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Analysis of a HVAC zoning control system with an air-to-water heat pump and a ducted fan coil unit in residential buildings

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ABSTRACT

This paper presents an analysis of a thermal zoning system integrated in a Heating, Ventilation and Air-Conditioning (HVAC) system based on an air-to-water heat pump with a ducted fan coil. Zoned systems are based on independently controlling the temperature of each of the zones of a building. When a zone is not occupied or not in demand, the control board sends a control signal to the zone's motorised damper which interrupts the airflow supply to that room. Although this control system is gaining popularity in the residential sector, the results obtained in terms of thermal comfort and energy consumptions are not evident and should be documented.

Besides, the control strategy is based on an algorithm that allows acting on the heat pump, setting the set-point temperature, and on the fan-coil, setting the fan speed. Based on this, it is possible to design an algorithm to optimize the performance of the installation ensuring thermal comfort and achieving energy savings.

The thermal zoning and the HVAC control system are modelled and simulated in Trnsys17 for the case of study of a residential dwelling, compared with two different configurations: a non-zoned ducted fan coil and individual fan coils. Important benefits are obtained in the evaluation of thermal comfort, with higher values of PPD in all zones. From the point of view of energy consumption, the influence of the thermal zoning on the performance of the heat pump reports important energy savings. Finally, an economic analysis results in payback periods lower than 4.9 years.

1. Introduction.

Nowadays, in European countries like Spain, France, or Italy among others, Air-to-Water Heat Pumps (AWHP) are widely used as a key technology in the residential sector as the main option for covering the domestic hot water (DHW) and the heating and air conditioning needs, replacing solar thermal collectors and conventional direct expansion HVAC systems, respectively [1,2,3]. The main reasons are the high thermal performance of this type of heating and cooling source and the capacity of meeting the minimum renewable energy contribution established by the Directive 2009/28/EC of the European Parliament and of the Council [4]. As an example, air source heat pumps (ASHP) are a direct alternative to existing boiler installations for heating in UK [5]. In a wider perspective, Madonna et al. [6] analyzed the influence of the climate, settings and building characteristics in the performance of reversible AWHPs in small residential buildings in Italy. The article shows that a weather compensation strategy allows to improve the seasonal performance up to 23% and the annual performance up to 19%. But not only in Europe aerothermal energy is gaining popularity. Asaee et al. [7] proposed a techno-economic impact of AWHP systems on the energy consumption of the Canadian housing stock for space and DHW heating purposes. The results indicate that about 71% of houses are eligible for the AWHP retrofit, with a reduction of 36% in end-use energy consumption.

The AWHP can be combined with different terminal units: FCs (fan coils), radiant floor or radiators. Actually, the traditional tendency of using a ducted direct expansion inverter system based on the on/off control of a single zone [8] in the residential sector has been replacing by other terminal units used with the heat pumps, as the fan coils (FC). Taking advantage of the potential of the heat pump, the HVAC configuration composed by a heat pump with a ducted fan coil (NZonFC) is considering as one of the first alternatives. This terminal unit operates similarly to an inverter unit but with an ASHP as the heating and cooling

Nomenclature

Variables

C_c	Stan-by degradation coefficient [–]
C_d	Degradation coefficient [–]
C_p	Specific heat [J/kgK]
COP	Coefficient of Performance [–]
COP_{dec}	Nominal COP of the heat pump [–]
COP_{pl}	Current COP of the heat pump [–]
D_{zi}	Damper position (1 open, 0 close) of the zone I [–]
FC	Fan coil [–]
HPT	Heat pump set point temperature [C]
IC	Initial cost [€]
m	Mass flow rate [kg/s]
N_{Zones}	Number of zones of the building [–]
$N_{Zones,dem}$	Number of zones in demand [–]
OC	Operating cost [€]
PB	Payback period [years]
PLF	Partial Load Factor [–]
PLR	Partial Load Ratio [–]
P_{hp}	Electric power consumption of the heat pump [W]
P_{max}	Nominal power consumption of the circulating pump [W]
$P_{nom,NZon}$	Nominal capacity of the heat pump in a non-zoned system [W]
$P_{nom,Zon}$	Nominal capacity of the heat pump in a zoned system [W]
$P_{ocup,zone\ i}$ (t)	Occupational profile value of zone i (1 occupied or 0 not occupied) [–]
PPD	Predicted Percentage of Dissatisfied [%]
P_{pump}	Electric power consumption of the circulating pump [W]
P_{vel}	Weight of the zones [–]
Q	Heat exchanged in the fan coil between the water and the air [W]
$Q_{capacity}$	Current capacity of the heat pump [W]
Q_{cool}	Cooling load [kW]

$Q_{dem,zone\ i}$ (t)	Thermal load of the zone i in each time step t. [W]
Q_{heat}	Heating load [kW]
$Q_{load,hp}$	Building's thermal demand [W]
Q_{max}	Maximum heat which the minimum fluid can transfer [W]
$Q_{r,s}$	Sensible load met by the fan coil [W]
$T_{a,out}$	Supply air temperature from the fan coil [C]
$T_{min,HPT}$	Minimum HPT [C]
$T_{max,HPT}$	Maximum HPT [C]
$T_{min,out}$	Minimum outside air temperature [C]
$T_{max,out}$	Maximum outside air temperature [C]
T_{out}	Outside air temperature [C]
T_{ret}	Return air temperature from the zone/s to the fan coil [C]
T_{setzi}	Set point temperature of the zone I [C]
$T_{w,in}$	Inlet water temperature of the heat pump [C]
$T_{w,set}$	Set point water temperature of the heat pump [C]
T_{zi}	Air temperature of the zone I [C]
$T_{zi\ (t-1)}$	Air temperature of the zone i at the previous time step [C]
ΔT_{max}	Maximum temperature difference between zone air temperature and set point temperature of zone i [C]
$\Delta T_{z,i}$	Temperature difference between zone air temperature and set point temperature of master zone in cooling mode [C]
U	Overall heat transfer coefficient [W/m^2 K]

Greek letters

ε	Fan coil effectiveness [–]
γ	Circulating pump control signal [–]

Subscripts

a	Air
I	Inlet
o	Outlet
w	Water
z	Zone

source. In both cases, regarding the control system, this kind of HVAC equipment conventionally ensures the comfort level in the zone where the thermostat is placed but, regarding the rest of the zones, if the load profile is different from that of the control zone (use, orientation, thermal loads, etc.), their temperatures can fall outside the comfort range.

As a solution, generally, the most common controllers in buildings with FCs are those based on hysteresis and proportional-integral-derivative (PID) due to their practical feasibility, but they focus only on indoor environment conditioning rather than efficient control strategies. The new Energy Performance of Buildings Directive (EPDB) 2018/844 of the European Parliament [9] promotes the use of HVAC control systems in buildings with the aim of optimizing the energy management and raising end-user's awareness about energy consumption. With this aim, an increasing number of developments of more sophisticated controllers has been studied: the model predictive controls (MPC), fuzzy logic (FL), neural networks (NN), etc. Controllers should not only be applied individually to each element of the installation (heat pump, FC, valves, pumps, etc.) but they should also control the performance of the whole system.

In relation to the heat pump, a literature review reflects high efforts to improve its performance from different aspects. Particularly, a great deal of research has been done on employing different controlling approaches. Underwood [10] modelled a fuzzy controller to regulate both plant capacity and evaporator superheat degree using a variable speed compressor drive and obtaining a reduced cycling and a reduction in energy consumption of 20%. Wang et al. [11] proposed a MPC strategy for an ASHP heating system optimizing compressor frequency and water mass flow rate in real-time, which can lead to the optimized real-time

variable water temperature difference. Kudela et al. [12] described different algorithms for a heat pump control: equithermal regulation, a binary algorithm for temperature source selection, a predictive algorithm for the heat storage discharging, and an algorithm for deferred heat storage discharging, with a potential increasing of SCOP of 5.19%. In this framework, all the authors confirm that the heat pump supply water temperature is a key parameter to improve the energy performance of the heat pump. Sun et al. [13] showed how the COP of an ASHP would decrease by 2–4% when the supply water temperature was increased by 1 °C. Therefore, the regulation of the supply water temperature according to the building heating/cooling demand would be essential with variable water temperature control method. Several authors used meteorological parameters to adjust the water temperature. Oldewurtel et al. [14] introduced a simple approximation based on the outdoor dry-bulb temperature, wet-bulb temperature, and solar radiation in a climate compensation model for supply water temperature using a stochastic prediction method. A more complex method was developed by Lee et al. [15]. They presented a hybrid optimization method to find the optimal set point of supply water temperature based on a minimum energy consumption objective. In both cases, the complexity of the system and the difficulty to accurately obtain a large number of parameters were crucial problems addressed by Sun et al. [16] by means of an adaptive control method for supply water temperature of a ASHP in real time, reducing the power consumption of the ASHP unit by 38.20%. Despite this, in conventional installations with heat pumps the water temperature is fixed according to the terminal unit or some heating and cooling curves are adopted to control the water temperature depending only on the measured outdoor air temperature.

