it operates,

$$Q_{machine}(t) = \frac{Q_{cold}^{k}(t)\,\dot{m}_t}{\dot{m}_k}. \quad \textbf{(5.3)}$$

The power injected into any other zone (j) depends on the total fraction of mass flow that it receives,

$$Q_{cold}^{j}(t) = Q_{machine}(t)\frac{\dot{m}_j}{\dot{m}_t}, \quad \textbf{(5.4)}$$

that will generally be adjusted to its requirements. If at that moment t zone j presents a greater load, it will not be met, it being overheated and thus incurring in improper use of the possibilities of the primary machine.

Since the evolution of the system is governed by what happens in the controlled zone, we can consider fitting a lower capacity machine capable of covering the real demand at the moment when maximum load occurs in the controlled zone. Nevertheless, this should not be done as we should not overlook the fact that when the user feels uncomfortable in a non-controlled zone, he/she will change the temperature setting for the zone to achieve conditions of comfort wherever he/she is. On the other hand, the controlled zone could vary, for instance, in the case of a wireless thermostat or if the user decides to change its place.

To illustrate the calculation procedure, we explain the case for Malaga. Table 5-5 shows the heating and cooling peaks for Malaga, depending on whether the year is extreme (cold and cloudy, or warm and sunny). As was to be expected, the sum of the cold peaks in the warm, sunny year condition the dimensioning of the machine for Malaga (the simultaneous point is shown for information purposes only).

	HEATING POINT [kW]		COOLING POINT [kW]	
	HOT YEAR	*COLD YEAR*	*HOT YEAR*	*COLD YEAR*
LIVING ROOM	1.28	1.47	2.56	2.28
KITCHEN	1.02	1.15	1.63	1.45
OFFICE	0.77	0.90	1.04	0.88
PARENTS'ROOM	0.79	0.93	1.58	1.38
CHILDREN'S ROOM	0.61	0.71	0.99	0.80
SUM OF POINTS	4.47 (49.71 W/m^2)	5.16 (57.39 W/m^2)	**7.80 (86.75 W/m^2)**	6.79 (75.52 W/m^2)
SIMULTANEOUS POINT	4.30	5.01	6.78	5.79

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-5: Absolute heating and cooling peaks for extreme years of cold and heat, for the house in Malaga, with the non-Zoning system

The table above shows the peak loads, i.e., the maximum absolutes of load distributions for each zone. In order to dimension the machine correctly, it is advisable to study the histogram of such series since, normally, the peak load and its nearby neighbours represent uncommon extreme values which, if appropriately biased, lead to a smaller sized machine that is more suited to the application, the only drawback being that it is unable to counteract the load for a few hours in summer. In this regard, note that the meteorological conditions used do not apply any bias (percentile of the UNE standard …). In this research, we applied the percentile concept to the load resulting from extreme climatic conditions, not to the exterior conditions themselves. Load depends on exterior conditions and on the *sensitivity* of the building to the different types of environmental excitation.

Throughout the year of extreme heat, the living room demands cooling (with varying intensity) for 6,590 hours. Figure 5-6 shows the histogram for cold loads for this zone (to display the rest, 2,170 hours of zero or heat power have been excluded), clearly showing how the extreme points are marginal. This same situation recurs in the other zones.

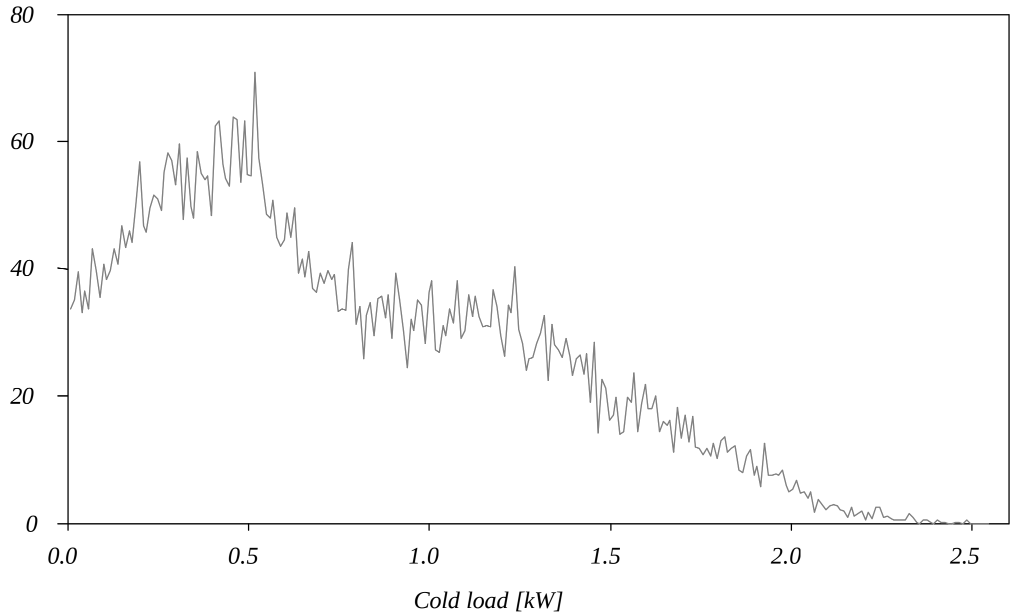


Figure 5-6: Histogram of cooling load for the living room, in Malaga, for a year of extreme cold and cloud. Number of hours vs. Cold load.

The series of cooling and heating load will be truncated, taking the percentile of 99% on an annual basis (Table 5-7).

	HEATING PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	HOT YEAR	*COLD YEAR*	*HOT YEAR*	*COLD YEAR*
LIVING ROOM	0.85	1.24	1.98	1.71
KITCHEN	0.69	0.97	1.23	1.05
OFFICE	0.55	0.79	0.79	0.63
PARENTS'ROOM	0.50	0.76	1.24	1.02
CHILDREN'S ROOM	0.38	0.58	0.77	0.63
SUM OF POINTS	2.97 (33.03 W/m^2)	4.34 (48.27 W/m^2)	**6.01 (66.84 W/m^2)**	5.04 (56.06 W/m^2)
SIMULTANEOUS POINT	2.90	4.23	5.28	4.45

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-7: Percentiles of heating and cooling for years of extreme cold and heat, for the house in Malaga, with the non-Zoning system.

The values in the table above represent the load in the building. The total load that the machine has to counteract is the sum of the load in the building with the load for ventilation, which in this case, makes up for the lack of return from the kitchen (to avoid odours in the rest of the house). Nevertheless, the load for ventilation depends on the mass flow driven to the kitchen, which in turn, depends on the machine selected. So the calculation starts by a machine covering the loads shown in Table 5-7. For this machine, the flow driven to each zone (Table 3) is calculated in accordance with the fractions of load in Table 5-7:

$$f_i = \frac{Q_i}{\sum Q_i} \ . \qquad \textbf{(5.5)}$$

Once the mass flow to the kitchen is known, the load for ventilation can be calculated, as follows

$$Q_{vent} = g\, \dot{m}_{kitchen} c_p \left| T_{kitchen} - T_{room} \right| , \qquad \textbf{(5.6)}$$

$m_{kitchen}$ being the nominal mass flow driven to the kitchen and g a variable with a value of 0 when the kitchen is unoccupied and 1 when it is occupied (Table 4-8). It is assumed that the drive to the kitchen can be withdrawn when it is unoccupied (extractor hood turned off).

Once the total load is known, checks are run to ensure that the machine initially selected is capable of meeting the demand. If it is not capable, the immediately higher one is selected and the calculations are repeated.

For the manufacturer DAIKIN, the combination RZP100DV1 (exterior unit) + FHYBP100BV1 (interior unit) supply a nominal sensitive power of 7.1 kW of cold and 12.8 kW of heat, so that it covers the load in the building, which has a demand of 6.01 kW in sensitivity (Table 5-7), whereas the superior version goes over this by a long way (9.1 kW). For this machine, whose interior unit moves 1,944 kgh^{-1} of air, Table 5-8 shows the nominal mass flow of drive to each zone.

CONCEPT	LIVING ROOM	KITCHEN	OFFICE	PARENTS' ROOM	CHILDREN'S ROOM	SUM
FRACTION	0.329	0.205	0.131	0.206	0.129	1.000
FLOW [kg/h]	639.576	398.520	254.664	400.464	250.776	1944.000

Table 5-8: Nominal mass flow to each zone, for Malaga, non-Zoning system

Once the flow to the kitchen is known, with the expression (5.6), the load for ventilation is calculated which, when added to the load in the zones, provides the total load in the machine (Table 5-9) for the dominant case of cooling.

LIVING ROOM [kW]	KITCHEN + VENTILAT. [kW]	OFFICE [kW]	PARENTS' ROOM [kW]	CHILDREN'S ROOM	SUM PEAKS [kW]
1.98	2.13	0.79	1.24	0.77	**6.91**

Table 5-9: Nominal mass flow to each zone, for Malaga

The machine selected is capable of overcoming the combined loads for the building and for ventilation, so that it is not necessary to repeat.

In Tables 5-5 and 5-7, what calls our attention is the similarity of the simultaneous peaks with the sums of peaks since the different orientation of seals in the house (east-south-west) produces asynchrony in the load for solar radiation. This fact means that the instant when the simultaneous maximum peak for cold occurs in the zones, it is basically determined by the room temperature, which is understand if we consider that the days surrounding any day in summer have similar radiation profiles (except for a cloudy day), room temperature being the factor that differentiates them. Figure 5-10 illustrates this circumstance for the day in which the maximum simultaneous peak occurs (31st August at 18:00).

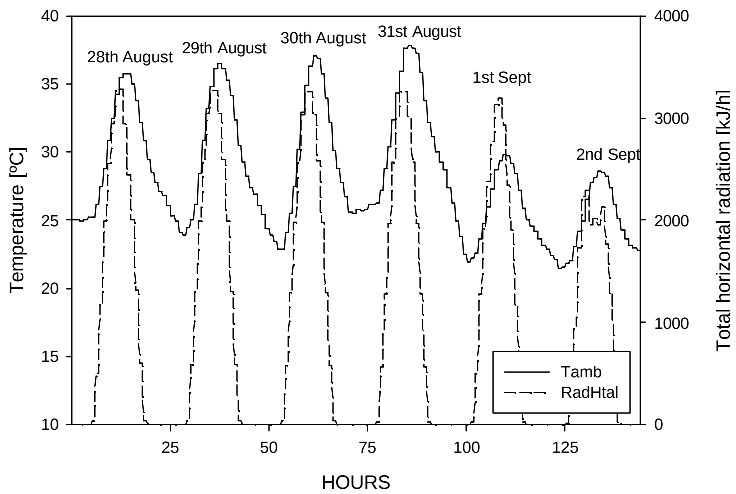


Figure 5-10: Evolution of room temperature and horizontal radiation around the day when the maximum cold peak occurs for Malaga

In heating, it is logical for the same to occur and, although more clearly (5.16 kW as opposed to 5.01), since the maximum demand occurs during the coldest night and almost simultaneously in all the zones.

In the case of the other towns and cities, it operates in like manner. The peak and percentile loads in the building are shown in Tables 5-11, 5-12 (Madrid; 5-13, 5-14 (Barcelona; 5-15, 5-16 (Valencia and 5-17, 5-18 (Seville).

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.63	1.99	2.39	2.14
KITCHEN	1.28	1.56	1.45	1.34
OFFICE	1.02	1.25	0.90	0.81
PARENTS' ROOM	1.00	1.28	1.39	1.15
CHILDREN'S ROOM	0.79	1.01	0.91	0.77
SUM OF PEAKS	5.72 (63.62 W/m^2)	**7.09 (78.86 W/m^2)**	7.04 (78.30 W/m^2)	6.21 (69.07 W/m^2)
SIMULTANEOUS PEAK	5.62	6.94	6.01	5.36

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-11: Absolute heating and cooling peaks for extreme cold and hot years, for the house in Madrid, with the Zoning system.

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.40	1.85	1.95	1.61
KITCHEN	1.10	1.42	1.20	0.97
OFFICE	0.90	1.18	0.74	0.56
PARENTS' ROOM	0.87	1.15	1.12	0.94
CHILDREN'S ROOM	0.68	0.92	0.75	0.57
SUM OF PEAKS	4.95 (55.05 W/m^2)	**6.52 (72.52 W/m^2)**	5.76 (64.06 W/m^2)	4.65 (51.72 W/m^2)
SIMULTANEOUS PEAK	4.85	6.43	5.00	4.04

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-12: Heating and cooling percentiles for extreme cold and hot years, for the house in Madrid, with the non-Zoning system.

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.31	1.79	2.47	2.13
KITCHEN	1.01	1.41	1.54	1.28
OFFICE	0.82	1.09	1.05	0.78
PARENTS' ROOM	0.83	1.09	1.41	1.25
CHILDREN'S ROOM	0.64	0.82	0.99	0.78
SUM OF PEAKS	4.61 (51.27 W/m^2)	6.20 (68.96 W/m^2)	**7.46 (82.97 W/m^2)**	6.22 (69.18 W/m^2)
SIMULTANEOUS PEAK	4.47	6.12	6.62	5.26

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-13: Absolute heating and cooling peaks for extreme cold and hot years, for the house in Barcelona, with the non-Zoning system.

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.06	1.40	1.85	1.49
KITCHEN	0.84	1.09	1.11	0.87
OFFICE	0.70	0.90	0.75	0.54
PARENTS' ROOM	0.64	0.87	1.13	0.90
CHILDREN'S ROOM	0.51	0.68	0.75	0.56
SUM OF PEAKS	3.75 (41.71 W/m^2)	4.94 (54.94 W/m^2)	**5.59 (62.17 W/m^2)**	4.36 (48.49 W/m^2)
SIMULTANEOUS PEAK	3.62	4.85	4.96	3.75

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-14: Heating and cooling percentiles for extreme cold and hot years, for the house in Barcelona, with the non-Zoning system.

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.12	1.62	2.45	2.26
KITCHEN	0.88	1.24	1.55	1.42
OFFICE	0.69	1.01	1.01	0.92
PARENTS' ROOM	0.66	1.02	1.59	1.29
CHILDREN'S ROOM	0.55	0.82	0.94	0.88
SUM OF PEAKS	3.90 (43.38 W/m^2)	5.71 (63.51 W/m^2)	**7.54 (83.86 W/m^2)**	6.77 (75.30 W/m^2)
SIMULTANEOUS PEAK	3.76	5.56	6.48	5.92

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-15: Absolute heating and cooling peaks for extreme cold and hot years, for the house in Valencia, with the non-Zoning system.

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.94	1.35	1.99	1.60
KITCHEN	0.75	1.05	1.22	0.96
OFFICE	0.60	0.89	0.79	0.59
PARENTS' ROOM	0.56	0.84	1.28	1.00
CHILDREN'S ROOM	0.42	0.66	0.78	0.60
SUM OF PEAKS	3.27 (36.37 W/m^2)	4.79 (53.27 W/m^2)	**6.06 (67.40 W/m^2)**	4.75 (52.83 W/m^2)
SIMULTANEOUS PEAK	3.16	4.61	5.28	4.17

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-16: Heating and cooling percentiles for extreme cold and hot years, for the house in Valencia, with the non-Zoning system.

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.29	1.67	2.65	2.34
KITCHEN	1.01	1.30	1.74	1.55
OFFICE	0.81	1.03	1.13	1.00
PARENTS' ROOM	0.83	1.02	1.64	1.32
CHILDREN'S ROOM	0.63	0.83	1.00	0.88
SUM OF PEAKS	4.57 (50.83 W/m^2)	5.85 (65.06 W/m^2)	**8.16 (90.76 W/m^2)**	7.07 (78.63 W/m^2)
SIMULTANEOUS PEAK	4.42	5.72	7.15	6.26

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-17: Absolute heating and cooling peaks for extreme cold and hot years, for the house in Seville, with the non-Zoning system.

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.05	1.41	2.18	1.98
KITCHEN	0.83	1.10	1.41	1.27
OFFICE	0.69	0.90	0.89	0.77
PARENTS' ROOM	0.63	0.87	1.32	1.10
CHILDREN'S ROOM	0.50	0.67	0.85	0.72
SUM OF PEAKS	3.70 (41.71 W/m^2)	4.95 (54.94 W/m^2)	**6.65 (62.17 W/m^2)**	5.84 (48.49 W/m^2)
SIMULTANEOUS PEAK	3.60	4.85	6.01	5.32

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-18: Heating and cooling percentiles for extreme cold and hot years, for the house in Seville, with the non-Zoning system.

The same machine as for Malaga (DAIKIN RZP100DV1 + FHYBP100BV1) is, initially, capable of counteracting the load percentile for the building in the other towns and cities studied. Note that for Madrid the heat peak is higher than the cooling peak.

With the total mass flow moved by the interior unit and by applying the load ratios in the equation (5.5), the nominal mass flows continue to the zone (Table 5-19). With the flow for the kitchen and the equation (5.6), the load for ventilation is calculated and, therefore, the total load in the machine (Table 5-20). Malaga is added for the sake of completeness.

TOWN/CITY	CONCEPT	LIVING ROOM	KITCHEN	OFFICE	PARENTS' ROOM	CHILDREN'S ROOM
MADRID	FRACTION	0.284	0.218	0.181	0.176	0.141
	FLOW [kg/h]	552.096	423.792	351.864	342.144	274.104
BARCELONA	FRACTION	0.331	0.198	0.134	0.202	0.135
	FLOW [kg/h]	643.464	384.912	260.496	392.688	262.440
VALENCIA	FRACTION	0.328	0.201	0.130	0.211	0.130
	FLOW [kg/h]	637.632	390.744	252.720	410.184	252.720
SEVILLE	FRACTION	0.327	0.212	0.134	0.198	0.129
	FLOW [kg/h]	635.688	412.128	260.496	384.912	250.776

Table 5-19: Mass nominal flows to each system for the non-Zoning system

TOWN/CITY	DEMAND	LIVING ROOM [kW]	KITCHEN + VENTILAT. [Kw]	OFFICE [kW]	PARENTS' ROOM [kW]	CHILDREN'S ROOM [kW]	SUM OF PEAKS [kW]
MADRID	HEATING	1.85	3.38	1.18	1.15	0.92	**8.48**
	COOLING	1.95	2.14	0.74	1.12	0.75	6.70
BARCELONA	COOLING	1.85	1.72	0.75	1.13	0.75	**6.20**
MALAGA	COOLING	1.98	2.13	0.79	1.24	0.77	**6.91**
VALENCIA	COOLING	1.99	2.03	0.79	1.28	0.78	**6.87**
SEVILLE	COOLING	2.18	2.85	0.89	1.32	0.85	**8.09**

Table 5-20: Design powers for the non-Zoning system

The machine selected, with nominal sensitive powers of 7.1 kW in cold and 12.8 kW in heat, is capable of covering the peak load in all the towns and cities studied, except in Seville in cold mode, with a defect of 1 kW. The decision to fit model "100" or the immediately superior model, the "125", in Seville is a design-based decision at the discretion of the designer. Since the leap in installed power is substantial (from 7.1 kW to 9 kW), only due to ventilation, in this research work, the decision was taken to continue with the "100" model for the non-Zoning system in Seville. Simulation of the full problem (building + machine, Figure 5-3) for a typical year shows that the lower capacity machine can reasonably meet the demands of the house in Seville.