Concerning the FCs, hysteresis and PID controllers are still a common practice for temperature control in buildings, but many studies proposed different energy saving methods and optimization strategies to improve their performance. Tianyi et al. [17] modelled a duty ratio fuzzy control method (DRFCM) on a FC installation with a functioning of three-stage speed, to perform the control of the water valve and air volume. Edwards et al. [18] presented a novel generalised control strategy, requiring only minimal input data, to optimise fan speed in order to reduce daily energy consumption. Compared to fixed fan speed or to the benchmark fan speed control strategies energy savings up to 43% are achieved.

From the point of view of the HVAC control system, engineers face the challenge, in a retrofit context, of designing HVAC systems whose control system must be able to manage all the elements of the installation effectively, even when the manufacturers of these elements are different. In that sense, MPC applied in building management systems has been recognized as one of the most promising solutions to achieve considerable energy savings [19]. Afram et al. [20] applied a NN based MPC in the optimization of HVAC systems. They estimated the operating cost savings in the residential sector in Ontario (Canada) between 6% and 73% while maintaining the thermal comfort constraints. Chu et al. [21] proposed a least enthalpy estimator (LEE) that combines the definition of thermal comfort level and the theory of enthalpy into a load predicting way to provide timely suitable settings for a fan coil unit fuzzy controller used in HVAC. However, nowadays MPC is not generally implemented as the first option in the residential sector. It should be noted that, as the complexity of the system increases (number of sensors, computational costs, etc.), the cost of the installation could become unjustified. According to Stopps et al. [22], this is the reason why the sensing and actuation infrastructure in North American residential buildings has failed to evolve along with improved building control system methods. In a broader context, Yao et al. [23] described the practical difficulties of the MPC as real-time application of complex optimization techniques may result in longer computation time, problems with the stability of the controller, rejection capability of disturbances factors, setting time of MPC controller, lack of skilled and efficient experts, cost of retrofitting, etc. For all this, cycling on/off control strategies are the most prevalent [24]. Nevertheless, as mentioned before, they could not be adequate if only a single zone is controlled.

As a solution, a zoned control system implemented in the ducted fan coil (ZonFC) is presented in this paper as the first alternative to the conventional on/off control system. The standard EN 15232 [25] regulates the use of HVAC control systems in buildings including thermal zoning as a fundamental condition to ensure energy efficiency and thermal comfort. In other words, a zoned system is based on independently controlling the temperature of each of the zones of a building. To do this, a thermostat is installed in each room, allowing the thermal demand in each of the zones to be determined, and the selection of an independent set-point temperature depending on the preferences of the user. When the set-point temperature in a zone is reached, a control signal is sent to the zone's motorized damper which interrupts the air supply to that room. So, in this situation, a zoning HVAC control system provides the ability to control each zone temperature independently, keeping the dampers of the zones which are in demand open, and closing the zones which are not in demand or are not occupied. Therefore, thermal comfort in each zone of the building could be achieved.

Another existing option that also works with thermal zoning is the implementation of a HVAC configuration based on a heat pump with individual fan coils in each zone (IndFC). This configuration has the advantages of the ZonFC in terms of thermal zoning because each fan coil can control the zones' air temperatures of the building thanks to the thermostats installed in each room and the control management of each fan coil. The main differences with respect to the ducted FC are related, first to the maintenance of the installation, as it is necessary to take water pipes to each room through the ceiling and, second to a higher noise level in the zones.

Therefore, the paper presents the ZonFC as a new HVAC control system for an AWHP with a ducted fan coil unit, in the residential sector, compared with two typical fan coil unit typologies: the ducted fan coil without zoned control system (NZonFC) and individual fan coils (IndFC). Besides the thermal zoning, the ZonFC is based on another fundamental element: the communication gateway, a device that enables two-way communications between the control board and HVAC units, i.e. heat pumps and fan coils. This is only possible if the manufacturer shares the communication protocol and allows to read and set operational parameters of the unit, such as: operating mode, fan-coil unit's fan speed or water set-point temperature of the heat pump (HPT). From the information of the heat pump and fan coil parameters, and the thermal situation of each zone, an algorithm is designed to optimize the performance of the installation, ensuring thermal comfort, and achieving energy savings.

The thermal zoning and the control algorithm model are described in detail and modelled and simulated in Trnsys17 [26]. The case of study is a typical residential dwelling, placed in three different European cities. Results are compared with the cases of NZonFC and IndFC, according to different methods to set the water supply temperatures of the AWHP. The main performance evaluation indices are thermal comfort, based on the calculation of PPD and PMV values, and energy savings according to the energy consumption of the heat pump and the fan coils. Finally, an economic analysis is presented with the calculation of the payback period of the ZonFC with respect the other systems.

2. Systems description

The following scenarios are presented in this section:

- The non-zoned ducted fan coil system (NZonFC).
- The zoned ducted fan coil system (ZonFC).
- The individual fan coil system (IndFC).

2.1. Non-Zoned ducted fan coil system (NZonFC)

Hot or cold water is pumped from the heat pump and exchanges heat with the air supplied by the fan of the ducted fan coil. The air is supplied to the different zones of the building. Fig. 1 shows the scheme of a non-zoned ducted fan coil system. This is the control system considered as conventional in this paper.

The control system proposed is the option most common in the residential sector. It consists of a control board which acts over the heat pump selecting the operating mode (M): heating or cooling, and over the hydraulic circuit (the circulating pump (P) and the three-way mixing valve (V)) ensuring the water is supplied to the coil in the suitable conditions. The water set point temperature control could be fixed to 7 and 45 °C in cooling and heating mode, respectively, or it could apply a heating and cooling curve according to the outside air temperature. The three-way mixing valve avoids overheating or undercooling, adapting the water supply to the fan coil. Besides, the fan speed (F) is sequentially changed according to the temperature difference between the master zone air and set point temperature.

This conventional control system maintains the controlled zone temperature of the master zone in the comfort range, while other zone air temperatures depend on the evolution of its thermal loads. Consequently, it is appropriate for facilities with similar uses and thermal loads profile, but, as it can be seen in the results section, not for the residential buildings.

2.2. Zoned ducted fan coil system (ZonFC).

Fig. 2 shows the scheme of the zoned ducted fan coil system. The configuration is the same as the NZonFC, but with a different control
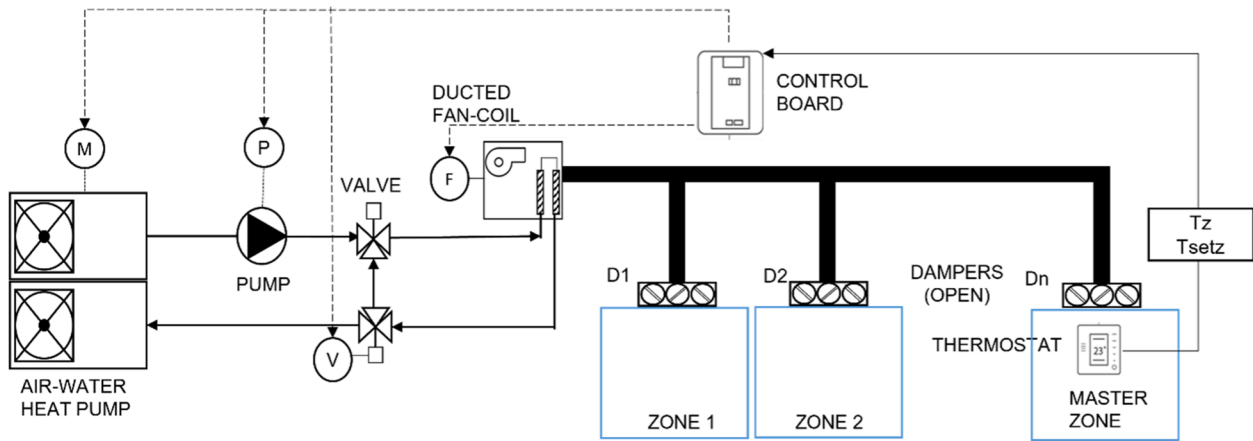
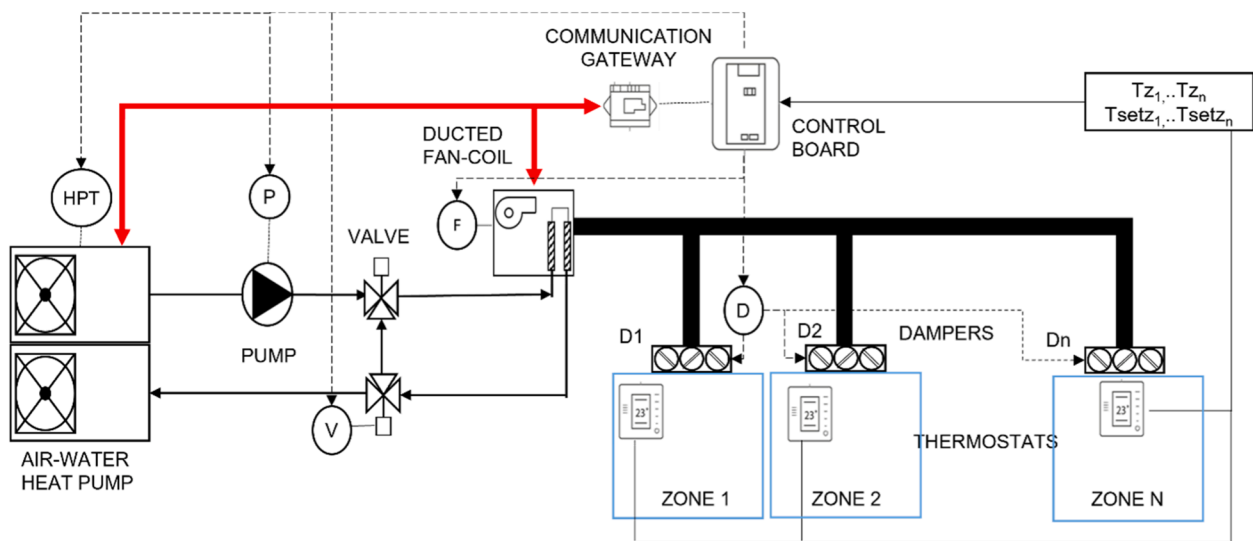**Fig. 1.** Scheme of the non-zoned ducted fan coil system.

Fig. 2. Scheme of the zoned ducted fan coil control system.

system. It consists of two elements: the control board and the communication gateway. The first one receives the information of the rooms: the air temperature (T_{z1}, T_{z2},..T_{zn}) and the set point temperature (T_{setZ1}, T_{setZ2},..,T_{setZn}) imposed by the users, from the thermostat placed in each zone. With this information, the algorithm imposes, thanks to the communication gateway, the control strategy with the configuration of the next elements:

- The fan speed of the fan-coil (F) is selected dynamically according to the airflow rate demanded by each zone, and the temperature difference between the zone and the set point air temperature.
- The water set-point temperature of the heat pump (HPT) is chosen based on the difference between the air and set-point temperatures in each zone and the number of zones in thermal demand.
- The position of the dampers (fully opened or closed) of each room which controls the air supplied to the zones (D_1, D_2,..,D_n).
- The operation signal of the circulating pump (P).
- The operation signal for the three-way mixing valve (V).

Compared to the NZonFC, the zoned control system provides the ability to control each zone temperature independently, keeping the dampers of the zones which are in demand open, and closing the zones which are not in demand or not occupied. Therefore, thermal comfort in each zone of the building could be achieved. Besides, the algorithm

receives the information about the thermal situation of each zone which is used to adjust the speed of the fan coil and the HPT with the aim of optimizing the performance of the system.

2.3. Individual fan coils system (IndFC)

The case of the individual fan coils can be also considered as a zoned system like the ZonFC because a fan coil is installed in each zone and can be controlled with its own thermostat to maintain the thermal comfort conditions. The control board conveniently selects the speed of the fan of each fan coil when comparing the zone air temperature with respect to the set point zone temperature. However, just as in the case of NZonFC, the system does not have the option of managing the hot or cold water temperature production, and or the HPT is fixed, setting to 7 °C and 45 °C for cooling and heating, respectively, or as in the case of NZonFC, a heating and cooling curve according to the outside air temperature is usually applied. Fig. 3 shows the scheme of the IndFC.

3. Mathematical modelling

New models of the heat pump, fan-coils and control systems are created and implemented in Trnsys, for a better adaptation to the requirements of the simulation conditions.
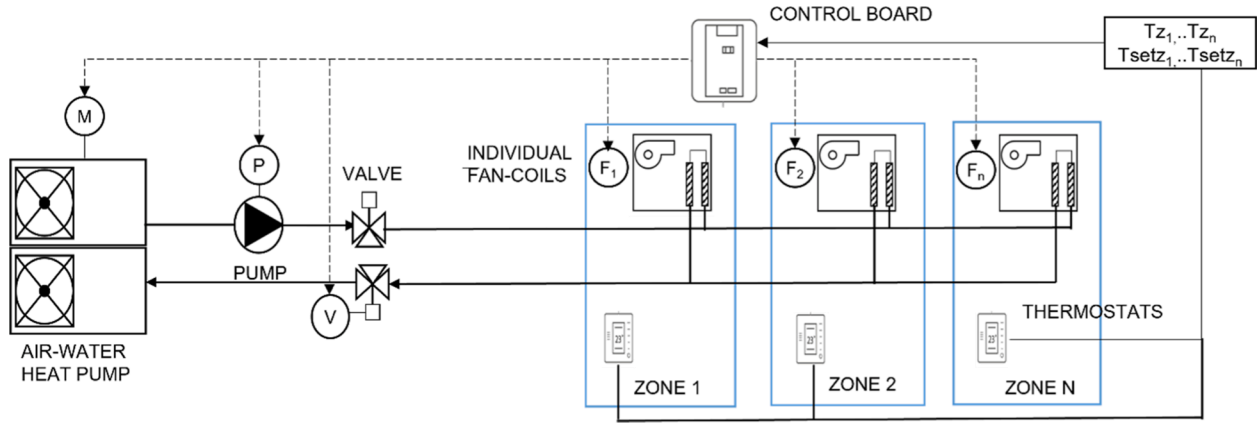
Fig. 3. Scheme of the individual fan coils control system.

3.1. Heat pump, fan coil and hydronic circuit model

The air-water heat pump model is developed based on the typical set of curves of a manufacturer [27,28]. These curves provide information about heating and cooling capacity ($Q_{capacity}$) and electric consumption (P_{cons}) as a function of outside air temperature (T_{out}) and the required HPT. The coefficient of performance (COP) and energy efficiency ratio (EER) are calculated from the capacity and the electric consumption. Fig. 4 shows the COP and EER performance curves of a heat pump unit used in the case of study.