5.4 Zoning system with no power reduction in the machine

In this case, the air distribution network is fitted with mechanisms for adjusting thermal supply in the system to the demand for each zone separately. As a result of this, the machine should now be dimensioned to meet the maximum simultaneous demand for the zones,

$$PNM_Z = max\left\{ max\left(\sum_{i=1}^{Nzones} Q_{heat}^{i}(t) \right), max\left(\sum_{i=1}^{Nzones} Q_{cold}^{i}(t) \right) \right\} \quad . \qquad \textbf{(5.7)}$$

In the non-Zoning system, the machine's controller attempted to maintain the set conditions in the controlled zone. In the Zoning system, the machine will attempt to maintain the drive temperature at a constant, leaving the temperature for each zone to their respective drive damper, which acts by introducing the required amount of air.

In the case in point, internal gains will continue the profile shown in Table 4-8, for which reason the simultaneous peaks calculated in the section above are not strictly valid. Note that to dimension a non-Zoning system, the maximum internal generation possible in each zone separately should be assumed. As regards all other aspects, the methodology is the same, so that we will not going into further detail.

The results are shown in the following tables: 5-21 (Malaga peaks), 5-22 (Malaga percentiles), 5-23 (Madrid peaks), 5-24 (Madrid percentiles), 5-25 (Barcelona peaks), 5-26 (Barcelona percentiles), 5-27 (Valencia peaks), 5-28 (Valencia percentiles), 5-29 (Seville peaks), 5-30 (Seville percentiles), 5-31 (Mass flows) and 5-32 (total simultaneous load).

	HEAT PEAK [kW]		**COOLING PEAK [kW]**	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.29	1.47	2.43	2.15
KITCHEN	1.03	1.16	1.44	1.25
OFFICE	0.77	0.90	1.04	0.88
PARENTS' ROOM	0.80	0.94	1.52	1.32
CHILDREN'S ROOM	0.62	0.71	0.98	0.80
SUM OF PEAKS	4.51	5.18	7.41	6.40
SIMULTANEOUS PEAK	4.34 (48.27 W/m^2)	5.01 (55.72 W/m^2)	**6.43 (71.52 W/m^2)**	5.45 (60.62 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-21: Absolute heating and cooling peaks for extremely cold and hot years, for the house in Malaga, with the Zoning system with no power reduction

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.87	1.25	1.86	1.59
KITCHEN	0.70	0.98	1.04	0.86
OFFICE	0.56	0.80	0.79	0.63
PARENTS' ROOM	0.51	0.76	1.18	0.97
CHILDREN'S ROOM	0.38	0.58	0.76	0.60
SUM OF PEAKS	3.02	4.37	5.63	4.65
SIMULTANEOUS PEAK	2.94 (32.70 W/m^2)	4.27 (47.49 W/m^2)	**4.92 (54.72 W/m^2)**	4.09 (45.49 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-22: Heating and cooling percentiles for extremely cold and hot years, for the house in Malaga, with the Zoning system with no power reduction

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.63	2.00	2.27	2.01
KITCHEN	1.29	1.56	1.26	1.14
OFFICE	1.03	1.26	0.90	0.81
PARENTS' ROOM	1.00	1.28	1.34	1.10
CHILDREN'S ROOM	0.80	1.01	0.91	0.76
SUM OF PEAKS	5.75	7.11	6.68	5.82
SIMULTANEOUS PEAK	5.64 (62.73 W/m^2)	**6.95 (77.30 W/m^2)**	5.69 (63.28 W/m^2)	5.02 (55.83 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-23: Absolute heating and cooling peaks for extremely cold and hot years, for the house in Madrid, with the Zoning system with no power reduction

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.41	1.86	1.83	1.48
KITCHEN	1.11	1.43	1.02	0.78
OFFICE	0.91	1.19	0.74	0.56
PARENTS' ROOM	0.87	1.16	1.07	0.89
CHILDREN'S ROOM	0.69	0.92	0.72	0.55
SUM OF PEAKS	4.99	6.56	5.38	4.26
SIMULTANEOUS PEAK	4.87 (54.16 W/m^2)	**6.46 (71.85 W/m^2)**	4.66 (51.83 W/m^2)	3.67 (40.82 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-24: Heating and cooling percentiles for extremely cold and hot years, for the house in Madrid, with the Zoning system with no power reduction

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.32	1.80	2.34	2.00
KITCHEN	1.01	1.42	1.34	1.08
OFFICE	0.83	1.09	1.05	0.78
PARENTS' ROOM	0.84	1.09	1.35	1.20
CHILDREN'S ROOM	0.65	0.83	0.99	0.78
SUM OF PEAKS	4.65	6.23	7.07	5.84
SIMULTANEOUS PEAK	4.51 (50.16 W/m^2)	6.15 (68.40 W/m^2)	**6.29 (69.96 W/m^2)**	4.92 (54.72 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-25: Absolute heating and cooling peaks for extremely cold and hot years, for the house in Barcelona, with the Zoning system with no power reduction

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.07	1.42	1.72	1.37
KITCHEN	0.86	1.10	0.94	0.69
OFFICE	0.71	0.90	0.75	0.53
PARENTS' ROOM	0.65	0.87	1.08	0.85
CHILDREN'S ROOM	0.51	0.68	0.72	0.54
SUM OF PEAKS	3.80	4.97	5.21	3.98
SIMULTANEOUS PEAK	3.66 (40.71 W/m^2)	**4.88 (54.28 W/m^2)**	4.61 (51.27 W/m^2)	3.38 (37.59 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-26: Heating and cooling percentiles for extremely cold and hot years, for the house in Barcelona, with the Zoning system with no power reduction

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.13	1.63	2.31	2.13
KITCHEN	0.89	1.25	1.35	1.22
OFFICE	0.69	1.02	1.00	0.92
PARENTS' ROOM	0.67	1.03	1.47	1.17
CHILDREN'S ROOM	0.56	0.83	0.93	0.88
SUM OF PEAKS	3.94	5.76	7.06	6.32
SIMULTANEOUS PEAK	3.81 (42.38 W/m^2)	5.61 (62.40 W/m^2)	**6.09 (67.73 W/m^2)**	5.55 (61.73 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-27: Absolute heating and cooling peaks for extremely cold and hot years, for the house in Valencia, with the Zoning system with no power reduction

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.96	1.37	1.85	1.45
KITCHEN	0.76	1.07	1.03	0.77
OFFICE	0.61	0.90	0.79	0.59
PARENTS' ROOM	0.57	0.86	1.17	0.91
CHILDREN'S ROOM	0.43	0.67	0.75	0.57
SUM OF PEAKS	3.33	4.87	5.59	4.29
SIMULTANEOUS PEAK	3.23 (35.92 W/m^2)	4.70 (52.27 W/m^2)	**4.86 (54.05 W/m^2)**	3.77 (41.93 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-28: Heating and cooling percentiles for extremely cold and hot years, for the house in Valencia, with the Zoning system with no power reduction

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.30	1.68	2.53	2.21
KITCHEN	1.02	1.31	1.54	1.35
OFFICE	0.82	1.03	1.23	0.99
PARENTS' ROOM	0.83	1.02	1.61	1.31
CHILDREN'S ROOM	0.64	0.83	1.00	0.87
SUM OF PEAKS	4.61	5.87	7.91	6.73
SIMULTANEOUS PEAK	4.45 (49.49 W/m^2)	5.76 (64.06 W/m^2)	**6.83 (75.96 W/m^2)**	5.93 (65.95 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-29: Absolute heating and cooling peaks for extremely cold and hot years, for the house in Seville, with the Zoning system with no power reduction

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.06	1.42	2.07	1.86
KITCHEN	0.84	1.11	1.22	1.08
OFFICE	0.69	0.90	0.89	0.77
PARENTS' ROOM	0.64	0.87	1.28	1.08
CHILDREN'S ROOM	0.50	0.67	0.82	0.70
SUM OF PEAKS	3.73	4.97	6.28	5.49
SIMULTANEOUS PEAK	3.62 (40.26 W/m^2)	4.89 (54.39 W/m^2)	**5.67 (63.06 W/m^2)**	4.99 (55.50 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-30: Heating and cooling percentiles for extremely cold and hot years, for the house in Seville, with the Zoning system with no power reduction

Commence by applying the machine RZP100DV1+FHYBP100DV1 for all the towns and cities.

TOWN/CITY	CONCEPT	LIVING ROOM	KITCHEN	OFFICE	PARENTS' ROOM	CHILDREN'S ROOM
MALAGA	FRACTION	0.330	0.185	0.140	0.210	0.135
	FLOW [kg/h]	641.520	359.640	272.160	408.24	262.440
MADRID	FRACTION	0.284	0.218	0.181	0.177	0.140
	FLOW [kg/h]	552.096	423.792	351.864	344.088	272.160
BARCELONA	FRACTION	0.331	0.180	0.144	0.207	0.138
	FLOW [kg/h]	643.464	349.920	279.936	402.408	268.272
VALENCIA	FRACTION	0.331	0.184	0.141	0.209	0.135
	FLOW [kg/h]	643.464	357.696	274.104	406.296	262.44
SEVILLE	FRACTION	0.329	0.194	0.142	0.204	0.131
	FLOW [kg/h]	639.576	377.136	276.048	396.576	254.664

Table 5-31: Mass nominal flows to each zone for the Zoning system with no power reduction

TOWN/CITY	YEAR	DEMAND	LIVING ROOM [kW]	KITCHEN [kW]	OFFICE [kW]	PARENTS' ROOM [kW]	CHILDREN'S ROOM [kW]	SIMULTANEOUS PEAK [kW]
MALAGA	CAL.	COOLING	1.86	1.84	0.79	1.18	0.76	**5.73**
MADRID	FRIO	HEATING	1.86	3.54	1.19	1.16	0.92	**7.76**

	CAL.	COOLING	1.83	1.95	0.74	1.07	0.72	5.60
BARCELONA	CAL.	COOLING	1.72	1.47	0.75	1.08	0.72	**5.17**
VALENCIA	CAL.	COOLING	1.85	1.75	0.79	1.17	0.75	**5.60**
SEVILLE	CAL.	COOLING	2.07	1.79	0.89	1.28	0.82	**6.15**

Table 5-32: Design powers for the Zoning system with no power reduction

Although in Madrid the heat peak is dominant, the cold peak is also shown to indicate that it can be met (the machine counteracts less sensitive cold load than hot due to the latent component). The combination selected RZP100DV1+FHYBP100DV1 is suitable for all the towns and cities, except for Barcelona where the immediately lower capacity model can be installed (RZP71DV1+FHYBP71DV1), with a nominal cold sensitive power of 5.2kW.

5.5 Zoning system with power reduction in the machine

The capacity of the Zoning system to differentiated between the state of the zones makes it possible not to acclimatize those that are unoccupied or which, being occupied, do not demand acclimatization. As shown in §6, and as can easily be sensed, this involves an important source of saving in operational costs of the system. Also, since in many applications it is unlikely for all the demands to occur simultaneously (e.g., for all the rooms in a house to be occupied at the same time), a further step can be taken by exploiting Zoning to reduce the installed power and, thus, cut down on investment costs. We refer to this issue with the term *machine power reduction*.

The nominal reduced power will be equal to the simultaneous load in the occupied zones

$$PNM_{RDZ} = max\left\{ max\left(\sum_{i=1}^{Nzones} Q_{heat}^{i}\left(t\right)\cdot OC_{i}\left(t\right) \right), max\left(\sum_{i=1}^{Nzones} Q_{cold}^{i}\left(t\right)\cdot OC_{i}\left(t\right) \right) \right\} \quad , \qquad \textbf{(5.8)}$$

where

$$OC_i\left(t\right) = \begin{cases} 1 & \text{if the zone is occupied} \\ 0 & \text{if the zone is unoccopied} \end{cases} .$$

The resulting value of (5.8) is nothing but an approximation since if the zones adjacent to an acclimatized zone are not really that, the demand in the acclimatized zone will not be equal to the thermal load, but rather will be higher or lower, depending on the circumstances. The dimensioning obtained from the above criteria must be checked by making an explicit simulation with the machine, as in §6 under real usage conditions. Anticipating events, the machines selected provide satisfactory results.

The procedure for dimensioning is identical to the one followed above: the equation terms are calculated (5.8), a machine is selected, depending on the building load, the nominal flows to the zone are calculated by the ratios (5.5), the net load in the machine is calculated and checks are run to ensure that the machine selected continues to have the capacity to counteract it.

The following tables show the maximums, percentiles, sum of peaks and simultaneous peaks in each town or city.

	HEAT PEAK [kW]		**COOLING PEAK [kW]**	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.73	0.93	2.43	2.14
KITCHEN	0.79	0.93	1.44	1.25
OFFICE	0.33	0.48	1.04	0.88
PARENTS' ROOM	0.76	0.88	1.52	1.32
CHILDREN'S ROOM	0.54	0.65	0.98	0.73
SUM OF PEAKS	3.15	3.87	7.41	6.32
SIMULTANEOUS PEAK	2.20 (24.47 W/m^2)	2.59 (28.81 W/m^2)	**4.89 (54.39 W/m^2)**	4.26 (47.38 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-33: Absolute heating and cooling peaks for extreme years of cold and heat, for the house in Malaga, with the Zoning system and with power reduction

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.23	0.58	1.86	1.58
KITCHEN	0.46	0.71	1.04	0.86
OFFICE	0.02	0.27	0.77	0.61
PARENTS' ROOM	0.47	0.73	1.18	0.97
CHILDREN'S ROOM	0.35	0.55	0.69	0.55
SUM OF PEAKS	1.53	2.84	5.54	4.57
SIMULTANEOUS PEAK	0.91 (10.12 W/m^2)	1.64 (18.24 W/m^2)	**3.80 (42.26 W/m^2)**	3.20 (35.59 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-34: Heating and cooling percentiles for extreme years of cold and heat, for the house in Malaga, with the Zoning system and with power reduction

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	1.07	1.44	2.27	1.99
KITCHEN	1.13	1.33	1.26	1.14
OFFICE	0.55	0.85	0.89	0.80
PARENTS' ROOM	0.99	1.23	1.34	1.09
CHILDREN'S ROOM	0.75	0.94	0.91	0.72
SUM OF PEAKS	4.49	5.79	6.67	5.74
SIMULTANEOUS PEAK	2.87 (31.92 W/m^2)	3.91 (43.49 W/m^2)	**4.40 (48.94 W/m^2)**	3.93 (43.71 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-35: Absolute heating and cooling peaks for extreme years of cold and heat, for the house in Madrid, with the Zoning system and with power reduction

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.77	1.22	1.82	1.47
KITCHEN	0.86	1.16	1.02	0.78
OFFICE	0.40	0.67	0.72	0.54
PARENTS' ROOM	0.85	1.14	1.07	0.89
CHILDREN'S ROOM	0.64	0.87	0.66	0.48
SUM OF PEAKS	3.52	5.06	5.29	4.16
SIMULTANEOUS PEAK	2.10 (23.36 W/m^2)	3.18 (35.37 W/m^2)	**3.63 (40.37 W/m^2)**	2.92 (32.48 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-36: Heating and cooling percentiles for extreme years of cold and heat, for the house in Madrid, with the Zoning system and with power reduction

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.78	1.18	2.34	2.00
KITCHEN	0.81	1.20	1.34	1.08
OFFICE	0.40	0.63	1.05	0.78
PARENTS' ROOM	0.78	1.09	1.35	1.20
CHILDREN'S ROOM	0.58	0.81	0.95	0.78
SUM OF PEAKS	3.35	4.91	7.03	5.84
SIMULTANEOUS PEAK	2.21 (24.58 W/m^2)	2.89 (32.14 W/m^2)	**4.72 (52.50 W/m^2)**	3.85 (42.82 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-37: Absolute heating and cooling peaks for extreme years of cold and heat, for the house in Barcelona, with the Zoning system and with power reduction

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.45	0.78	1.72	1.37
KITCHEN	0.60	0.83	0.94	0.69
OFFICE	0.18	0.41	0.70	0.52
PARENTS' ROOM	0.62	0.85	1.08	0.85
CHILDREN'S ROOM	0.46	0.64	0.66	0.48
SUM OF PEAKS	2.31	3.51	5.10	3.91
SIMULTANEOUS PEAK	1.33 (14.79 W/m^2)	2.08 (23.13 W/m^2)	**3.51 (39.04 W/m^2)**	2.70 (30.03 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-38: Heating and cooling percentiles for extreme years of cold and heat, for the house in Barcelona, with the Zoning system and with power reduction

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.57	1.07	2.32	2.14
KITCHEN	0.69	1.04	1.35	1.22
OFFICE	0.25	0.59	1.00	0.92
PARENTS' ROOM	0.66	0.98	1.51	1.23
CHILDREN'S ROOM	0.47	0.74	0.88	0.85
SUM OF PEAKS	2.64	4.42	7.06	6.36
SIMULTANEOUS PEAK	1.76 (19.57 W/m^2)	2.97 (33.03 W/m^2)	**4.68 (52.05 W/m^2)**	4.28 (47.60 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-39: Absolute heating and cooling peaks for extreme years of cold and heat, for the house in Valencia, with the Zoning system and with power reduction