The COP should be modified as a function of the part load ratio (PLR). The PLR is the ratio of the building's thermal demand to the current capacity of the heat pump, as following:

$$Q_{load,hp} = \dot{m}_w \bullet Cp_w \bullet \left(T_{w,set} - T_{w,in}\right) \tag{1}$$

$$PLR = \frac{Q_{load,hp}}{Q_{capacity}} \tag{2}$$

and, in addition, it is a function of the Part Load Fraction (PLF), which is defined as the relation between the current COP and the nominal COP of the equipment. The PLR depends on numerous factors related to its constructive details and the control strategies used to match load and capacity (compressor/s, pump/s and fan/s regulation) [29]. According to European Standards for the calculation of seasonal energy efficiency EN 14825 [30], the PLF is calculated as a function of the PLR and the parameter C_c, which includes the impact of the decrease of the energy efficiency of the device due to the cyclic operation and power consumption in the stand-by mode, resulting as follows:

$$PLF = \frac{COP_{pl}}{COP_{dec}} = \frac{PLR}{PLR \bullet C_c + (1 - C_c)} \tag{3}$$

The performed simulations have adopted the curves of PLF from the standard [30]. Although the C_c value should be consulted in the manufacturer datasheet, unfortunately, it is not provided. The standard often suggests an accepting value of $C_c = 0.9$.

The part load operation is a critical issue of research in the literature. Recently, Xu et al. [31] criticized the simple approximations used for standard methodologies. Also, Piechusrki et al. [32] analyzed experimentally the impact of the work under partial load on the energy efficiency of an air water heat pump and they obtained differences between the operational energy efficiency of the heat pump and the efficiency declared by the manufacturer in function of different values of the parameter C_c. In this case, the partial load operation is very important because residential heat pumps predominately operate at part load and the influence of the thermal zoning in the part load operation is interesting with respect to a non-zoned system. As thermal zoning saves energy by turning off non-occupied zone's dampers, it makes efficient use of the system by increasing the number of hours that the heat pump works under part load conditions.

Finally, the electric power consumption is obtained from the full-load (PLF) power (according to the PLR), and the COP$_{,dec}$ of the system:

$$P_{hp} = \frac{Q_{capacity}}{COP_{dec}} \bullet PLF \tag{4}$$

The fan-coil model is based on the effectiveness-NTU method, and it uses a constant value of effectiveness for every speed of the fan. The effectiveness can be calculated from the data given in the fan-coil catalogue [33].

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{m_w \bullet Cp_w \bullet \Delta T_w}{\min(m_a \bullet Cp_a, m_w \bullet Cp_w) \bullet \left(T_{w,i} - T_{ret}\right)} \tag{5}$$

where Q is the heat exchanged with the air and Q_{max} is the maximum
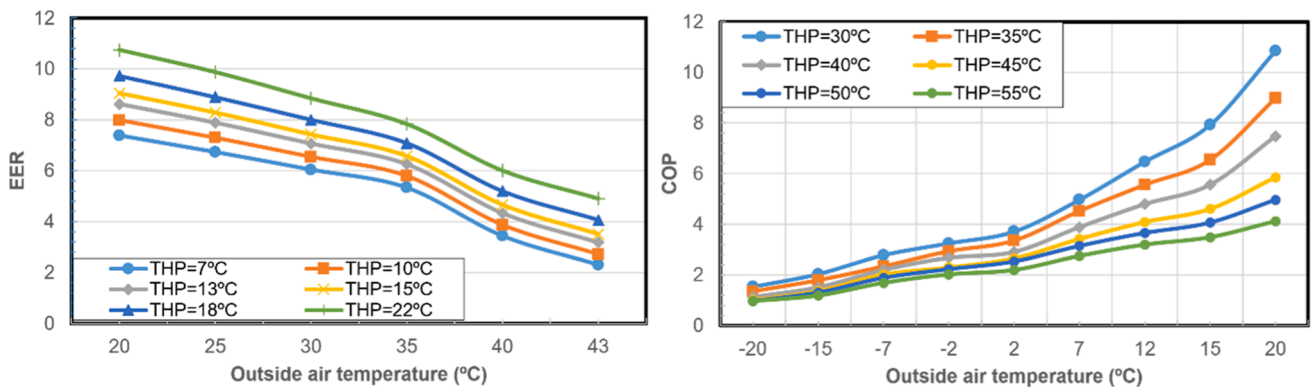

Fig. 4. Scheme of the hydronic and zoned control system [27].

heat which the minimum fluid (the fluid with a lower value of the product of mass flow rate and specific heat) can transfer from the inlet fluid temperature until the inlet temperature of the other fluid. From Eq. (5), the supply air temperature and water outlet temperature of the ducted fan coil are calculated, in heating mode, in Eqs. (6) and (7):

$$T_{a,o} = T_{ret} + \varepsilon \bullet \frac{\min(m_a \bullet Cp_a, m_w \bullet Cp_w) \bullet (T_{w,i} - T_{ret})}{m_a \bullet Cp_a} \qquad (6)$$

$$T_{w,o} = T_{w,i} - \varepsilon \bullet \frac{\min(m_a \bullet Cp_a, m_w \bullet Cp_w) \bullet (T_{w,i} - T_{ret})}{m_w \bullet Cp_w} \qquad (7)$$

where T_{ret} and $T_{w,i}$ are the return air temperature from the zones and the water inlet temperature, respectively. The sensible load handled by the fan coil is calculated as follows:

$$Q_{rs} = m_a \bullet Cp_a \bullet (T_{a,o} - T_{ret}) \qquad (8)$$

The fan electricity consumption is a function of the speed of the fan according to the manufacturer data [33].

The pumps are modelled with standard Trnsys component Type3. The power consumption is calculated as a linear function of mass flow rate according to the value of pump control signal (γ):

$$P_{pump} = \gamma \bullet P_{max} \qquad (9)$$

3.2. Ducted fan coil zoned control system (ZonFC) model

The ZonFC acts at two levels: zones and system. At zone level, the system is continuously turning on and off depending on if the temperature of the zone is inside or outside the dead-band temperature range (typically 0.5 °C) with respect to the set point air temperature. As each zone has its own thermostat and motorized damper, when the zone air temperature is in the comfort dead-band, a control signal is sent to the zone's motorized damper which interrupts the air supply to the zone. Also, the dampers are closed in the zones that are not occupied. Dampers will be opened when the zone is in demand and closed when it is in comfort.

Regarding the system level, the control algorithm sets the fan-coil's fan speed and the HPT as a function of the thermal behavior and needs of each zone.

The fan speed control is based on the number of zones in thermal demand ($N_{zones,dem}$). The weight of the zones (P_{vel}) is proportional to the total number of zones (N_{zones}).

$$P_{vel} = \frac{N_{zones,dem}}{N_{zones}} \qquad (10)$$

Table 1 shows how the algorithm sets the dynamic selection of the fan speed according to the number of fan speeds and the weight of the zones. The maximum number of zones is limited to 8 and the maximum number of fan speeds is limited to 5.

Therefore, in the case of study, the number of zones in the building are 5, and the selection criteria of the fan speed is represented in Table 2.

Another advantage of changing the fan speed and then, adjusting the air mass flow rate to the zones as dampers are being sequentially closed, is avoiding an overpressure in the ductwork when only one or two zones are opened. In the case of a zoned control system but without the work of

Table 1
Fan-coil fan speed selection in function of the weight of the zones.

Velocity selected	Number of fan speeds			
	2	3	4	5
1	0.01–0.50	0.01–0.34	0.01–0.25	0.01–0.20
2	0.51–1	0.35–0.67	0.26–0.50	0.21–0.40
3	–	0.68–1	0.51–0.75	0.41–0.60
4	–	–	0.76–1	0.61–0.80
5	–	–	–	0.81–1

Table 2
Fan-coil fan speed selection in a building with 5 zones.

Nzones	Zones in demand	Pvel	Velocity
5	1	0.2	Low
	2	0.4	Medium
	3	0.6	Medium
	4	0.8	High
	5	1	High

the communication gateway, which allows changing the speed of the fan, a by-pass should be needed. For that reason, in this study, the air distribution model in ductwork is not modelled.

Considering typical water temperatures in fan-coils, the range of temperatures are, in heating mode, $HPT_{min} = 42$ °C, $HPT_{med} = 44$ °C and $HPT_{max} = 45$ °C, and in cooling mode, $HPT_{min} = 7$ °C, $HPT_{med} = 8$ °C and $HPT_{max} = 10$ °C. When several zones have been conditioned at the same time, setting the HPT is not trivial. For this reason, the following two effects are checked.