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.30	0.72	1.87	1.47
KITCHEN	0.51	0.79	1.03	0.78
OFFICE	0.08	0.29	0.79	0.58
PARENTS' ROOM	0.53	0.80	1.21	0.95
CHILDREN'S ROOM	0.39	0.60	0.69	0.52
SUM OF PEAKS	1.81	3.20	5.59	4.30
SIMULTANEOUS PEAK	1.04 (11.57 W/m^2)	1.94 (21.58 W/m^2)	**3.82 (42.49 W/m^2)**	2.95 (32.81 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-40: Heating and cooling percentiles for extreme years of cold and heat, for the house in Valencia, with the Zoning system and with power reduction

	HEAT PEAK [kW]		COOLING PEAK [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.72	1.12	2.53	2.21
KITCHEN	0.80	1.11	1.54	1.35
OFFICE	0.38	0.56	1.13	0.99
PARENTS' ROOM	0.77	1.03	1.61	1.31
CHILDREN'S ROOM	0.57	0.75	1.00	0.81
SUM OF PEAKS	3.24	4.57	7.81	6.65
SIMULTANEOUS PEAK	2.16 (24.02 W/m^2)	3.12 (34.70 W/m^2)	**5.16 (57.39 W/m^2)**	4.53 (50.38 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-41: Absolute heating and cooling peaks for extreme years of cold and heat, for the house in Seville, with the Zoning system and with power reduction

	HEAT PERCENTILE [kW]		COOLING PERCENTILE [kW]	
	WARM YEAR	*COLD YEAR*	*WARM YEAR*	*COLD YEAR*
LIVING ROOM	0.40	0.74	2.07	1.86
KITCHEN	0.58	0.83	1.22	1.08
OFFICE	0.10	0.35	0.89	0.77
PARENTS' ROOM	0.61	0.85	1.29	1.08
CHILDREN'S ROOM	0.45	0.64	0.76	0.65
SUM OF PEAKS	2.14	3.41	6.23	5.44
SIMULTANEOUS PEAK	1.26 (14.01 W/m^2)	2.04 (22.69 W/m^2)	**4.33 (48.16 W/m^2)**	3.86 (42.93 W/m^2)

(*) Only the acclimatized surface of the house is used (89.91m^2)

Table 5-42: Heating and cooling percentiles for extreme years of cold and heat, for the house in Seville, with the Zoning system and with power reduction

Synthesizing these results, the following table shows the conditioning columns (cooling for a warm year)

TOWN/CITY	YEAR	%	LIVING ROOM [kW]	KITCHEN [kW]	OFFICE [kW]	PARENTS' ROOM [kW]	CHILDREN'S ROOM [kW]	SIMULTANEOUS PEAK [kW]
MADRID	HOT	100	2.27	1.26	0.89	1.34	0.91	4.40
		99	1.82	1.02	0.72	1.07	0.66	**3.63**
BARCELONA	HOT	100	2.34	1.34	1.05	1.35	0.95	4.72
		99	1.72	0.94	0.70	1.08	0.66	**3.51**
MALAGA	HOT	100	2.43	1.44	1.04	1.52	0.98	4.89
		99	1.86	1.04	0.77	1.18	0.69	**3.80**
VALENCIA	HOT	100	2.32	1.35	1.00	1.51	0.88	4.68
		99 %	1.87	1.03	0.79	1.21	0.69	**3.82**
SEVILLE	HOT	100 %	2.53	1.54	1.13	1.61	1.00	5.16
		99 %	2.07	1.22	0.89	1.29	0.76	**4.33**

Table 5-43: Load peaks in each zone and simultaneous peak for each town or city

The (exterior unit) machine DAIKIN RZP71DV1 + FHYBP71BV1 (interior unit), with a nominal cold power of 5.2 kW, can counteract the simultaneous peak in the building in all of the towns and cities.

Once again, the simultaneous in the previous table represent the load in the building, to which we must add the load for ventilation from the kitchen to calculate the load in the machine. Table 5-44 shows the nominal flows to each zone for the machine above, and Table 5-45 shows the simultaneous peak in the machine. This acts as a based for dimensioning and which shows agreement with the initial selection.

TOWN/CITY	CONC.	LIVING ROOM	KITCHEN	OFFICE	PARENTS' ROOM	CHILDREN'S ROOM	TOTAL
MADRID	FRACTION	0.344	0.193	0.136	0.202	0.125	1.000
	FLOW [kg/h]	470.592	264.024	186.048	276.336	171.000	1368.000
BARCELONA	FRACTION	0.337	0.184	0.137	0.212	0.130	1.000
	FLOW [kg/h]	461.016	251.712	187.416	290.016	177.840	1368.000
MALAGA	FRACTION	0.336	0.188	0.139	0.213	0.124	1.000
	FLOW [kg/h]	459.648	257.184	190.152	291.384	169.632	1368.000
VALENCIA	FRACTION	0.334	0.184	0.141	0.216	0.125	1.000
	FLOW [kg/h]	456.912	251.712	192.888	295.488	171.000	1368.000
SEVILLE	FRACTION	0.332	0.196	0.143	0.207	0.122	1.000
	FLOW [kg/h]	454.176	268.128	195.624	283.176	166.896	1368.000

Table 5-44: Nominal mass flow to each zone for the interior unit DAIKIN fhybp71bv1

TOWN/CITY	SIMULTANEOUS COLD PEAK IN THE MACHINE [kW]
MADRID	4.19
BARCELONA	3.90
MALAGA	4.36
VALENCIA	4.35
SEVILLE	5.25

Table 5-45: Maximum load to be counteracted by the machine for the Zoning system with power reduction

5.6 Summary of case studies

Table 5-46 summaries the selection of machine and the nominal drive flows to each zone. For Barcelona, the same machine was used (model 71) for cases with and without power reduction, so that the results are identical for both. In order to broaden the conclusions arising from this study, the performance of machine 100 fitted in the Zoning system in Barcelona is also evaluated.

TOWN/CITY	SYSTEM	NOMINAL MASS FLOW [kg h^{-1}]					MACHINE (DAIKIN)
		LIVING ROOM	KITCHEN	OFFICE	PARENTS' ROOM	CHILD-REN'S ROOM	
MADRID	NZ	552.096	423.792	351.864	342.144	274.104	100
	Z SRP	552.096	423.792	351.864	344.088	272.160	100
	Z CRP	470.592	264.024	186.048	276.336	171.000	71
BARCELONA	NZ	643.464	384.912	260.496	392.688	262.440	100
	Z SRP	643.464	349.920	279.936	402.408	268.272	(100)
	Z CRP	461.016	251.712	187.416	290.016	177.840	71
MALAGA	NZ	639.576	398.520	254.664	400.464	639.576	100
	Z SRP	641.520	359.640	272.160	408.24	262.440	100
	Z CRP	459.648	257.184	190.152	291.384	169.632	71
VALENCIA	NZ	637.632	390.744	252.720	410.184	252.720	100
	Z SRP	643.464	357.696	274.104	406.296	262.44	100
	Z CRP	456.912	251.712	192.888	295.488	171.000	71
SEVILLE	NZ	635.688	412.128	260.496	384.912	250.776	100
	Z SRP	639.576	377.136	276.048	396.576	254.664	100
	Z CRP	454.176	268.128	195.624	283.176	166.896	71

Table 5-46: Dimensioning the systems

SIXTH CHAPTER

Results and Conclusions

6.1 Introduction

Regarding the characteristics of the house (§4) and of the machine and duct network (§5) in the model developed for the acclimatization system (§3), for each time passage in the simulation the interior conditions in all the zones are obtained (temperature and relative air humidity, surface temperature of the seals, etc.) and the operational conditions of the machine (thermal power supplied, electrical consumption, COP, etc.) as well as the value of all the heat and mass flows involved.

This chapter is divided into two parts:

- *Performance of the models* (§6.2), looking at the evolution over time of the most significant variables, illustrating curious effects and checking on the general performance of the models formulated in §3. To this end, the particular town specified in the simulation is irrelevant, for which reason only the results for Malaga are shown.
- *Technical characteristics of the systems* (§6.3), quantified in terms of the monthly and annual integrated electrical demand and the extent to which the set conditions and conditions of comfort are met.

6.2 Performance of the models

6.2.1 Overview

Asynchrony of load between the zones in the house, basically due to differences in orientation of the exterior seals, is clearly shown by following the evolution over time of the sensitive load in the acclimatized zones in the course of any day of the year, e.g., on 15th February (Figure 6-1). At the same time, some zones demand to be cooled or to be heated.

The living room demands its cooling peak in the afternoon, when the Sun falls strongly on the west side; the parents' bedroom has its peak at midday since it has a wide balcony facing south; the children's room and the office, with windows facing east, have their peak in the morning.

Sudden variations in load are due to changes in the use of the zones, which alter their internal gains, according to the agenda in Table 4-8. The clearest example of this is noted at 20:00, at which time the entire family is going to dine in the kitchen.

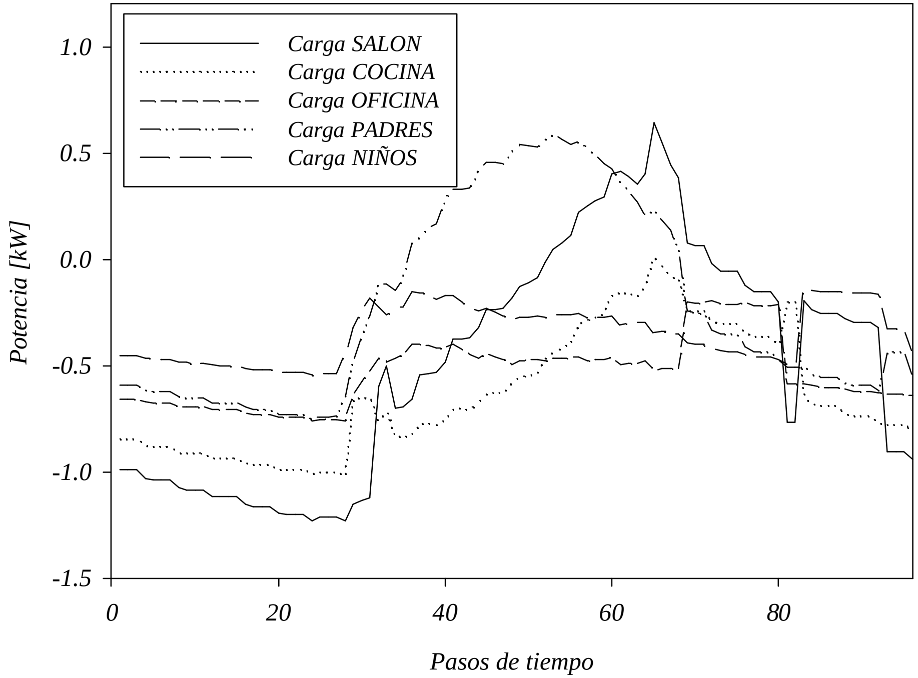


Figure 6-1: Load in each acclimatized zone for 15th February, in a typical year in Malaga (key to signs: Q>0 cooling; Q<0 heating). This covers a time passage of 15 minutes.

In reality, the zones do not permanently keep their set temperature, as is assumed in a load calculation, but rather:

- At each moment, the machina only operates in one of the two modes (hot or cold), so that the system is unable to COPE with any simultaneous thermal inversions that may occur in the house

- In the Zoning system, the unoccupied zones are not acclimatized
- In the non-Zoning system, there is only one zone with controlled temperature
- The machine may have power limitations within the margin of the design percentile
- The *law of control* for the controlled zones is illustrated in Figure 6-2, showing a band of thermal comfort (in dry air temperature) within which no action is required of the machine.

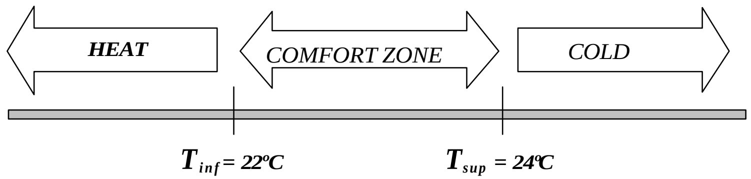


Figure 6-2: Machine control in terms of the temperature in the living room. In winter, the machine attempts to maintain the temperature at the lower limit (22ºC), and in summer at the upper (24ºC)

In order to consider these aspects and others such as the dependency of the machine's technical characteristics on its operational conditions, in some way, it is necessary to represent the mutual interaction between the building and the acclimatization system (§2.5), for which purpose the models explained in §3 were developed.

6.2.2 *Non-Zoning acclimatization system*

The non-Zoning control system depends on the demand from the living room, depending on the regions in Figure 6-2, it being the conditions in the other zones as a result of the situation at any moment in time. Figure 6-3 shows the temperatures that this system achieves in each room of the house.

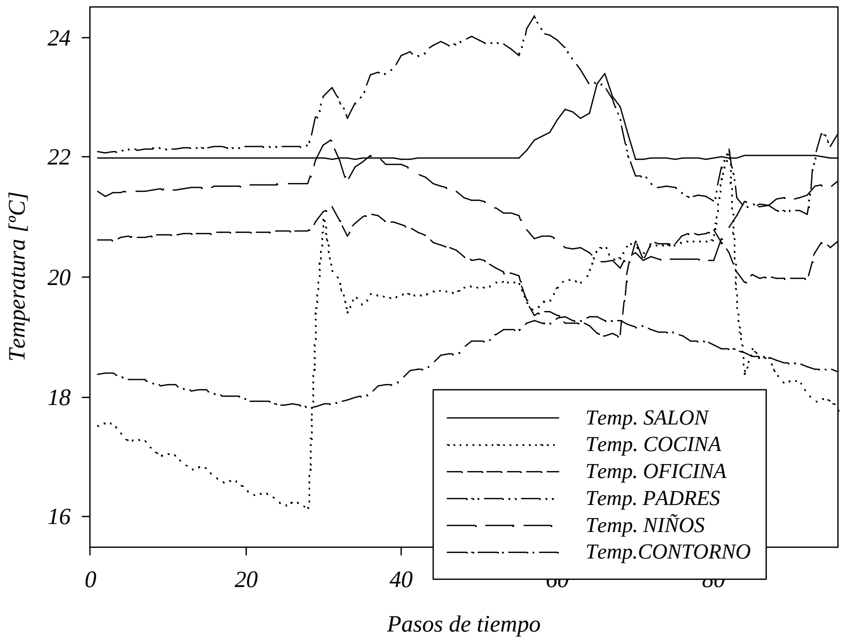


Figure 6-3: Evolution of temperature in the zones for 15th February in a typical year in Malaga, for the non-Zoning system. The profile zone models the adjacent flats (§4.3). The time passage covers 15 minutes.

The living room temperature is maintained at its set level of 22ºC (the small fluctuations are numeric noise, at all times framed in a margin of $\pm$ 0.05ºC), except from 14:45 and 17:45 when the local conditions mean that acclimatizing is not necessary, which leads to the machine turning off (Figure 6-4).

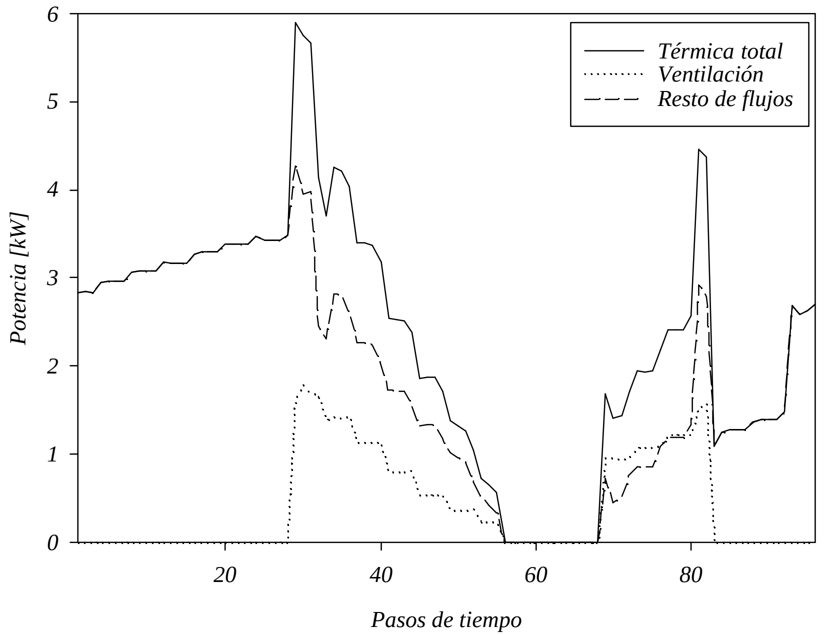


Figure 6-4: Interchanged thermal power in the course of 15th February, in a typical year in Malaga, for the non-Zoning system. The first maximum corresponds to the start of drive to the kitchen, which introduces the ventilation load into the machine. The second maximum concerns the disappearance of the internal heat sources when the family goes to dine in the kitchen. The considerable subsequent drop occurs after the evening meal, when load for ventilation is withdrawn and internal heat gains knock in once again. This time passage covers 15 minutes.

During the night, the kitchen temperature drops drastically, falling close to room temperature, as it is assumed that it receives no acclimatization except when it is occupied and with the extractor hood turned on.

Note in Figure 6-3 that, although the machine was in heat mode, just at the time when it stops, there is a slight increase of temperature in the parents' bedroom. This is explained by the decreasing trend noted in the bedroom temperature in the preceding hours. Since there are some cold zones, the mix of returns and their subsequent drive by the machine was providing a degree of cooling for the parents' bedroom (with considerable solar gains at this time) and the mix of heat needed to keep the living room in the required conditions. Withdrawal of the machine's action leads to a fall in temperature in the rest of the house. This is clarified in Figure 6-5, which includes the drive temperature.

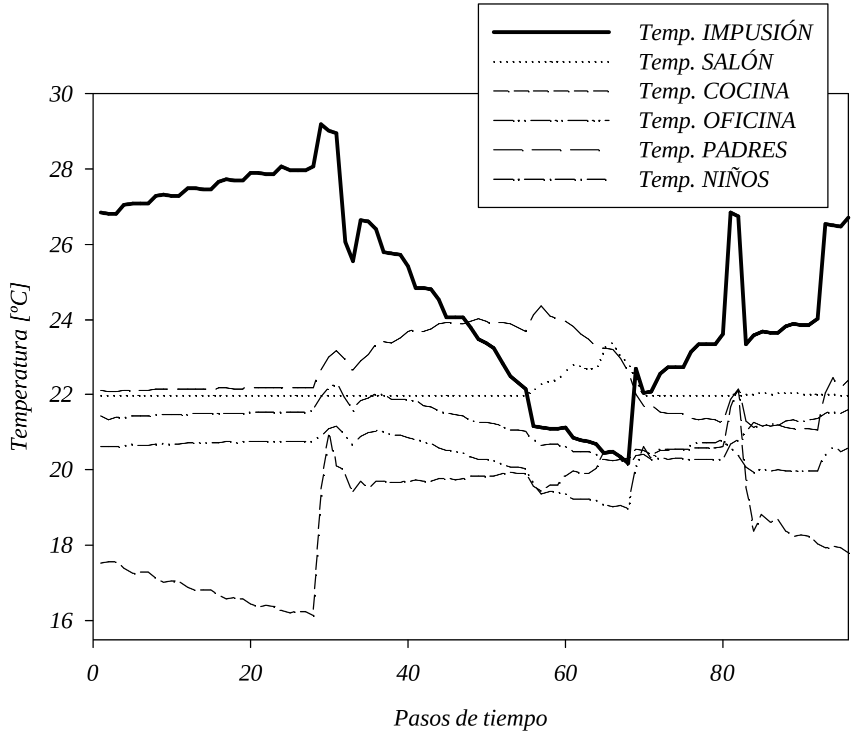


Figure 6-5: Evolution of the drive temperature, on 12th February, in a typical year in Malaga, for the non-Zoning system. During the time that the machine is stopped, the drive temperature drawn is that of the mix of returns (there is no real drive). The average temperature in the house falls. Time passage = 15 min.

As summer draws nearer, the living room temperature during the stoppage period will gradually increase until occasionally exceeding 24ºC, the top limit for the comfort window (Figure 6-2). Under these circumstances, the machine switches to cold mode to keep the living room temperature at 24ºC (Figures 6-6 and 6-7). Likewise, if in the summer period the living room temperature fell below 22ºC, heat mode would switch on to keep it at 22ºC. These situations, especially the former, are common in spring and autumn (Figure 6-9).

Figure 6-8 illustrates the typical conditions on a summer's day, when the set temperature is 24ºC. During this season of the year, it is uncommon for the living room temperature to fall below 24ºC, the machine keeping operating practically all of the time, as noted in Figure 6-9, which shows the temperature framework in the living room for the year.

In the non-controlled zones, temperature variations are far more intense. By way of an example, Figures 6-10 and 6-11 show the progression of temperature in the bedrooms for the entire year.

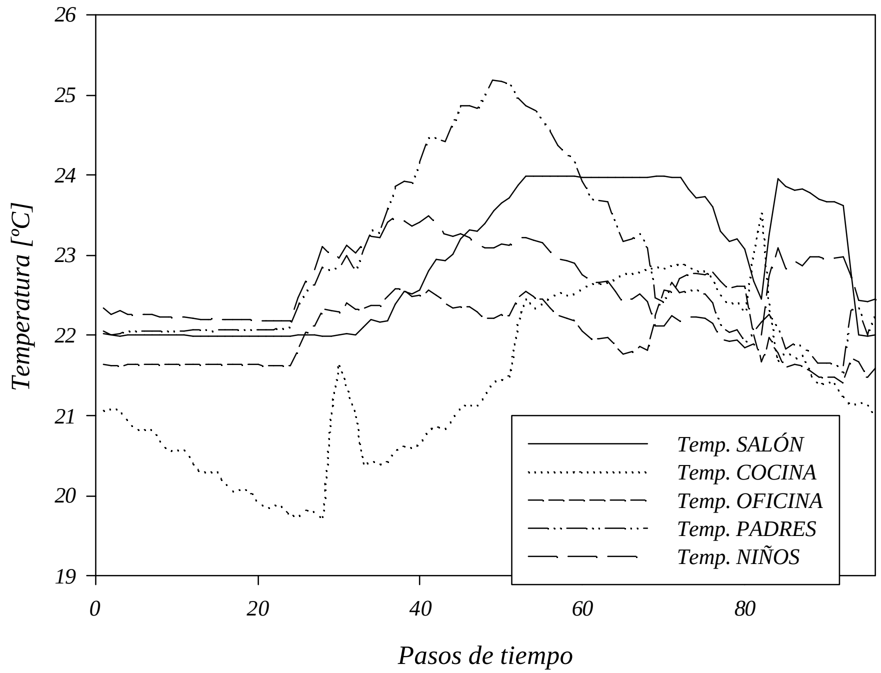


Figure 6-6: Evolution of temperature in the acclimatized zones on 11th April, in a typical year in Malaga, for the non-Zoning system. Since the early hours (9:00), the living room does not demand acclimatization, but as from 14:00, its temperature goes over 24ºC, so that cold mode is activated to keep the temperature at 24ºC (the nearest value). At 20:30, this need disappears.

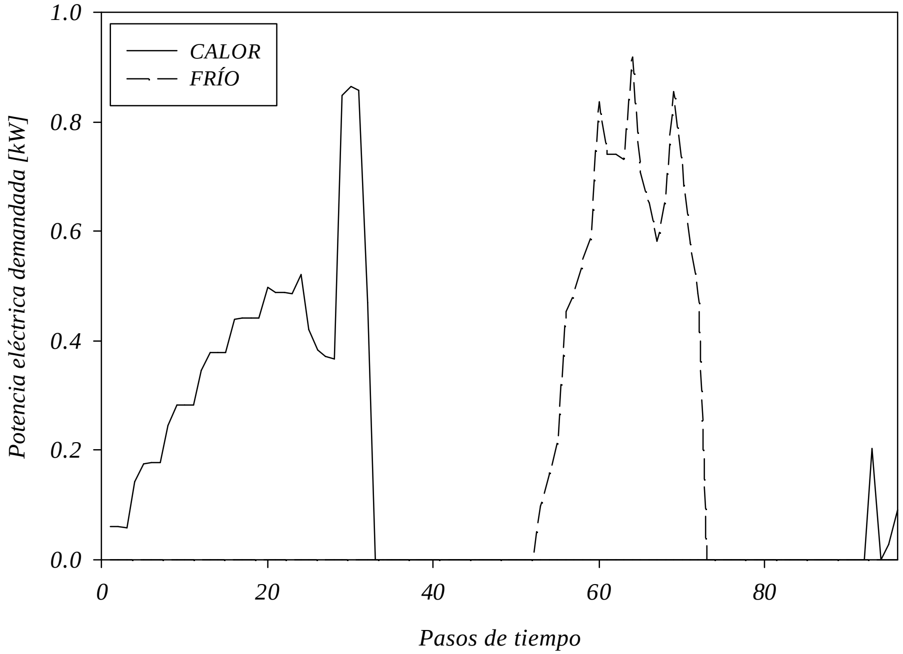


Figure 6-7: Electrical power demanded by the machine on 11th April, in a typical year, in heat and cold modes, in Malaga, for the non-Zoning system. Time passage is 15 minutes.

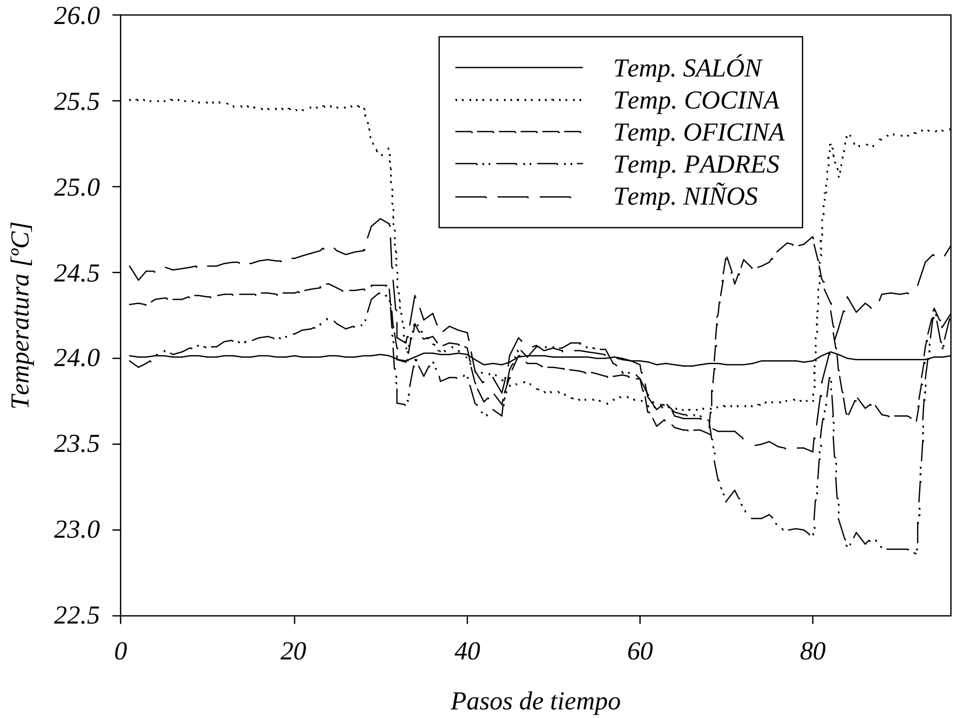


Figure 6-9: Temperature in the living room for a typical year, in Malaga, for the non-Zoning system. Its value is maintained within the range [22ºC, 24ºC], the machine operating as explained. Fluctuations around the limits of the range are numerical noise with a maximum amplitude of $\pm$0.05ºC.

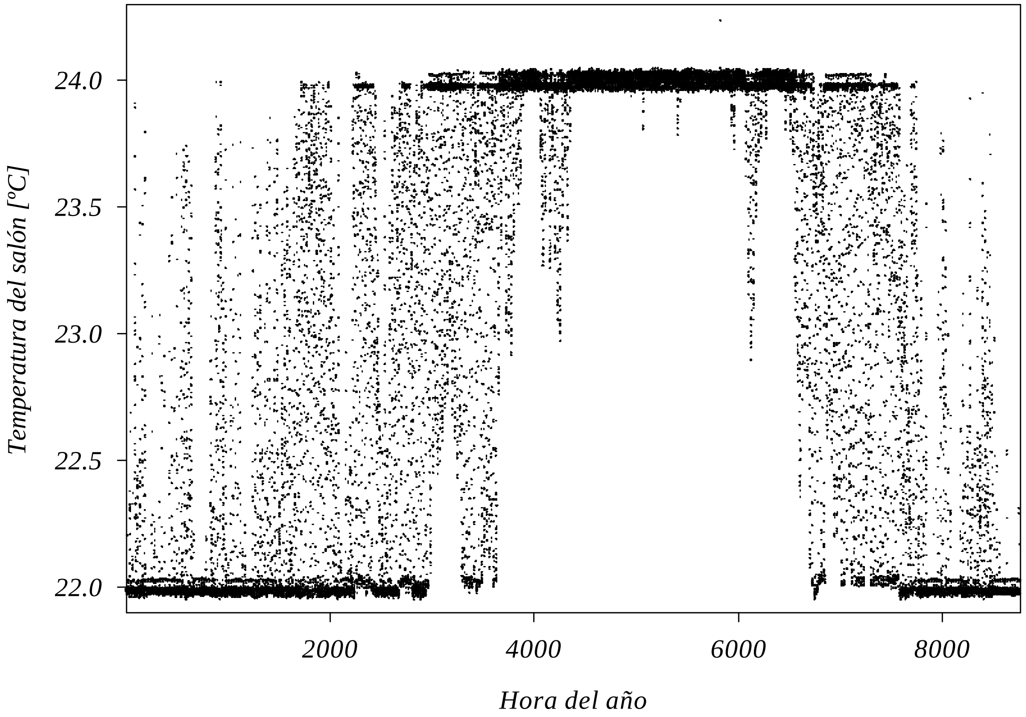


Figure 6-11: Temperature in the children's room during a typical year, in Malaga, for the non-Zoning system

Figure 6-9 highlights a break in the living room temperature in summer, on 31st August (Figure 6-12). In the mid hours of this day, the situation means that the machine is incapable of supplying the total load demanded from it.

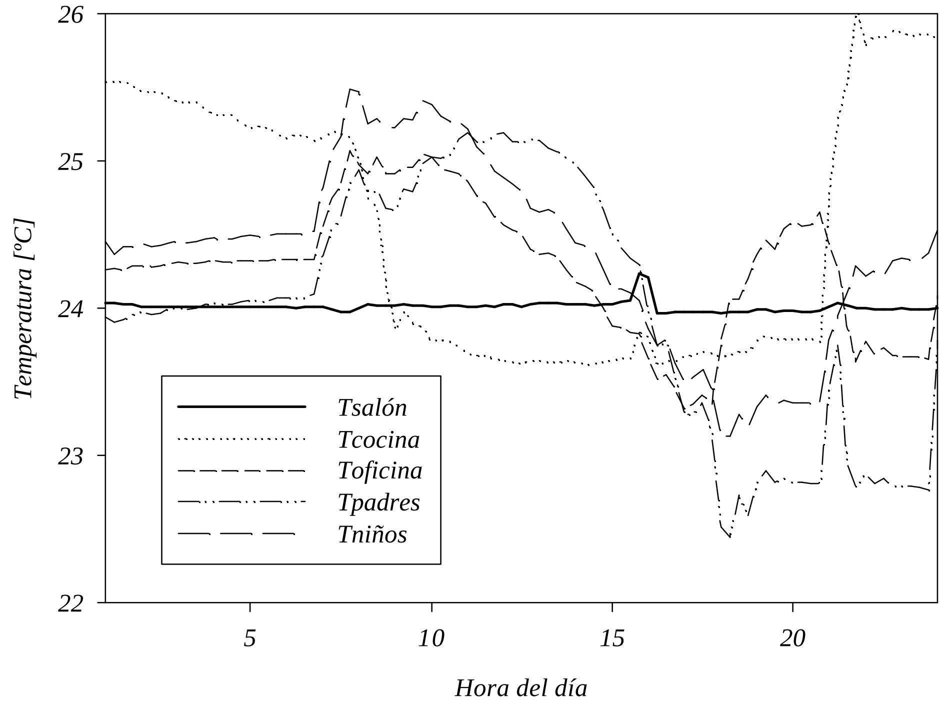


Figure 6-12: Temperature on 31st August, in a typical year, in Malaga, for the non-Zoning system

If instead of fitting the machine, in line with the dimensioning (§5.3), the immediately lower capacity machine were fitted – the RZP71DV1 – the power limitations would be less frequent, as shown in Figure 6-13 for the living room, which is overheated for a total of 46 hours. Although this level of discomfort may appear to be acceptable and a good reason for fitting the small machine, it should not be overlooked that in the non-Zoning system, users of the other rooms may change the setting in the living room to achieve acceptable conditions in the zones where they are (e.g., in the bedrooms on summer mornings). Depending on the circumstances, this change of setting entails an increase or decrease in the power consumed by the machine, which makes the dimensioning criteria articulated in §5.3 safer than the mere observance of the result in Figure 6-13 (see the interpretation of results in §7 for a more detailed explanation).

In Figures 6-14 to 6-19, the description of the system is completed, illustrating the electrical consumptions, drive temperature (which is kept within the normal limits), the relative humidity in the living room and the average operational time of the machine.