1. Firstly, the HPT depends on the difference between the zone air temperature and the set point temperature of each zone, and the biggest difference between all the zones in demand (ΔT_{max}) is considered. As shown in Fig. 5, the ΔT_{max} sets a preselected HPT which can be modified by the second effect.

For example, when the ΔT_{max} is higher than 2.5 °C, the zone is far from the set point temperature and the HPT_{max} (in heating) or the HPT_{min} (in cooling) is selected to supply a higher (in heating) or lower (in cooling) supply air temperature trying to encourage a faster temperature air evolution in the zone. As the temperature of the zone evolves towards the comfort band, the HPT is regulated to more favorable conditions.

2. The second effect is related to the selected fan-coil speed. When most of the zones are in demand, the air flow supplied should be high, so the heat pump is likely to work at full load conditions. When the number of zones in demand decreases, the HPT is sequentially changing to a more favorable temperature, and the performance of the heat pump improves, as indicated in Table 3.

The algorithm considers the possibility to change the preselected HPT, if it would be necessary due to the selected fan-coil speed. The reason is that the final HPT selected should guarantee the strictest effect.

$$HPT = stricteseffect(\Delta T_{max}, FanSpeed)$$

It could be clarified as follows for a case in the heating season. In a building with a 5-zone system working, in which all the zones are not in thermal comfort but near to achieve it, the temperature difference between the rooms and setpoints is low, so due to the ΔT_{max} effect, the HPT_{min} is imposed. However, when all the zones are on thermal demand, the highest fan speed is selected, the mass flow rate is high and, it is therefore necessary that, due to the Fan Speed effect, the energy exchange must be carried out with water at a high temperature (HPT_{max}). Therefore, the gateway will change the preselected HPT from low to high, allowing to correct an unfavorable situation in the zones. The opposite can also occur. In the same case, at the beginning, there are 4 zones that reach thermal comfort and one of them is in demand. The imposed fan speed would be low, and, in the same way, a low HPT would be preselected. However, if this zone in demand is at a high level of discomfort (ΔT_{max} is very high), the gateway would change the preselected HPT from low to high, because the ΔT_{max} effect is more restrictive.

3.3. Ducted fan coil Non-Zoned control system (NZonFC) model

At zone level, the same thermal comfort dead-band is imposed in the master zone where the thermostat is placed. At system level, the fan-coil fan speed control in a typical installation with a heat pump and a ducted fan-coil is a traditional on/off control with hysteresis and 3 fan speeds,
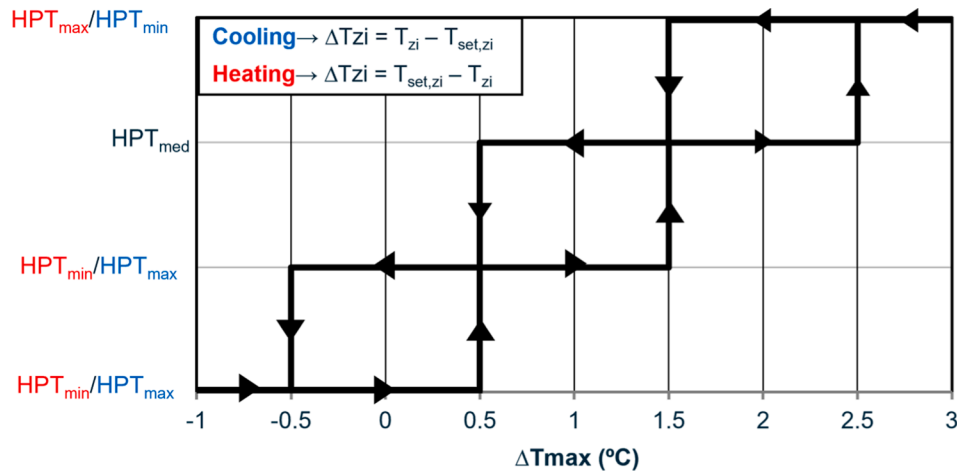
Fig. 5. HPT control selection according to the zone ΔT_{max}.

Table 3
HPT selection according to the fan speed.

Velocity selected	Number of fan speeds			
	2	3	4	5
1	HPT_{min}	HPT_{min}	HPT_{min}	HPT_{min}
2	HPT_{max}	HPT_{med}	HPT_{med}	HPT_{med}
3	–	HPT_{max}	HPT_{med}	HPT_{med}
4	–	–	HPT_{max}	HPT_{max}
5	–	–	–	HPT_{max}

which is the most common in residential sector, as shown in Fig. 6. In the case of different number of fan speeds, hysteresis will be adapted.

For example, in cooling mode, when the difference is higher than 2.5 °C, the maximum speed is chosen to decrease the zone air temperature as quickly as possible. Then, the speed of the fan is gradually changed to medium and low speed as the zone air temperature approaches the set-point temperature. When the difference is lower than −0.5 °C the fan is turned off. Thus, both systems have a similar performance with the aim to evaluate in detail the influence of thermal zoning in the results.

With respect to the heat pump, the heating or cooling mode is predefined by the user and the HPT is set according to two options:

1. Fixed HPT. HPT set point is automatically set to 45 °C and 7 °C in heating and cooling mode, respectively. This is the most common case for this HVAC configuration in the residential sector. The hydraulic circuit is configured to work with a fixed water temperature and variable water mass flow rate with a variable speed circulation pump and the water temperature can be adjusted by controlling the return set-point.

2. Variable HPT. HPT set point is set according to the outside air temperature which is measured by a temperature sensor. This option could be configured by an expert installer of HVAC systems or by the user of the building. The manufacturer allows to introduce a lineal relation between the HPT and the outside air temperature. The slope of this function depends on different factors like the type of building, internal gains, outdoor conditions, etc. For this reason, manufacturers give the option to configure this curve or incorporates some heating and cooling curves by default and users can select the most convenient. In this case, a simple calculation method of the HPT is proposed. Firstly, the slope of the lineal function (m) is calculated as a function of the minimum ($T_{min,HPT}$) and maximum ($T_{max,HPT}$) heating/cooling HPT available and the maximum and minimum annual outdoor air temperature ($T_{min,out}$ and $T_{max,out}$) as follows:

$$m = \frac{T_{max,HPT} - T_{min,HPT}}{T_{max,out} - T_{min,out}} \tag{11}$$

Secondly, the HPT is calculated in each time step according to the outdoor air temperature (T_{out}):

$$HPT = T_{min,HPT} + m \bullet \left(T_{max,out} - T_{out}\right) \tag{12}$$
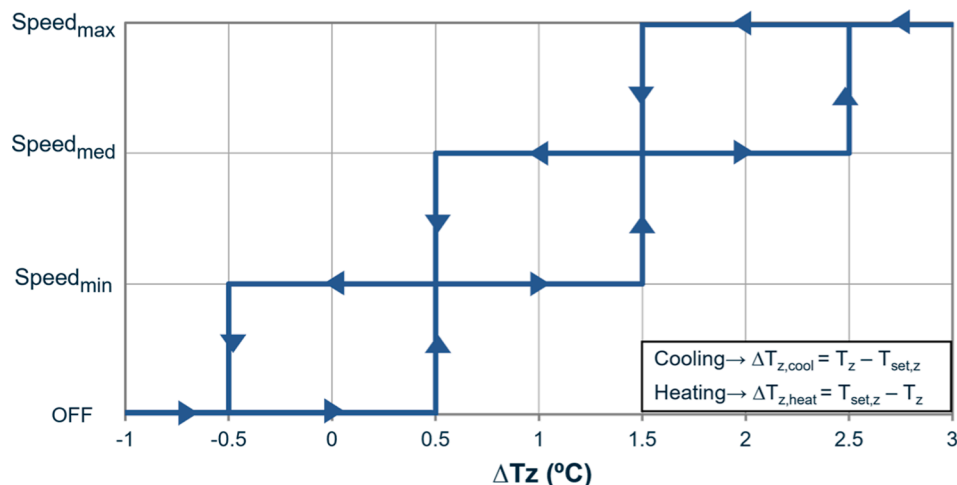

Fig. 6. Fan Speed selection according to the ΔT_z of the master zone.

The minimum and maximum outdoor temperatures have been obtained from EnergyPlus climatic data used in the simulation for each city. The maximum and minimum HPT are set to 5–10 °C for cooling and 40–50 °C for heating, according to the values recommended by typical manufacturers.

3.4. Individual fan coil control system (IndFC) model

The characteristics of the control system in the IndFC configuration are equal to the NZonFC model with the difference of there is a fan coil installed in each zone instead of only the ducted fan coil. The fan speed selection of each fan coil is as described in Fig. 6 but considering the temperature difference between the zone air and set point temperature of each zone.

For a system with multiple fan coils, the water temperature can be controlled locally using a bypass that mixes supply and return water flows. The options to set the HPT are also the same as the NZonFC model, although each fan coil only operates if the zone is occupied or is in thermal demand. On the contrary, a valve that supply the hot or cold water to this fan coil is closed. So, the heat pump only pumps water to the fan coils which are operative.

4. Case of study

The dwelling under study (Fig. 7) could be a typical representative residential housing. It has five conditioned zones (living room LR, kitchen K, office OF, children's bedroom CR and parents' bedroom PR), with a surface area of 121 m^2. The simulation is carried out in different European cities (Paris, Málaga and Milan) with different climate conditions. Paris has the typical Western European oceanic climate which is affected by the North Atlantic Current. Málaga is a Mediterranean city which is characterized by a mild, humid winter, and warm summers and Milan has a humid subtropical climate characterised by hot summers and cold, foggy winters. EnergyPlus weather files are used in the simulation [34]. The Trnsys' multizone building model Type 56 is used for the thermal modelling of the dwelling.

The enclosures are representative of the different regulations prevailing in each country. Ceiling and floor are enclosures that adjoin other floor or ceiling, respectively, of a similar dwelling. A boundary condition of average temperature between zone air temperature and outside air temperature is applied. The thermal performance of buildings is strongly affected by the materials of the envelope. To avoid a high source of uncertainties, typical buildings defined by the standards of each country have been selected. Table 4 shows the values considered for the overall heat transfer coefficient (U-value) of the different enclosures of the dwelling. It should be noted that the external walls are very isolated for Paris and Milan because of the severe climatic conditions in winter, whereas a higher U-value is defined for the Mediterranean climate of Málaga. The windows are typical double panel windows 4/16/4 with the U-values defined in Table 4.

Four categories of internal gains were considered: people, lights, electric equipment, and ventilation, which are the same for all the cases of the study presented.
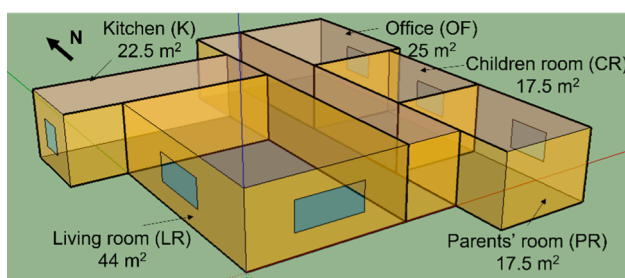

Fig. 7. 3D representation of the home.

Table 4
Threshold U-values (W/m^{2}K).

City	Regulation	Wall	Ceiling	Floor	Window
Malaga	RITE 2019 [35]	0.65	1.1	1.1	2.2
Paris	RT 2012 [36]	0.24	0.66	0.66	1.14
Milan	D.Lgs.N.192 2005 [37]	0.34	0.33	0.3	2.2

The influence of the occupant lifestyle patterns on the energy performance of residential buildings is widely studied in the literature. Yousefi et al. [38] evaluated the building energy consumption with different occupation profiles in various building envelopes and different climate zones. Yang et al. [39] highlighted that occupancy is one of the most important factors impacting energy use of HVAC systems and they analyzed its influence on the heating and cooling loads of a building. In this case, a typical occupancy profile is applied, for the different zones, in residential buildings, to determine the operation of the HVAC system (Fig. 8).

Five people with activity level designed as seated, very light work, equivalent to 120 W of total heat emission by a person [40] are considered. The lights are LED, with a power of 11 W. A lighting control has been implemented according to a set point of 250 W/m^2 of horizontal solar radiation. In the occupied zones with external windows, when this value is lower than the set point the lighting is on. Therefore, a reasonable profile of lighting turning on/off over the year is assumed. A ratio of 5 W/m^2 is used for the equipment thermal loads. Finally, the indoor air quality is maintained by renovation through infiltration and ventilation. A rate of 0.6 renewals/hour is set for outdoor ventilation airflow in all rooms except the kitchen, which is set at 2.9 renewals/hour.

5. Results and discussions

The objective of this study is to evaluate the effects of a zoned control system with respect to a typical non-zoned system. The criteria used for the comparison are the electricity consumption and the comfort level provided by each of them. Thus, this section is divided into the following parts: the sizing of the heat pump according to the influence of thermal zoning, the evolution of the zone air temperatures in a typical winter and summer day, the analysis of the comfort levels provided. It should be noted that the effect of the HPT described in section 3.3 and 3.4 has been analyzed only in energy consumption results and in economic analysis, and it has been resumed in Table 5.

5.1. Influence of the thermal zoning in the sizing of the heat pump

The calculation of thermal loads in the house in each city is done with the user comfort range between 22 °C and 24 °C. Considered cooling and heating periods are from October to April and from May to September, respectively. In a non-zoned system, the distribution network has no element that enable the system to deal separately with the needs of each zone. Therefore, to guarantee the possibility of meeting peak loads in all zones, the performance rating of the unit must be greater than the sum of the peak sensible loads of the zones, even if they are not simultaneous. Eq. (13) describes the calculation of the nominal capacity of the unit in a non-zoned system (P$_{nom,NoZon}$):

$$P_{nom,NZon} = \sum_{i=1}^{NZones} \max(Q_{dem,zonei}(t)) \qquad (13)$$

where Q$_{dem,zone\ i}$ (t) is the thermal load of the zone i in each step of time t.

On the other hand, in a zoned system, the distribution network has motorized dampers that allow adjusting the thermal contribution of the system to the demand of each zone separately. This means that the unit is sized by considering the maximum simultaneous sensible load of the
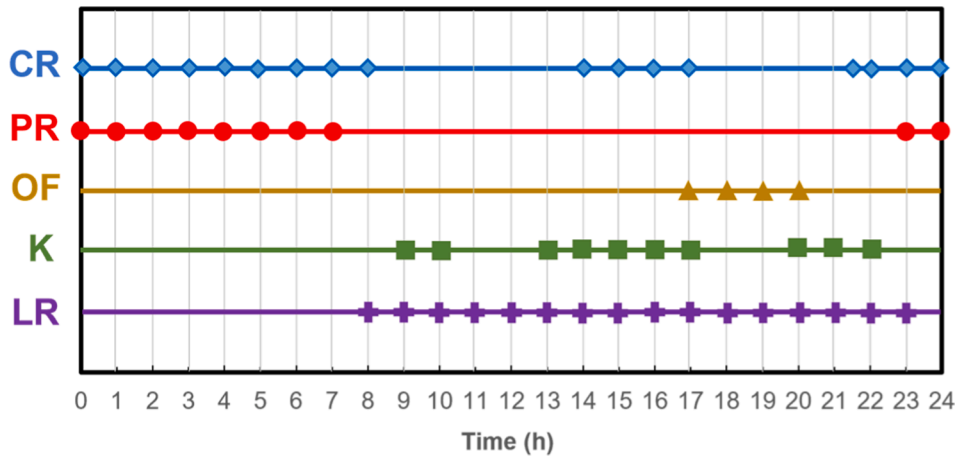
Fig. 8. Occupational profile.

Table 5
Control systems description. Cases of study.