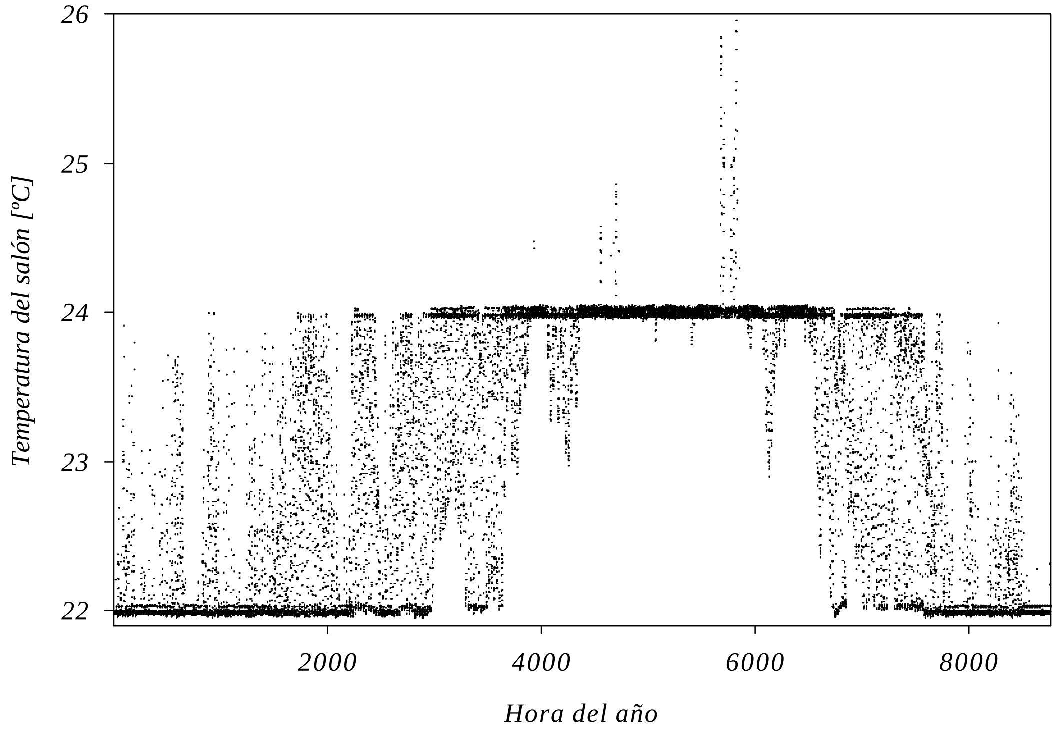


Figure 6-13: Annual evolution of temperature in the living room, during a typical year, in Malaga, for the non-Zoning system with the machine RZP71DV1

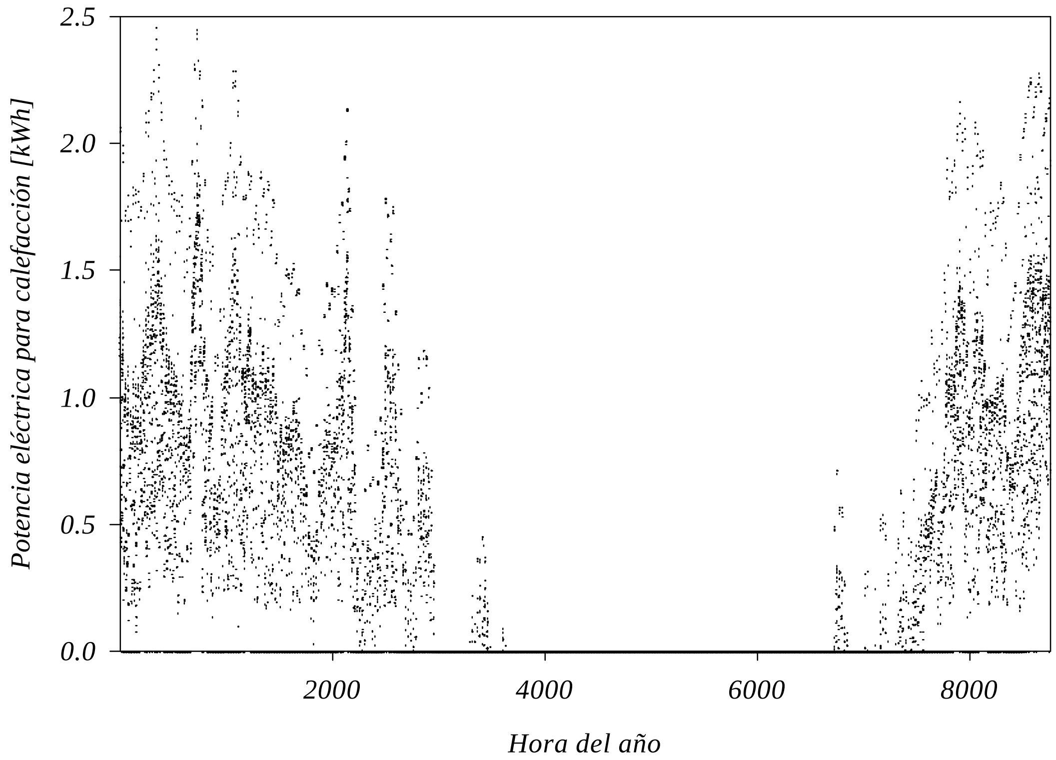


Figure 6-14: Electrical power demanded by the machine for heating

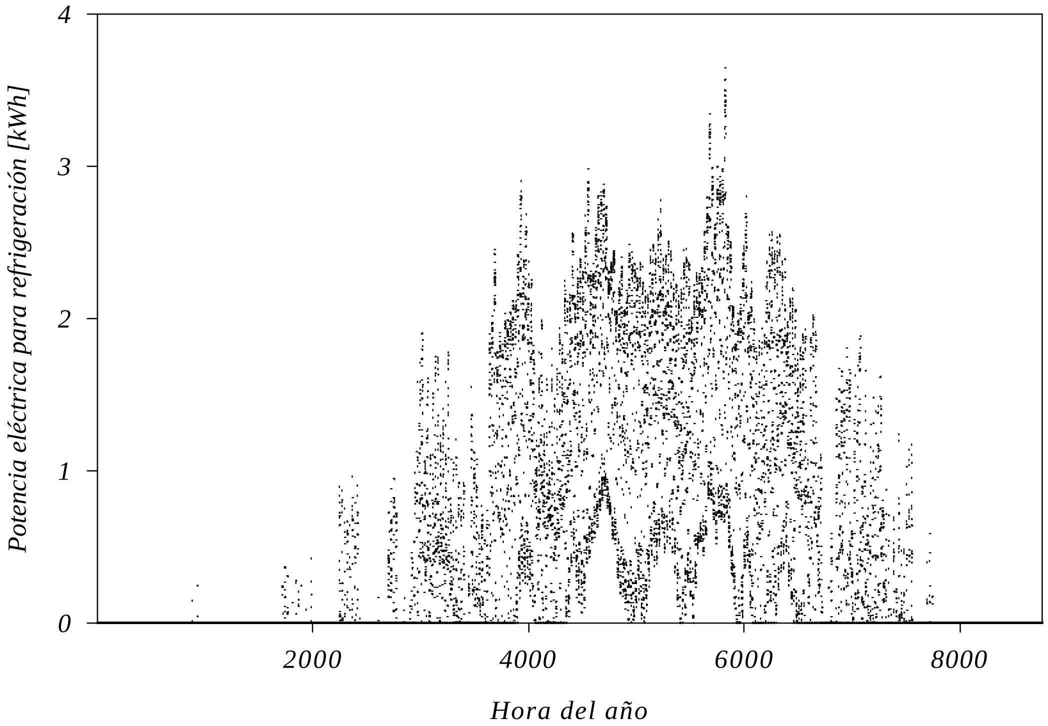


Figure 6-15: Electrical power demanded by the machine for cooling.

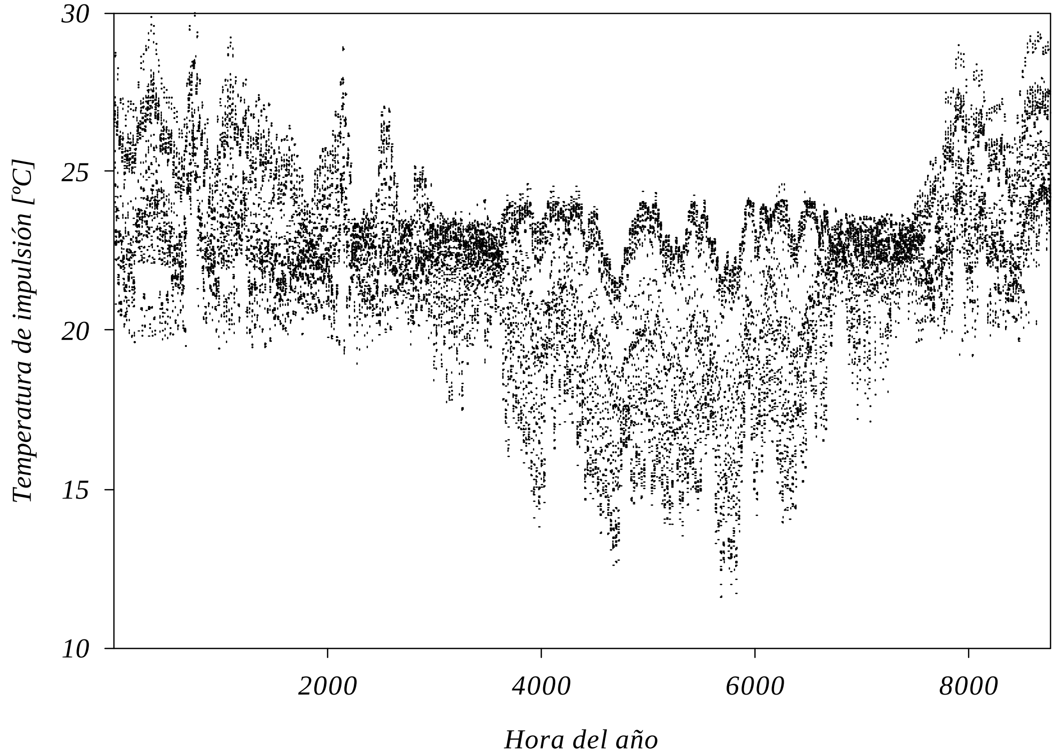


Figure 6-16: Drive temperature for a typical year, in Malaga, for the non-Zoning system.

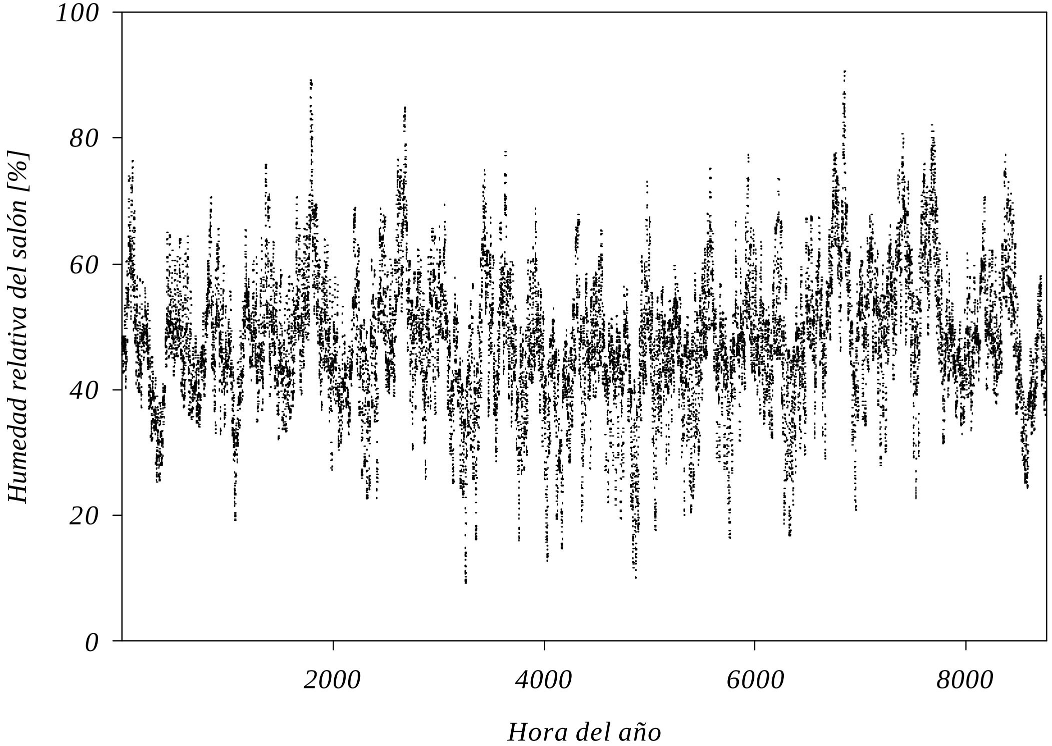


Figure 6-17: Relative humidity in the living room, for a typical year, in Malaga, for the non-Zoning system.

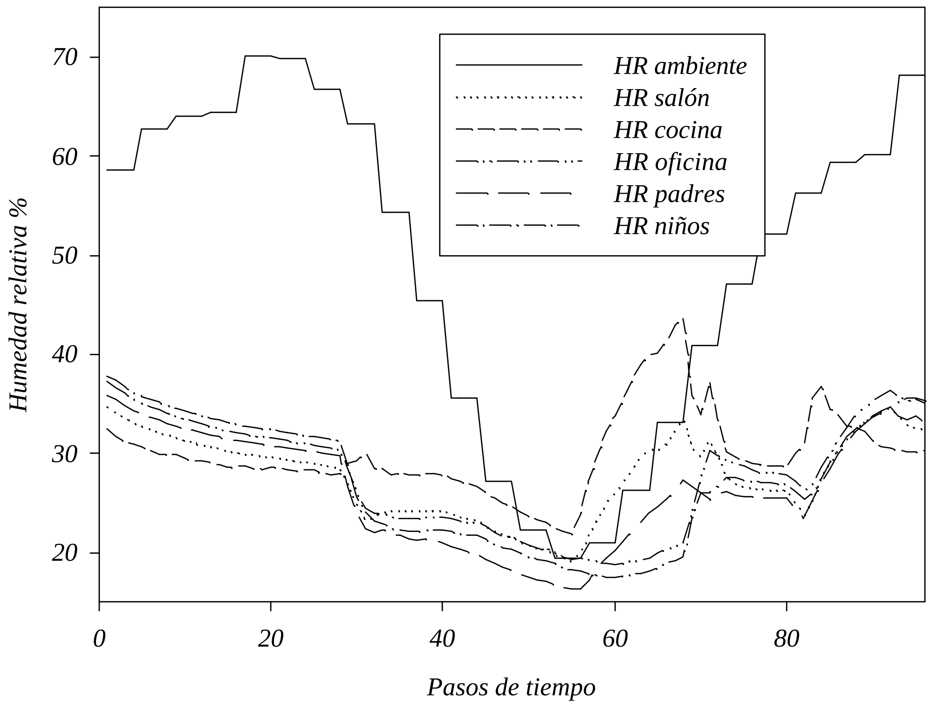


Figure 6-18: Relative humidity levels on 14th February, for a typical year, in Malaga, for the non-Zoning system. Time passage is 15 minutes.

In cold mode, the calculation of absolute drive humidity is a problem due to the lack of information on the dependence of the sensitive heat factor with part load (§3) and the limited range of data provided by the manufacturer. For these reasons, it is highly likely that the extreme values noted in Figure 6-17 during summer do not, in reality, appear. During the winter months, relatively low humidity levels are recorded, basically due to the effect of infiltrations and ventilation (Figure 6-18).

Finally, the following figure shows the so-called “Part Load Ratio”, i.e., the quotient between the thermal load demanded and the maximum load that the machine can supply for each time passage. When it is an all-nothing machine, this variable can be interpreted as the fraction of the time passage during which the machine should operate. As expected, in winter the machine is large for the needs of the building, whereas in summer it frequently operates close to full load.

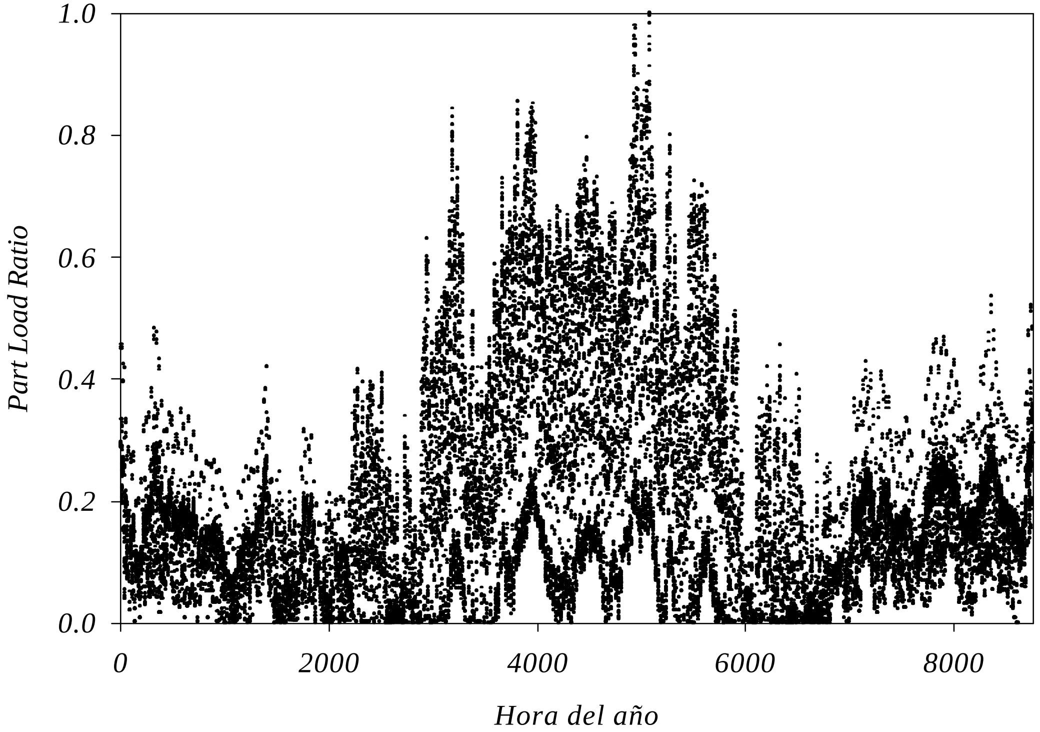


Figure 6-19: Part Load Ratio (thermal demand / possible maximum supply)

6.2.3 *Zoning system*

This system controls the temperature, separately, in each zone, and makes it possible to acclimatize unoccupied zones (*set-back)*, which means a substantial saving in energy (§6.3.2).

Figures 6-20 and 6-21 show the course of temperatures in the house for a day in winter and one in summer respectively. In the occupied zones, settings are kept, whereas in the unoccupied zones, no acclimatization is received, so that temperature evolves freely.

Note in Figure 6-30 that the parents' bedroom falls below 24ºC at midday, thus demanding cooling. This is a simultaneous thermal inversion situation, which the system has to deal with. In these cases, it has been organized such that it is the bedrooms by night and the living room by day that will decide if the machine operates in heat or cold mode. In practice, it is the user who does this, but this factor is somewhat arbitrary to model.

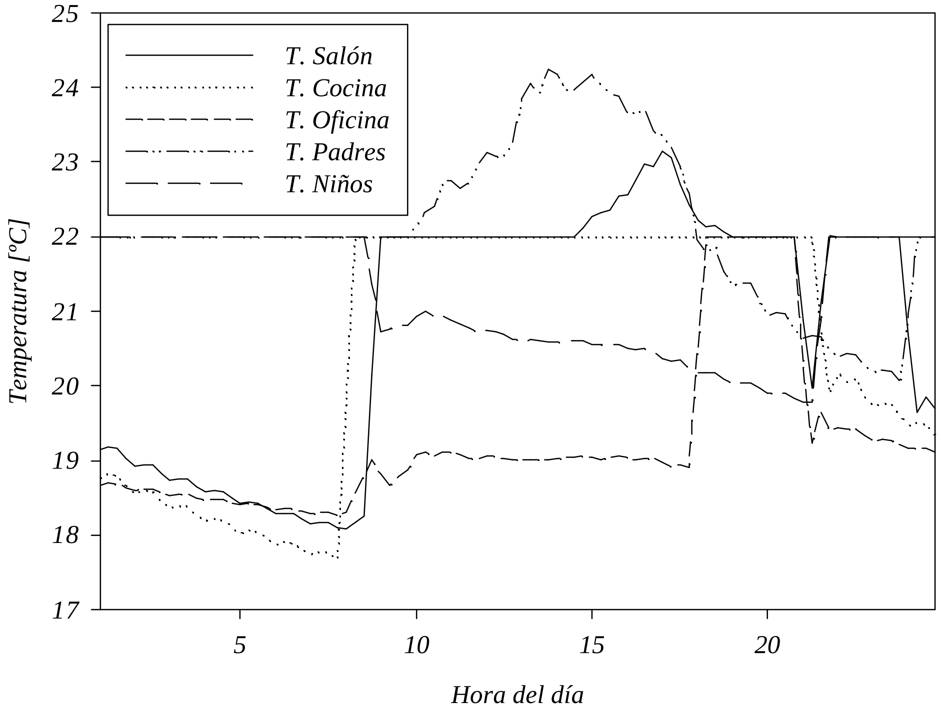


Figure 6-20: Temperature in the zones, on 23rd January, for a typical year, in Malaga, for the Zoning system.