Control system	Zoning	HPT	FC fan speed
ZonFC	Yes	Variable: strictest effect (ΔT_{max}, Fan Speed) (Fig. 5, Table 3)	N_{zones} and number of fan speeds (eq. (10), Table 1)
NZonFC1	No	Fixed (7/45 °C for cooling and heating)	ΔT_z of the master zone
NZonFC2	No	Variable: Heating and cooling curve (eqs. (11) and (12))	ΔT_z of the master zone
IndFC1	Yes	Fixed (7/45 °C for cooling and heating)	ΔT_z of the corresponding zone
IndFC2	Yes	Variable: Heating and cooling curve (eqs. (11) and (12))	ΔT_z of the corresponding zone

zones. In other words, for every time step, loads of all zones are added together, and the unit is sized based on the annual maximum for cooling and heating. The nominal capacity of the unit in a zoned system ($P_{nom,Zon}$) can be obtained as follows:

$$P_{nom,Zon} = \max\left(\sum_{i=1}^{NZones} Q_{dem,zonei}(t) \bullet P_{ocup,zonei}(t) \right) \quad (14)$$

where $P_{ocup,zone\ i}$ (t) is the occupational profile value of zone i (1 if it is occupied or 0 if it is not occupied).

Table 6 shows the peak (NZon) and simultaneous (Zon) loads for the three cities, the percentage of thermal load saving in heating (%Heat) and cooling (%Cool) and the abbreviation of the model of the heat pump from a common manufacturer [27,28]. The heat pump unit model selected from the simultaneous loads (Zon) is used for the ZonFC and IndFC systems. The unit selected from the peak loads (NZon) is used for the NZonFC system.

It should be noted that the sizing of the unit, according to the simultaneous loads, allows to install a heat pump with a lower capacity

Table 6
Thermal loads of each city and heat pump model sized.

City	Malaga		Paris		Milan	
System	Zon	NZon	Zon	NZon	Zon	NZon
Qheat (kW)	3.5	4.5	7.3	8.0	8.1	8.8
Qcool(kW)	4.2	5.3	2.5	4.0	3.9	4.8
% Heat	22.2		9.3		8.0	
% Cool	20.8		37.0		18.8	
Heat Pump model [18,19]	Q005	Q007	GA06	GA08	Q011	Q011

than the typical peak load sizing procedure in Malaga and Paris, but not in Milan. As the weather is colder, the influence of thermal zoning in the heating season is lower in this aspect. This is because, when a zone is not in demand or there are no people in it, the load of the zone increases and, when it is activated, the unit should meet a higher accumulated load. This effect has more influence in colder climates where the difference between the peak and simultaneous load is not very high.

The models of the ducted and individual fan coils are resumed in Table 7 according to the thermal loads results obtained in the three cities. The configuration of individual fan coils is considered as a zoned system because each fan coil controls the zone air temperature individually, so in this case the heat pump model is equal to the Zoned system.

5.2. Thermal comfort evaluation

From this analysis, a more detailed comfort analysis is evaluated in each zone. As comfort indicators, the Predicted Percentage of Dissatisfied (PPD) and the Predicted Mean Vote (PMV) are used according to the ISO 7730 standard [40]. The design criteria for a standing or relaxed person (1 met) wearing typical winter (1 clo) and summer indoor clothing (0.8 clo) are considered with air velocity set to 0.1 m/s. According to this standard, the comfort categories are presented as follows:

- Category A: PPD < 6% and PMV between −0.2 and 0.2.
- Category B: PPD < 10% and PMV between −0.5 and 0.5.
- Category C: PPD < 15% and PMV between −0.7 and 0.7.

The operation, performance, and thermal comfort evaluation of the 3 control systems for the living room (LR), kitchen (K) and parents' room (PR) is analyzed on a typical winter day, the 8th of February, in Paris (Fig. 9), where the outdoor air temperature oscillates between 4 and 9 °C, which is representative of a cold winter. Zone air temperatures (continuous lines) are represented in the left axis and the PPD value

Table 7
Terminal units' models.

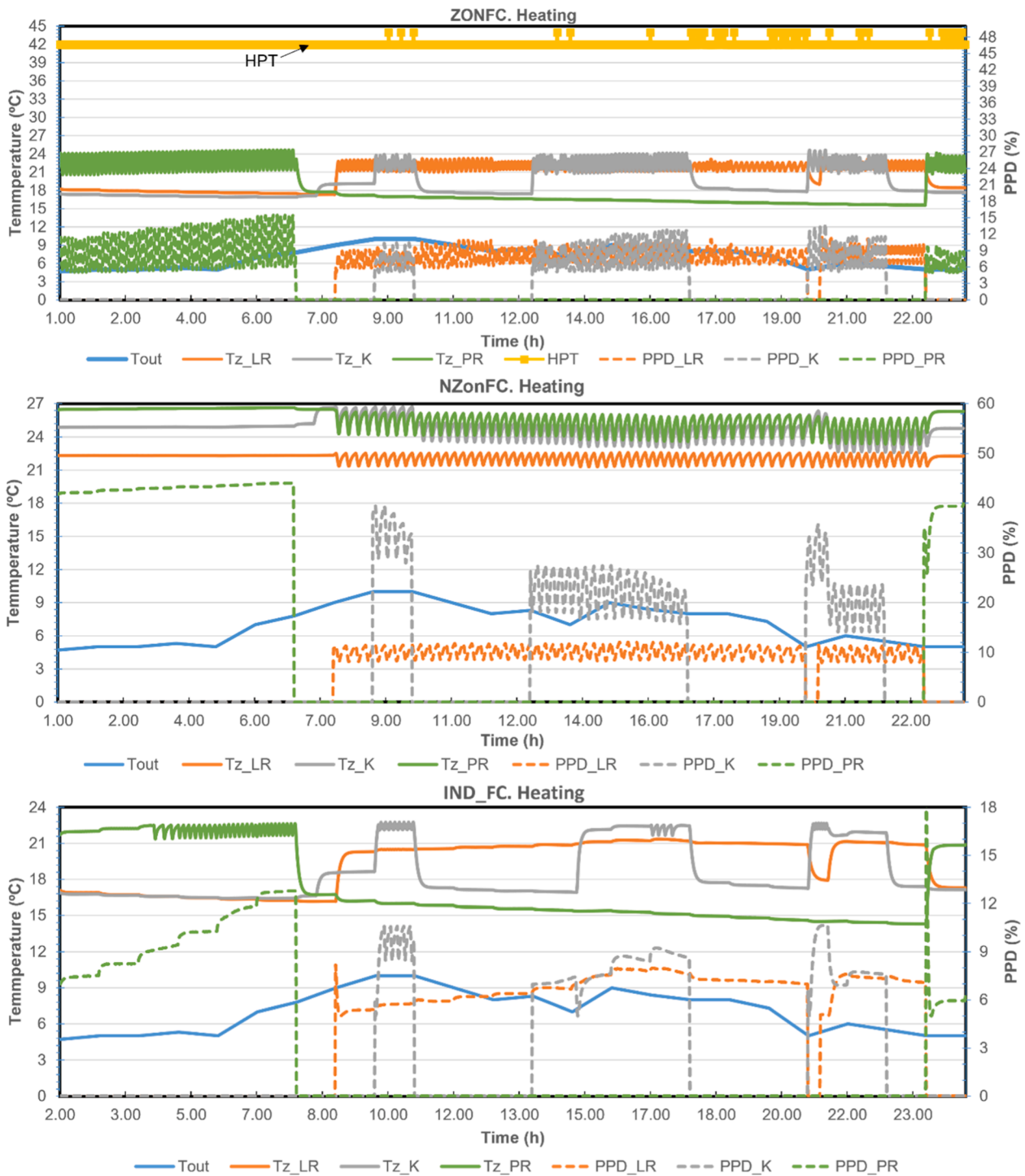| FC model	Thermal zone	Málaga	Paris	Milan
Ducted Zoned FC [41]	–	FWD06	FWD08	FWD08
Ducted Non-Zoned FC [41]	–	FWD08	FWD10	FWD10
Individual FC [33]	LR	FWT02CT	FWT02CT	FWT04CT
	K	FWT02CT	FWT02CT	FWT03CT
	OF	FWT02CT	FWT03CT	FWT03CT
	CR	FWT02CT	FWT02CT	FWT02CT
	PR	FWT02CT	FWT02CT	FWT02CT

Fig. 9. ZonFC, NZonFC and IndFC system performance during a typical winter day in Paris (8th February).