Figure 6-22 illustrates the thermal power provided by the machine for a day in winter. This value is stable during the night and increases considerably in the morning, when people wake up and new zones (kitchen and living room) demand acclimatization. As the day unveils, the heat demand falls until it increases again by night. The sharp fall at 20:30 is due to disconnecting the zones when the entire family is having evening meal in the kitchen.

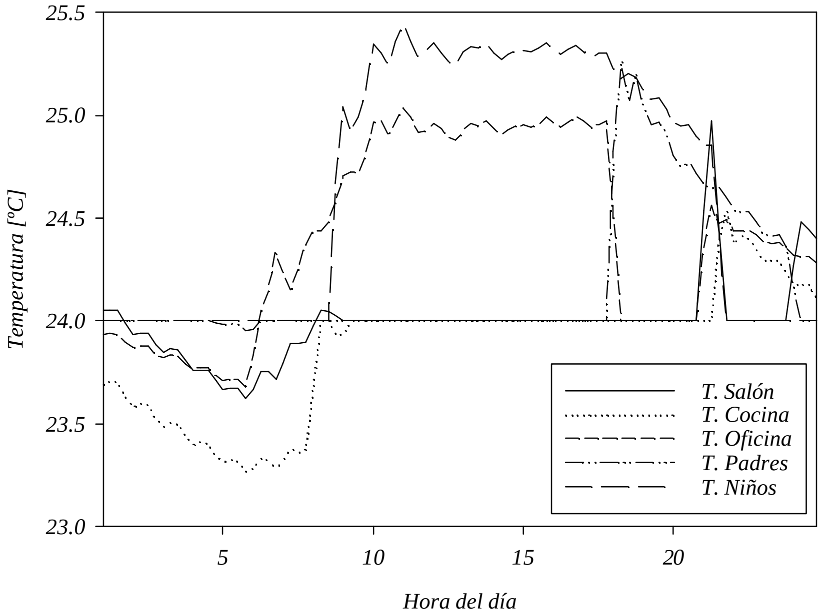


Figure 6-21: Temperature in the zones on 28th June, for a typical year, in Malaga, for the Zoning system.

Likewise, Figure 6-23 illustrates the thermal cold power supplied by the machine in the course of a day in summer. Obviously, the cooling demand is out of phase in terms of the heating demand (Figure 6-22), since it is almost zero during the night and increases as the day unveils and then falls again.

In the results on electrical consumption in §6.3.2, we see how the Zoning system always saves more in heat mode than in cold mode. This is easily explained in terms of the above: the heat demand is mostly during the winter nights, when the Zoning system only counteracts the demand in the bedrooms. The saving arising from not conditioning the rest of the house is, therefore, substantial.

Conversely, in summer the demand for cold during the night is normally minimal, so that night *set-back* is far less effective in terms of saving. During the day, the maximum number of occupied zones and the maximum demand for cooling are simultaneous. This fact is clearly reflected in Figures 6-24 and 6-25.

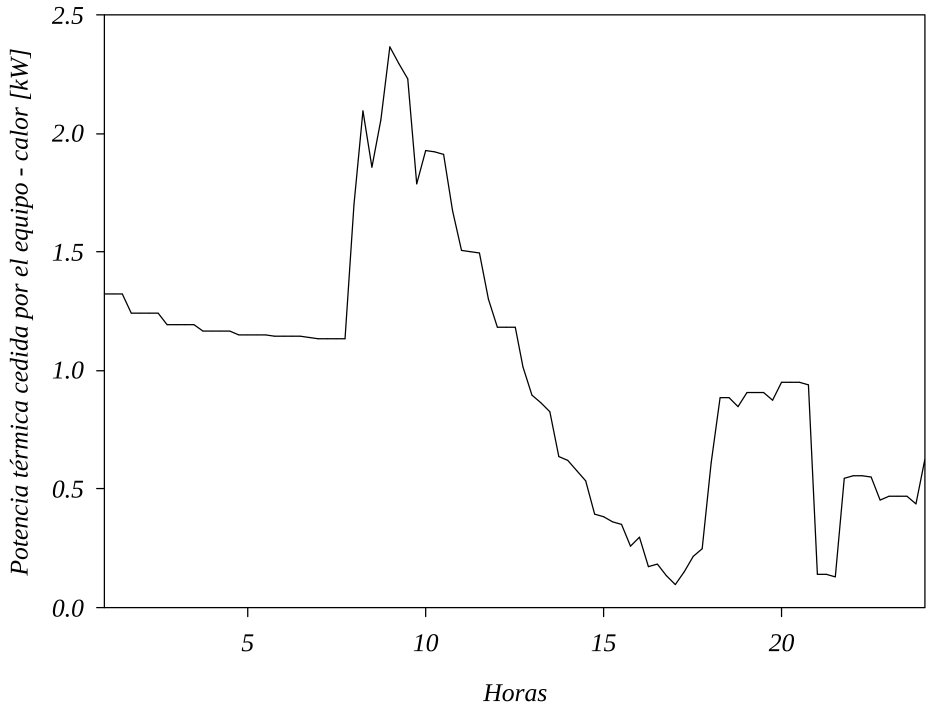


Figure 6-22: Thermal power supplied by the machine on 23rd January.

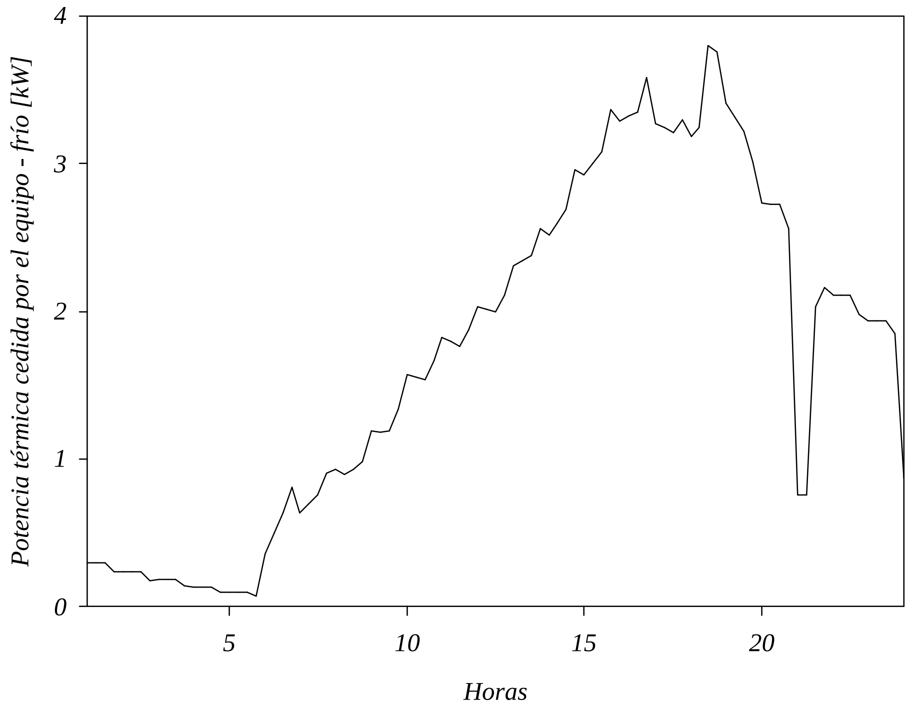


Figure 6-23: Temperature in the zones on 28th June, for a typical year, in Malaga, with the Zoning system.

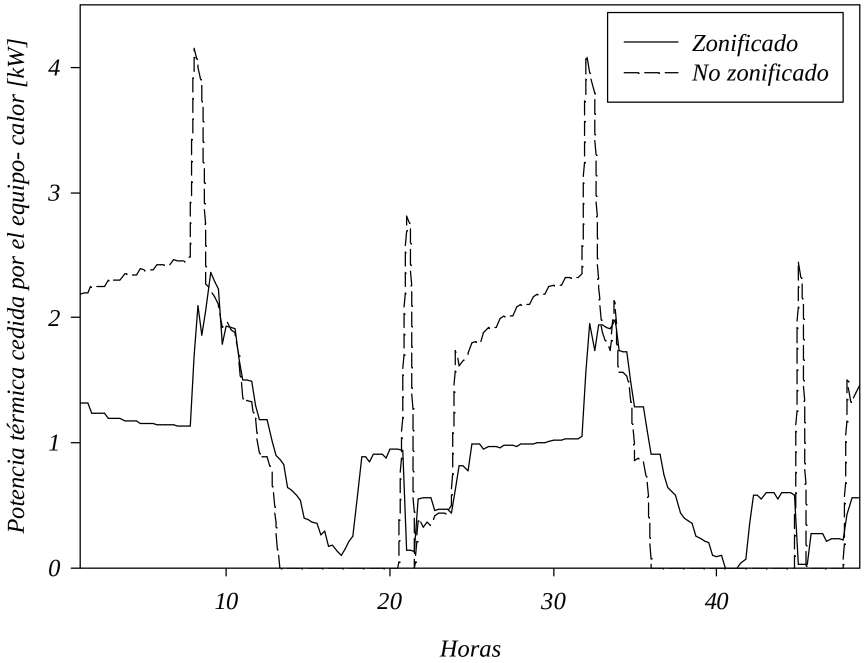


Figure 6-24: Evolution of the thermal power supplied by the machine over 48 hours typical of winter for each system. Night saving for the Zoning system is very considerable. Sudden changes are related to the start up of ventilation for the kitchen (annual weight is small, in the order of 5%), with changes in internal gains.

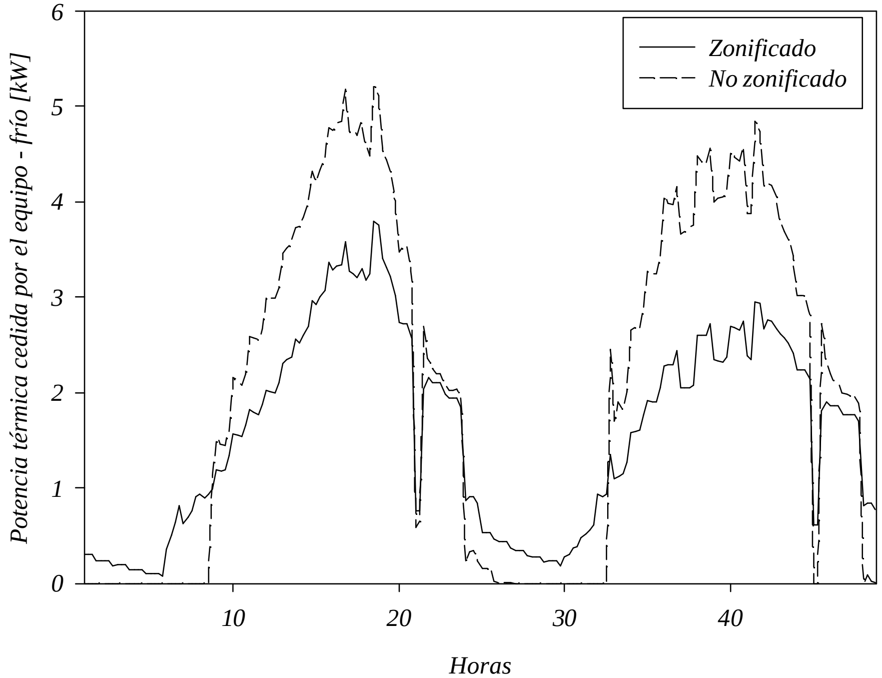


Figure 6-25: Evolution of the thermal power supplied by the machine over 48 hours typical of summer, with each acclimatization system. The night *set-back* does not, in this case, entail any saving, and in fact, the Zoning system consumers more due to the fact that it meets the conditions for comfort in the bedrooms.

As we have seen, the temperature in the unoccupied zones evolves freely, moving away from the setting value. This has repercussions on the adjacent acclimatized zones, which experience a changed heat flow through the seals that they share with the non-acclimatized zones. For this reason, it may occasionally happen that the demand in a zone is higher than the peak load used to select nominal mass flow (§5), so that the system is unable to inject the total energy required into a zone. This situation occurs when there are many zones disconnected and is more intense in the zones with a lower nominal mass flow (office and children's bedroom).

To meet the load need in a zone with this lack, the programme increase the mass flow to the zone as much as is needed, but always taking into account the limit of total flow moved by the ventilator. The physical counterpart to this adjustment is the non-ideal performance of the by-pass that never sends to the plenum all the excess flow in the duct network.

The following figures show, for the entire year, the quotient between the driven mass flow and the nominal flow in each zone (Table 5-46).

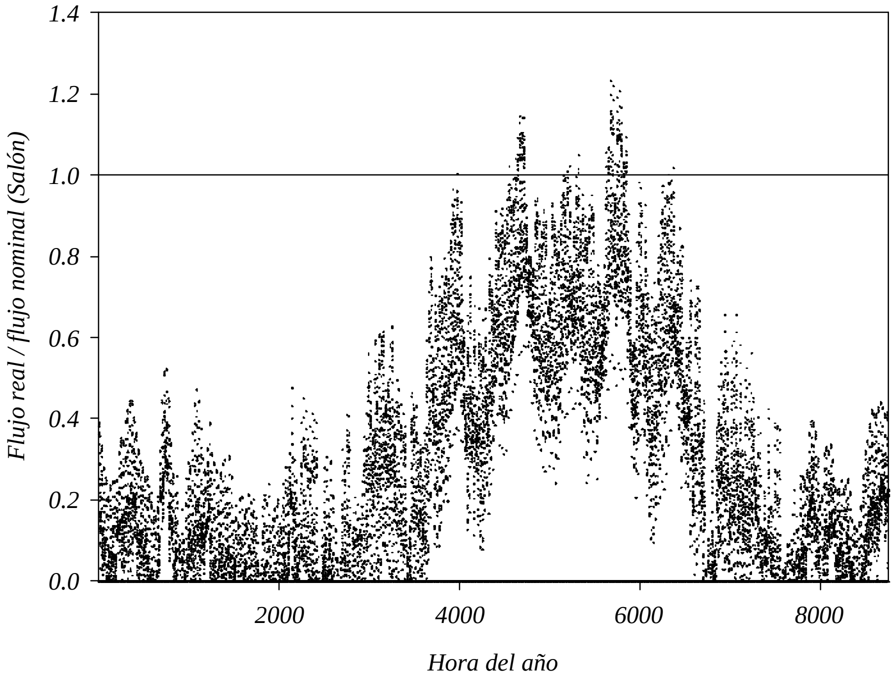


Figure 6-26: Excess flow for the living room, Malaga, Zoning system with power reduction.

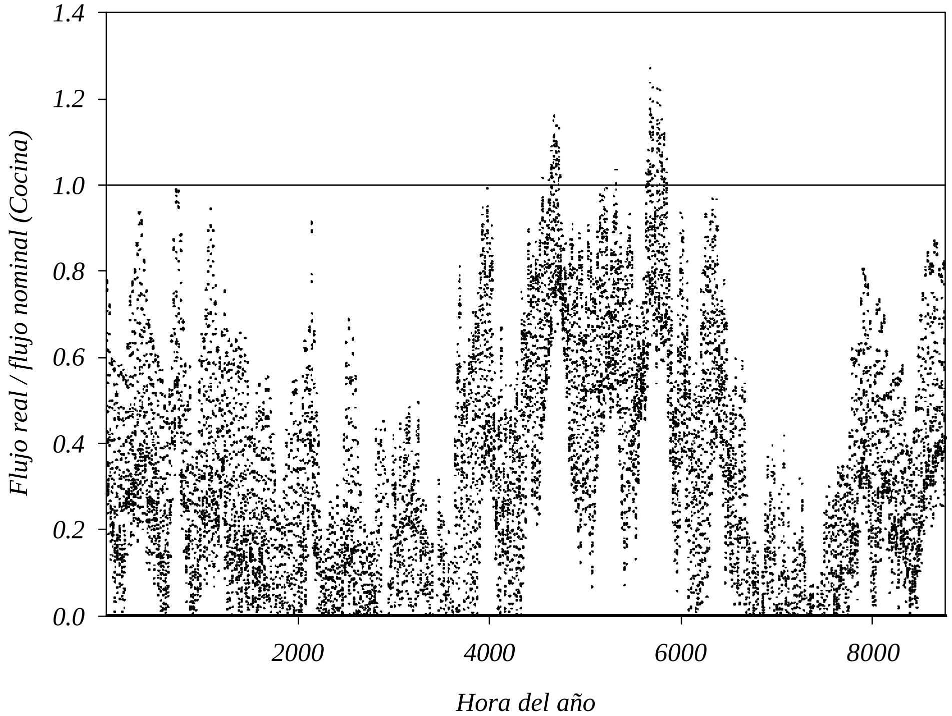


Figure 6-27: Excess flow for the kitchen, Malaga, Zoning system with power reduction

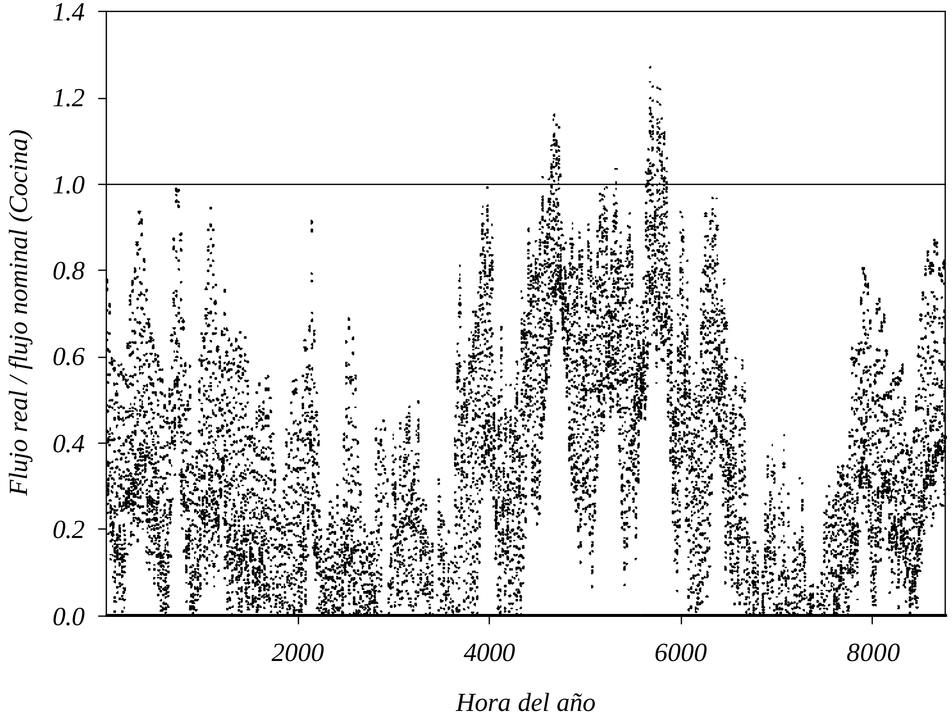


Figure 6-28: Excess flow for the office, Malaga, Zoning system with power reduction

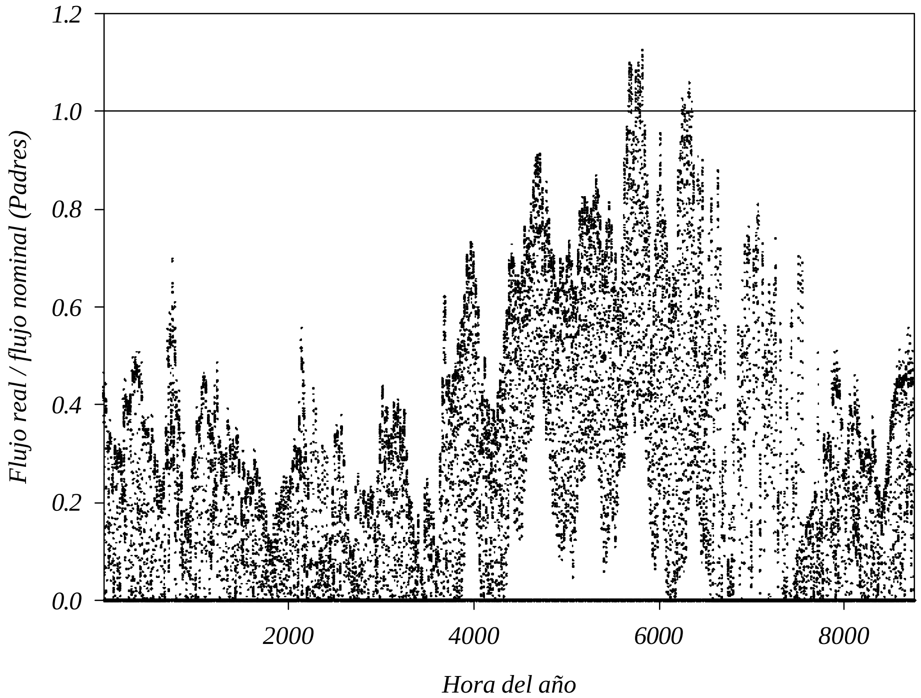


Figure 6-29: Excess flow for the parents’ bedroom, Malaga, Zoning system with power reduction

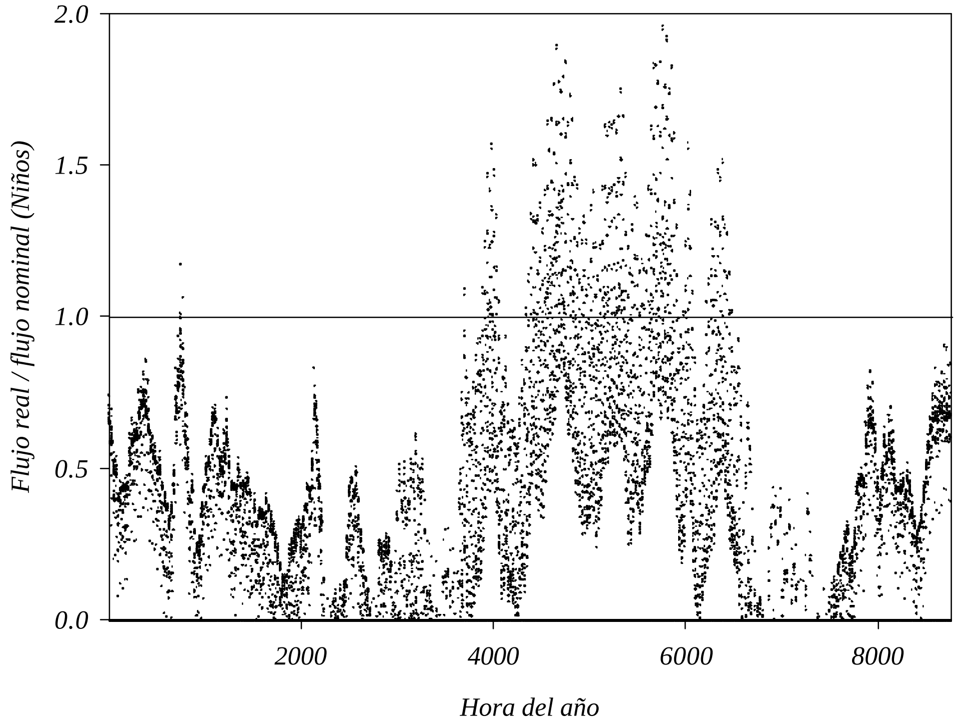


Figure 6-30: Excess flow for the children’s bedroom, Malaga, Zoning system with power reduction

The figure below shows the range of capacities of the inverter machine fitted in Malaga for a Zoning system with power reduction. Once again, during winter the demand is appreciably less than the maximum capacity, whereas in summer operation is nearer to full load.

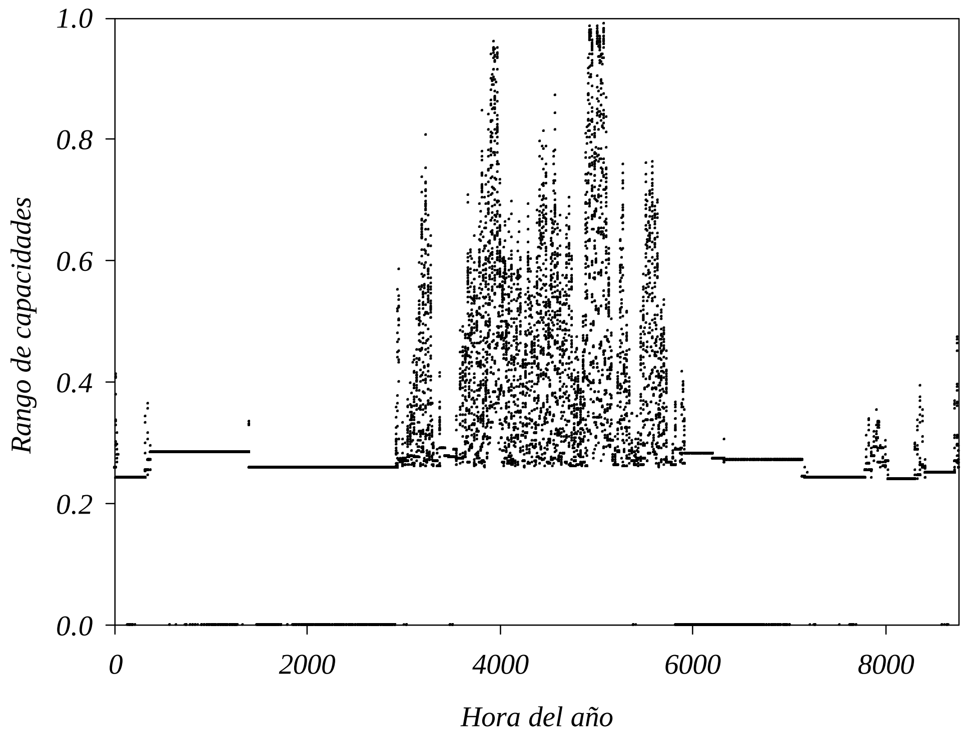


Figure 6-31: Range of capacities for the Zoning system with power reduction, in Malaga.

6.3 Technical characteristics of the systems

To get an idea of the performance of these systems and to be able to compare them, results from the simulations must be drawn up to obtain some values that summarize certain aspects of interest: thermal comfort and electrical consumption.

6.3.1 Thermal comfort

For the case in point, we take thermal comfort to mean the acclimatization system's capacity to maintain the dry air temperature (which is the variable measured) for each zone within the range of 22ºC to 24ºC (Figure 6-2).

Tables 6-33, 6-34 and 6-35 show a breakdown of the hours in which each zone is occupied over a year in hours with comfort and hours with discomfort, either in overheating or in undercooling. For the Zoning systems with (6-34) and without power reduction (6-35), practically the same levels of comfort are obtained according to the calculation hypotheses. The slight differences are due to the machine being different and, therefore, the operational conditions being instantaneous (sensitive and latent fractions, etc.).

An improved level of comfort with the Zoning system is noted. For this system, hours in discomfort are essentially due to the time taken for the system to start up in a previously unoccupied zone (Figure 6-32). For the office, which is only occupied for a few hours, this factors holds more weight in the result, for which reason it gives the highest percentage of hours in discomfort.

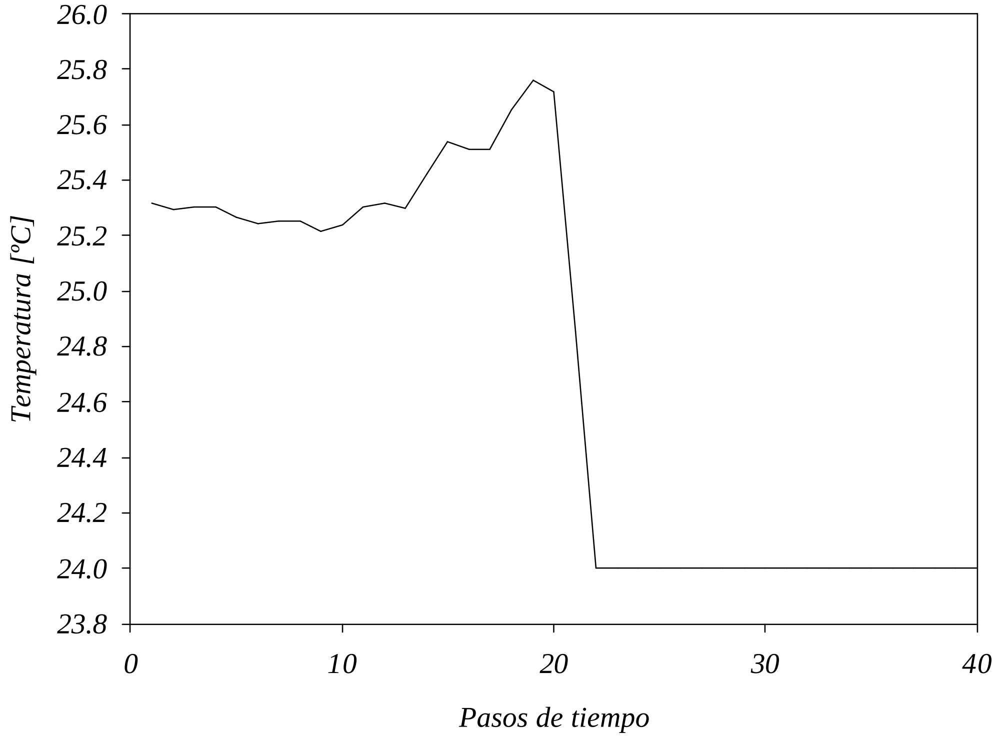


Figure 6-32: Example of starting up the system for a zone in the Zoning system: in the time passage in which the machine is connected up, the average temperature is not the same as the set temperature.

In the non-Zoning system, the living room is always maintained in comfort due to the fact that it is targeted by the machine's control and the fact that the house is always occupied, so that there are no disconnections. If there were any, there would be hours in discomfort due to start up (Figure 6-32).

NON-ZONING SYSTEM					
TOWN/CITY	**ZONE**	**HOURS IN COMFORT (%)**	**HOURS IN DISCOMFORT (%)**	**HOURS IN OVERHEATING (%)**	**HOURS IN UNDERCOOLING (%)**
MADRID	LIVING ROOM	5383.75 (100 %)	0.00 (0 %)	0.00 (0 %)	0.00 (0 %)
	KITCHEN	2006.75 (40.73 %)	2920.75 (59.27 %)	99.50 (2.02 %)	2821.25 (57.25 %)
	OFFICE	474.75 (43.36 %)	620.25 (56.64 %)	0.00 (0 %)	620.25 (56.64 %)
	PARENTS' B'ROOM	4430.25 (67.43 %)	2139.75 (32.57 %)	1283.75 (19.54 %)	856.00 (13.03 %)
	CHILDREN'S B`ROOM	2814.75 (68.55 %)	1291.50 (31.45 %)	345.75 (8.42 %)	945.75 (23.03 %)
BARCELONA	LIVING ROOM	5383.75 (100 %)	0.00 (0.00 %)	0.00 (0.00 %)	0.00 (0.00 %)
	KITCHEN	2082.00 (42.25 %)	2845.50 (57.75 %)	145.00 (2.94 %)	2700.50 (54.81 %)
	OFFICE	410.25 (37.47 %)	684.75 (62.53 %)	121.25 (11.07 %)	563.50 (51.46 %)
	PARENTS' B'ROOM	4225.50 (64.32 %)	2344.50 (35.68 %)	1427.25 (21.72 %)	917.25 (13.96 %)
	CHILDREN'S B`ROOM	1334.25 (32.49 %)	2772.00 (67.51 %)	827.75 (20.16 %)	1944.25 (47.35 %)
MALAGA	LIVING ROOM	5383.25 (99.99 %)	0.50 (0.01 %)	0.50 (0.01 %)	0.00 (0.00%)
	KITCHEN	2678.00 (54.35 %)	2249.50 (45.65 %)	171.75 (3.49 %)	2077.75 (42.17 %)

	OFFICE	510.00 (46.58 %)	585.00 (53.42 %)	193.00 (17.62 %)	392.00 (35.80 %)
	PARENTS' B'ROOM	4319.50 (65.75 %)	2250.50 (34.25 %)	1861.50 (28.33 %)	389.00 (5.92 %)
	CHILDREN'S B`ROOM	1578.25 (38.44 %)	2528.00 (61.56 %)	1127.75 (27.46 %)	1400.25 (34.10 %)
VALENCIA	LIVING ROOM	5383.75 (100.00 %)	0.00 (0.00 %)	0.00 (0.00 %)	0.00 (0.00 %)
	KITCHEN	2455.00 (49.82 %)	2472.50 (50.18 %)	174.00 (3.53 %)	2298.50 (46.65 %)
	OFFICE	432.25 (39.47 %)	662.75 (60.53 %)	201.50 (18.40 %)	461.25 (42.13 %)
	PARENTS' B'ROOM	4432.75 (67.47 %)	2137.25 (32.53 %)	1730.50 (26.34 %)	406.75 (6.19 %)
	CHILDREN'S B`ROOM	1428.50 (34.79 %)	2677.75 (65.21 %)	1081.25 (26.33 %)	1596.50 (38.88 %)
SEVILLE	LIVING ROOM	5383.75 (100.00 %)	0.00 (0.00 %)	0.00 (0.00 %)	0.00 (0.00 %)
	KITCHEN	2623.00 (53.23 %)	2304.50 (46.77 %)	179.00 (3.63 %)	2125.50 (43.14 %)
	OFFICE	431.25 (39.38 %)	663.75 (60.62 %)	239.25 (21.85 %)	424.50 (38.77 %)
	PARENTS' B'ROOM	4034.00 (61.40 %)	2536.00 (38.60 %)	2024.50 (30.81 %)	511.50 (7.79 %)
	CHILDREN'S B`ROOM	1290.00 (31.42 %)	2816.25 (68.58 %)	1310.75 (31.92 %)	1505.50 (36.66 %)

Table 6-33: Condiciones interiores para el NON-ZONING SYSTEM

S R P Z O N I N G S Y S T E M					
TOWN/CITY	**ZONE**	**HOURS IN COMFORT (%)**	**HOURS IN DISCOMFORT (%)**	**HOURS IN OVERHEATING (%)**	**HOURS IN UNDERCOOLING (%)**
MADRID	LIVING ROOM	5236.75 (97.27 %)	147.00 (2.73 %)	41.50 (0.77 %)	105.50 (1.96 %)
	KITCHEN	4720.00 (95.79 %)	207.50 (4.21 %)	17.00 (0.35 %)	190.50 (3.87 %)
	OFFICE	982.50 (89.73 %)	112.50 (10.27 %)	23.50 (2.15 %)	89.00 (8.13 %)
	PARENTS' B'ROOM	6398.75 (97.39 %)	171.25 (2.61 %)	84.00 (1.28 %)	87.25 (1.33 %)
	CHILDREN'S B`ROOM	3979.50 (96.91 %)	126.75 (3.09 %)	33.75 (0.82 %)	93.00 (2.26 %)
BARCELONA	LIVING ROOM	5243.75 (97.40 %)	140.00 (2.60 %)	44.00 (0.82 %)	96.00 (1.78 %)
	KITCHEN	4744.50 (96.29 %)	183.00 (3.71 %)	16.50 (0.33 %)	166.50 (3.38 %)
	OFFICE	983.50 (89.82 %)	111.50 (10.18 %)	24.50 (2.24 %)	87.00 (7.95 %)
	PARENTS' B'ROOM	6327.00 (96.30 %)	243.00 (3.70 %)	150.50 (2.29 %)	92.50 (1.41 %)
	CHILDREN'S B`ROOM	3984.50 (97.04 %)	121.75 (2.96 %)	32.25 (0.79 %)	89.50 (2.18 %)
MALAGA	LIVING ROOM	5251.50 (97.54 %)	132.25 (2.46 %)	58.50 (1.09 %)	73.75 (1.37 %)
	KITCHEN	4812.00 (97.66 %)	115.50 (2.34 %)	33.75 (0.68 %)	81.75 (1.66 %)
	OFFICE	1016.75 (92.85 %)	78.25 (7.15 %)	32.50 (2.97 %)	45.75 (4.18 %)
	PARENTS' B'ROOM	6223.75 (94.73 %)	346.25 (5.27 %)	304.00 (4.63 %)	42.25 (0.64 %)

	CHILDRE N'S B`ROOM	4017.50 (97.84 %)	88.75 (2.16 %)	51.00 (1.24 %)	37.75 (0.92 %)
VALENCIA	LIVING ROOM	5247.00 (97.46 %)	136.75 (2.54 %)	55.00 (1.02 %)	81.75 (1.52 %)
	KITCHEN	4786.25 (97.13 %)	141.25 (2.87 %)	34.50 (0.70 %)	106.75 (2.17 %)
	OFFICE	1017.75 (92.95 %)	77.25 (7.05 %)	31.50 (2.88 %)	45.75 (4.18 %)
	PARENTS' B'ROOM	6235.25 (94.90 %)	334.75 (5.10 %)	285.75 (4.35 %)	49.00 (0.75 %)
	CHILDRE N'S B`ROOM	4007.75 (97.60 %)	98.50 (2.40 %)	57.50 (1.40 %)	41.00 (1.00 %)
SEVILLE	LIVING ROOM	5242.00 (97.37 %)	141.75 (2.63 %)	64.50 (1.20 %)	77.25 (1.43 %)
	KITCHEN	4819.25 (97.80 %)	108.25 (2.20 %)	44.00 (0.89 %)	64.25 (1.30 %)
	OFFICE	1009.75 (92.21 %)	85.25 (7.79 %)	36.50 (3.33 %)	48.75 (4.45 %)
	PARENTS' B'ROOM	6236.25 (94.92 %)	333.75 (5.08 %)	2910 (4.43 %)	42.75 (0.65 %)
	CHILDRE N'S B`ROOM	4003.25 (97.49 %)	103.00 (2.51 %)	65.50 (1.60 %)	37.50 (0.91 %)

Table 6-34: Condiciones interiores para el ZONING SYSTEM sin reducción de potencia

C R P Z O N I N G S Y S T E M					
TOWN/CITY	**ZONE**	**HOURS IN COMFORT (%)**	**HOURS IN DISCOMFORT (%)**	**HOURS IN OVERHEATING (%)**	**HOURS IN UNDERCOOLIN G (%)**
MADRID	LIVING ROOM	5236.00 (97.26 %)	147.75 (2.74 %)	41.50 (0.77 %)	106.25 (1.97 %)
	KITCHEN	4718.50 (95.76 %)	209.00 (4.24 %)	17.00 (0.34 %)	192.00 (3.90 %)

	OFFICE	982.50 (89.73 %)	112.50 (10.27 %)	23.50 (2.15 %)	89.00 (8.12 %)
	PARENTS' B'ROOM	6397.75 (97.38 %)	172.25 (2.62 %)	84.00 (1.28 %)	88.25 (1.34 %)
	CHILDREN'S B`ROOM	3979.50 (96.91 %)	126.75 (3.09 %)	33.75 (0.81 %)	93.00 (2.26 %)
BARCELONA	LIVING ROOM	5243.25 (97.39 %)	140.50 (2.61 %)	44.50 (0.83 %)	96.00 (1.78 %)
	KITCHEN	4744.00 (96.28 %)	183.50 (3.72 %)	17.00 (0.35 %)	166.5 (3.38 %)
	OFFICE	983.00 (89.77 %)	112.00 (10.23 %)	25.00 (2.28 %)	87.00 (7.95 %)
	PARENTS' B'ROOM	6327.00 (96.30 %)	243.00 (3.70 %)	150.50 (2.29 %)	92.50 (1.41 %)
	CHILDREN'S B`ROOM	3984.50 (97.40 %)	121.75 (2.96 %)	32.25 (0.78 %)	89.50 (2.18 %)
MALAGA	LIVING ROOM	5247.50 (97.47 %)	136.25 (2.53 %)	62.50 (1.16 %)	72.75 (1.37 %)
	KITCHEN	4808.00 (97.57 %)	119.50 (2.43 %)	37.75 (0.77 %)	81.75 (1.66 %)
	OFFICE	1015.75 (92.76 %)	79.25 (7.24 %)	33.50 (3.06 %)	45.75 (4.18 %)
	PARENTS' B'ROOM	6221.00 (94.69 %)	349.00 (5.31 %)	306.75 (4.67 %)	42.25 (0.64 %)
	CHILDREN'S B`ROOM	4017.50 (97.84 %)	88.75 (2.16 %)	51.00 (1.24 %)	37.75 (0.92 %)
VALENCIA	LIVING ROOM	5241.50 (97.36 %)	142.25 (2.64 %)	60.50 (1.12 %)	81.75 (1.52 %)
	KITCHEN	4781.25 (97.03 %)	146.25 (2.97 %)	39.50 (0.80 %)	106.75 (2.17 %)
	OFFICE	1013.75 (92.58 %)	81.25 (7.42 %)	35.50 (3.24 %)	45.75 (4.18 %)
	PARENTS' B'ROOM	6234.00 (94.89 %)	336.00 (5.11 %)	287.00 (4.37 %)	49.00 (0.74 %)

	CHILDREN'S B`ROOM	4007.75 (97.60 %)	98.50 (2.40 %)	57.50 (1.40 %)	41.00 (1.00 %)
SEVILLE	LIVING ROOM	5153.50 (95.72 %)	230.25 (4.28 %)	153.00 (2.83 %)	77.25 (1.43 %)
	KITCHEN	4738.50 (96.16 %)	189.00 (3.84 %)	124.75 (2.53 %)	64.25 (1.30 %)
	OFFICE	1002.25 (91.53 %)	92.75 (8.47 %)	44.00 (4.02 %)	48.75 (4.45 %)
	PARENTS' B'ROOM	6160.25 (93.76 %)	409.75 (6.24 %)	367.00 (5.59 %)	42.75 (0.65 %)
	CHILDREN'S B`ROOM	4003.25 (97.49 %)	103.00 (2.51 %)	65.50 (1.60 %)	37.50 (0.91 %)

Table 6-35: Interior conditions for the Zoning system with power reduction

6.3.2 *Electrical consumption*

The following tables show the annual results on termal demand, COP and electrical consumption for the non-Zoning systems, Zoning without power reduction (ZWOPR) and Zoning with power reduction (ZWPR), for all-nothing machines (Table 6-36), model §3.4.1; inverter-1 (Table 6-37), model §3.4.2.1; inverter-2 (Table 6-38), model §3.4.2.2.

At this point, the basic hypotheses for modelling each system should be borne in mind as they affect the interpretation of the results:

a) All-nothing machine: COP calculated by applying the function proposed by DOE, function of the part load factor (PLF). The base value used to lower with this function is that given in the manufacturer`s catalogue for evaporation and condensation conditions in which the machine is working. The COP is also applied for ventilator consumption.

b) Basic model with the inverter system. Beyond the rang of the evaporation and condensation conditions shown in the manufacturer's catalogue, the machine's performance is assumed to be constant for compressor consumption. Variation in the COP is not considered at part load. In the region in which the power demand is higher than the minimum set for the machine, nominal consumption is allotted to the ventilator. Below this energy demand in the time passage, it is calculated proportionally to the time in which it should operate.

c) Model-1 of the inverter system: ventilator consumption is considered in the COP calculation as a whole. This is adjusted for the operating conditions based on the catalogue data in the range set for the inverter machine. The COP at part load, when energy demand in a time passage is less than the minimum that the machine would supply if it were operating throughout the time passage, is adjusted according to the function proposed by DOE, calculated on the part load factor (PLF).

In summary, each model is distinguished in the consideration of the part load being applied or not to the DOE function and in the manner in which ventilator consumption in the interior unit is considered.

Table 6-36: Results for machines with all-nothing control

TOWN/CITY	SYSTEM	HEAT			COLD			COSTS			
		THERMAL (kWh)	ELECTRICAL (kWh)	COP	THERMAL (kWh)	ELECTRICAL (kWh)	COP	HEAT (€)	COLD (€)	TOTAL (€)	SAVING (€)
MADRID	Non-Zoning	16205.40	6084.85	2.66	8164.05	2459.52	3.32	669.33	270.55	939.88	0
	SRP Zoning	11658.47	4918.29	2.37	7148.15	2187.29	3.27	541.01	240.60	781.61	158.27 (16.84%)
	CRP Zoning	11658.47	4082.91	2.85	6824.62	2307.95	3.12	449.12	253.87	703.00	236.88 (25.20%)
BARCELONA	Non-Zoning	9639.42	3649.11	2.64	7963.30	2307.84	3.45	401.40	253.86	655.26	0
	SRP Zoning	6997.64	2951.36	2.37	6776.64	2034.16	3.33	324.65	223.76	548.41	106.85 (16.30%)
	CRP Zoning	6997.64	2474.36	2.83	6483.89	2149.74	3.02	272.18	236.47	508.65	146.61 (22.37%)
MALAGA	Non-Zoning	5607.79	2140.78	2.62	10981.89	3243.38	3.39	235.48	356.77	592.25	0
	SRP Zoning	3919.71	1660.05	2.36	9503.14	2885.98	3.29	182.60	317.46	500.06	92.19 (15.57%)
	CRP Zoning	3919.71	1396.15	2.81	9076.15	3046.44	2.98	184.60	335.44	520.04	72.21 (12.19%)
VALENCIA	Non-Zoning	6621.57	2534.83	2.61	10727.40	3132.93	3.42	278.83	344.62	623.45	0
	SRP Zoning	4709.15	1999.69	2.35	8941.39	2708.97	3.30	219.96	298.00	517.96	105.49 (16.92%)
	CRP Zoning	4709.15	1679.54	2.80	8542.13	2859.59	2.99	184.75	314.55	499.30	124.15 (19.91%)
SEVILLE	Non-Zoning	7011.14	2658.13	2.64	13501.07	4170.94	3.24	292.39	458.80	751.19	0
	SRP Zoning	4985.81	2105.62	2.37	12390.67	3889.14	3.19	231.62	427.80	659.42	91.77 (12.22%)
	CRP Zoning	4985.81	1766.00	2.82	11656.74	4035.31	2.89	194.26	443.88	638.14	113.05 (15.05%)

Table 6-37: Resultados para equipos con control INVERTER (modelo 1)

TOWN/CITY	SYSTEM	HEAT			COLD			COSTS			
		THERMAL (kWh)	ELECTRICAL (kWh)	COP	THERMAL (kWh)	ELECTRICAL (kWh)	COP	HEAT (€)	COLD (€)	TOTAL (€)	SAVING (€)
MADRID	Non-Zoning	16205.40	3435.40	4.72	8164.05	1668.31	4.89	377.89	183.51	561.4	0.00
	SRP Zoning	11658.47	2757.10	4.22	7148.15	1312.52	5.44	303.28	144.38	447.66	113.74 (20.26 %)
	CRP Zoning	11658.47	2646.26	4.40	6824.62	1356.80	5.03	291.09	149.24	440.33	121.07 (21.57%)
BARCELONA	Non-Zoning	9639.42	2023.01	4.76	7963.30	1452.59	5.48	222.53	159.78	382.31	0.00
	SRP Zoning	6997.64	1648.25	4.24	6776.64	1172.60	5.78	181.31	129.00	310.31	72.00 (18.83 %)
	CRP Zoning	6997.64	1454.04	4.81	6483.90	1201.85	5.40	159.94	132.20	292.14	90.17 (23.58%)
MALAGA	Non-Zoning	5607.79	1191.37	4.71	10981.89	2134.72	5.14	131.05	234.82	365.87	0.00
	SRP Zoning	3919.71	933.49	4.20	9503.14	1706.90	5.57	102.68	187.76	290.44	75.43 (20.62%)
	CRP Zoning	3919.71	810.03	4.84	9076.15	1772.60	5.12	89.10	194.98	284.08	81.79 (22.35 %)
VALENCIA	Non-Zoning	6621.60	1409.52	4.70	10727.40	2073.30	5.17	155.05	228.06	383.11	0.00
	SRP Zoning	4709.15	1122.29	4.20	8941.39	1599.60	5.59	123.45	175.95	299.4	83.71 (21.85%)
	CRP Zoning	4709.15	977.23	4.82	8542.13	1657.09	5.15	107.50	182.28	289.78	93.33 (24.36%)
SEVILLE	Non-Zoning	7011.14	1476.00	4.75	13501.07	3004.30	4.49	162.36	330.47	492.83	0.00
	SRP Zoning	4985.81	1177.75	4.23	12390.67	2653.00	4.67	129.55	291.83	421.38	71.45 (14.50%)
	CRP Zoning	4985.81	1037.31	4.81	11656.74	2731.17	4.27	114.10	300.43	414.53	78.3 (15.89%)

Table 6-38: Resultados para equipos con control INVERTER (model-2)

TOWN/CITY	SYSTEM	HEAT			COLD			COSTS			
		THERMAL (kWh)	ELECTRICAL (kWh)	COP	THERMAL (kWh)	ELECTRICAL (kWh)	COP	HEAT (€)	COLD (€)	TOTAL (€)	SAVING (€)
MADRID	Non-Zoning	16205.40	6912.75	2.34	8164.05	3042.23	2.68	760.40	334.64	1095.04	0.00
	SRP Zoning	11658.50	5536.33	2.10	7148.14	2813.93	2.54	609.00	309.53	918.53	176.51 (16.12%)
	CRP Zoning	11658.50	4681.53	2.49	6824.64	2735.87	2.49	514.97	300.94	815.91	279.13 (24.49%)
BARCELONA	Non-Zoning	9639.42	4223.04	2.28	7963.30	2927.83	2.72	464.53	322.06	786.59	0.00
	SRP Zoning	6997.65	3325.51	2.10	6776.64	2662.63	2.54	365.80	292.89	658.69	127.90 (16.26%)
	CRP Zoning	6997.65	2974.40	2.35	6484.60	2623.94	2.47	327.18	288.63	615.81	170.78 (21.71%)
MALAGA	Non-Zoning	5607.79	2465.45	2.27	10981.89	4045.36	2.71	271.20	445.00	716.20	0.00
	SRP Zoning	3919.72	1856.16	2.11	9503.13	3734.05	2.54	204.18	410.75	614.93	101.27 (14.14%)
	CRP Zoning	3919.72	1704.75	2.30	9081.25	3612.78	2.51	187.52	397.40	584.92	131.28 (18.33%)
VALENCIA	Non-Zoning	6621.57	2925.53	2.26	10727.4	3899.87	2.75	321.81	429.00	750.81	0.00
	SRP Zoning	4709.15	2242.90	2.10	8941.39	3511.21	2.55	246.72	386.23	632.95	117.86 (15.70%)
	CRP Zoning	4709.15	2045.22	2.30	8547.00	3406.54	2.51	225.00	374.72	599.72	151.09 (20.12%)
SEVILLE	Non-Zoning	7011.14	3070.98	2.28	13501.07	5019.00	2.69	337.81	552.09	889.90	0.00
	SRP Zoning	4985.82	2366.04	2.10	12390.63	4777.80	2.59	260.26	525.56	785.82	104.08 (11.69%)
	CRP Zoning	4985.82	2131.03	2.34	11657.12	4564.07	2.55	234.41	502.05	736.46	153.44 (17.24%)

In the light of the results, the following conclusions may be drawn:

1.- Savings (economic or in electrical consumption), in absolute terms, should be analyzed along with the results on comfort obtained for each system (§6.3.1): unlike in the non-Zoning system, the Zoning system complies with the settings for all the occupied zones.

2.- In all cases and towns and cities studied, it is the Zoning system that consumes less electrical energy than the non-Zoning system. There is a typical saving rate of 20%, this being higher in the heating season for the reasons explained above, when analyzing Figures 6-24 and 6-25. Consequently, greater savings occur in the towns and cities studied where the harshest winters are noted. Economic saving in relative terms ranges from 12.2% to 25.2%. The maximum relative saving occurs with the all-nothing machine used in a Zoning system with power reduction. The least relative saving is recorded in the all-nothing system with power reduction used in Malaga, where demand for acclimatization is less and the machine operation conditions are more favourable. It is clear that the more demanding conditions of use (due to demand or the climate), the more interesting is the Zoning system in terms of electrical consumption. Evidently, the interest in improving comfort is always present.

3.- As far as machine performance is concerned, it is noted how the COP in the non-Zoning system is always better than in the Zoning system with no power reduction. This is due to the fact that the by-pass worsens the machine's operational conditions. Such a comparison should be carried out on identical machines, i.e., for the Zoning system with no power reduction (ZSWNPR). Air recirculation makes the temperature accessing the interior battery increase in winter and decrease in summer. The Zoning system with power reduction uses a different machine, with a better nominal COP for heating and worse for cooling.

4.- The difference in absolute terms is highly significant between the different approximations in the modelling used. This is not the case in relative terms. The inverter system, according to Model-1, in which ventilator consumption is calculated from the COP for the machine jointly, obtains greater annual COP than those found in the other models (both in the inverter machine and in the all-nothing). In cooling, the COPs for model-1 are around 5, and for heating, they are slightly less, but higher, in many cases, than 4.5. According to the current level of knowledge, it is not possible to state which of these models is closest to reality. *A priori*, it appears that model-1 introduces reasonable improvements in terms of the basic model, which make it more credible. Nonetheless, in terms of the results and in view of the fact that the COPs obtained are very different, it would appear to be necessary to conduct an experimental check on the result.

In closing, the lines to be dealt with in order to consolidate the results obtained in this work are given below:

1.- Improve the by-pass model to take into account the real imbalance occurring when the dampers are activated.

2.- Improve the models, both for the all-nothing machine and the inverter at part load, including the consumption due to start-up and stop. Furthermore, it would be advisable to extend the performance data on evaporation and condensation conditions to a wider range than that supplied by the manufacturers.