(discontinuous lines) are represented in the right axis. The influence of the thermal zoning in the thermal behavior of the zones is assessed. The simulation time step is one minute, the same as that of the operation of the control system. The HTP value for the case of ZonFC is also shown.

The ZonFC (first graph) shows how all the zones are in thermal comfort in the time range of the occupational profile, while the rest of the time, thermal comfort is not evaluated (PPD is set to 0), the evolution of the temperature is free and oscillates outside the comfort dead-band. For example, the parents' room is comfortable from 23:00 to 7:00, but the rest of the day the temperature decreases to a minimum near 15 °C. The kitchen occupational profile is more intermittent during the day,

whereas the living room is occupied during daytime hours. In both cases, the zone air temperature fluctuates in the comfort band during these occupational periods. From the point of view of the PPD evaluation, it should be noted that the values obtained are very positive, with values in all the zones lower than 12%. Only in the parents' room, during the period from 5:00 to 7:00 am the PPD reaches a value of 15%, due to the heat accumulation in the bedroom. The HTP is managed from 42 to 45 °C according to the algorithm described in Section 3.2.

On the other hand, the comparison with the NZonFC is very interesting (second graph). The evolution of the temperature in the living room is very stable, around the set point of 22 °C, but it can be seen how

the temperatures of the kitchen and parents' room are higher, from 24.5 to 26 °C during all day. Therefore, the PPD of the LR is 6% but overheating occurs in the other zones with the consequent thermal discomfort of the users, as it can be seen in the evolution of the PPD, about 40% during the night in PR and 25% in the kitchen at midday.

Finally, in the same way as the ZonFC, the IndFC system ensures an efficient thermal zoning thanks to the heat exchanged in each zone, maintaining also similar PPD values with respect to the ZonFC. So, from this point of view, it is noticeable that the IndFC and ZonFC can be used for guaranteeing thermal comfort in residential buildings.

On the other hand, the same analysis is carried out (Fig. 10) in cooling mode, the 10th of July. but in the Mediterranean climate of Málaga. The LR, OFF and CR zones are represented.

The conclusions are like those obtained in the heating period. ZonFC controls the zone air temperatures in the comfort band when the zone is activated in its period of occupational time. Besides, it should be emphasized that undercooling can occur in NZonFC when cold air is supplied to the zones with low cooling load, as it can be seen in the OF, where the air temperature is about 20–21 °C during all day. From the comfort point of view, when the OF or the CR zones are activated, at the beginning, the PPD value obtained is very high. This is due to the high period that these zones are not conditioned, so the accumulated thermal load is very high and there is an initial period of time to stabilize the temperature and obtain acceptable comfort values.

Regarding the comfort analysis according to the climate conditions, it is remarkable that the summer in Malaga is not very hot. In a typical
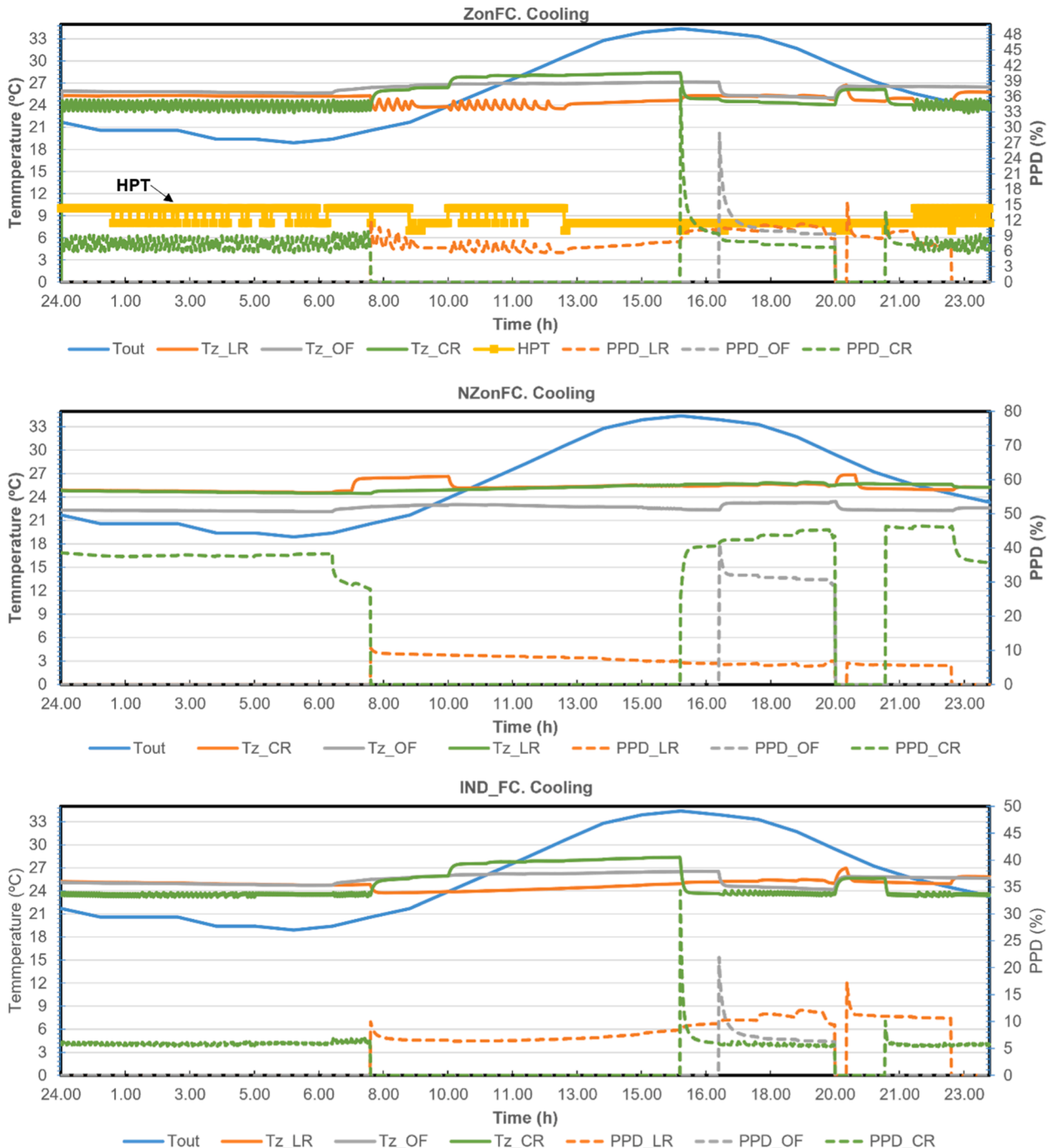

Fig. 10. ZonFC, NZonFC and IndFC system performance during a typical summer day in Malaga (30th July).

summer day the temperature in July or August is around 28–30 °C with a high humidity content. If the temporal evolution of the PPD and PMV values are analyzed, it should be noted that the deviation from the comfort zone is carried out in the months of May, June, or September, when the heat pump operates in part load performance. This situation confirms the advantages of the sizing procedure with simultaneous loads in the zoned system, which can adapt the power of the unit to the needs of the zones.

A deeper general analysis of thermal comfort is presented in Fig. 11. The percentage of hours where the thermal comfort is considered in each category, calculated as the ratio between the current PPD and the PPD limit according to the thermal comfort category. The LR and CR are chosen as the representative zones of the building but also the total building thermal comfort is included for the three cities considered.

Nowadays, the comfort requirements in buildings are very demanding. In the non-residential sector, a minimum of category B is usually expected, but in the residential sector category C is accepted. Then, in a first general analysis, the results of Fig. 11 confirm the conclusions presented in Figs. 9 and 10, where high percentages of hours in thermal discomfort can be observed in NZonFC compared to the other systems. It is noteworthy the thermal discomfort of 43% and 51% for CR in Malaga and Paris, respectively. In contrast, the ZonFC ensures a high percentage in thermal comfort higher than 80% in all the zones for categories A and B and a minimum total thermal discomfort from 1 to 6%, whereas the IndFC, which also guarantees the thermal zoning, the percentage of comfort for categories A and B is about 60%, with thermal discomfort of 12–16%.
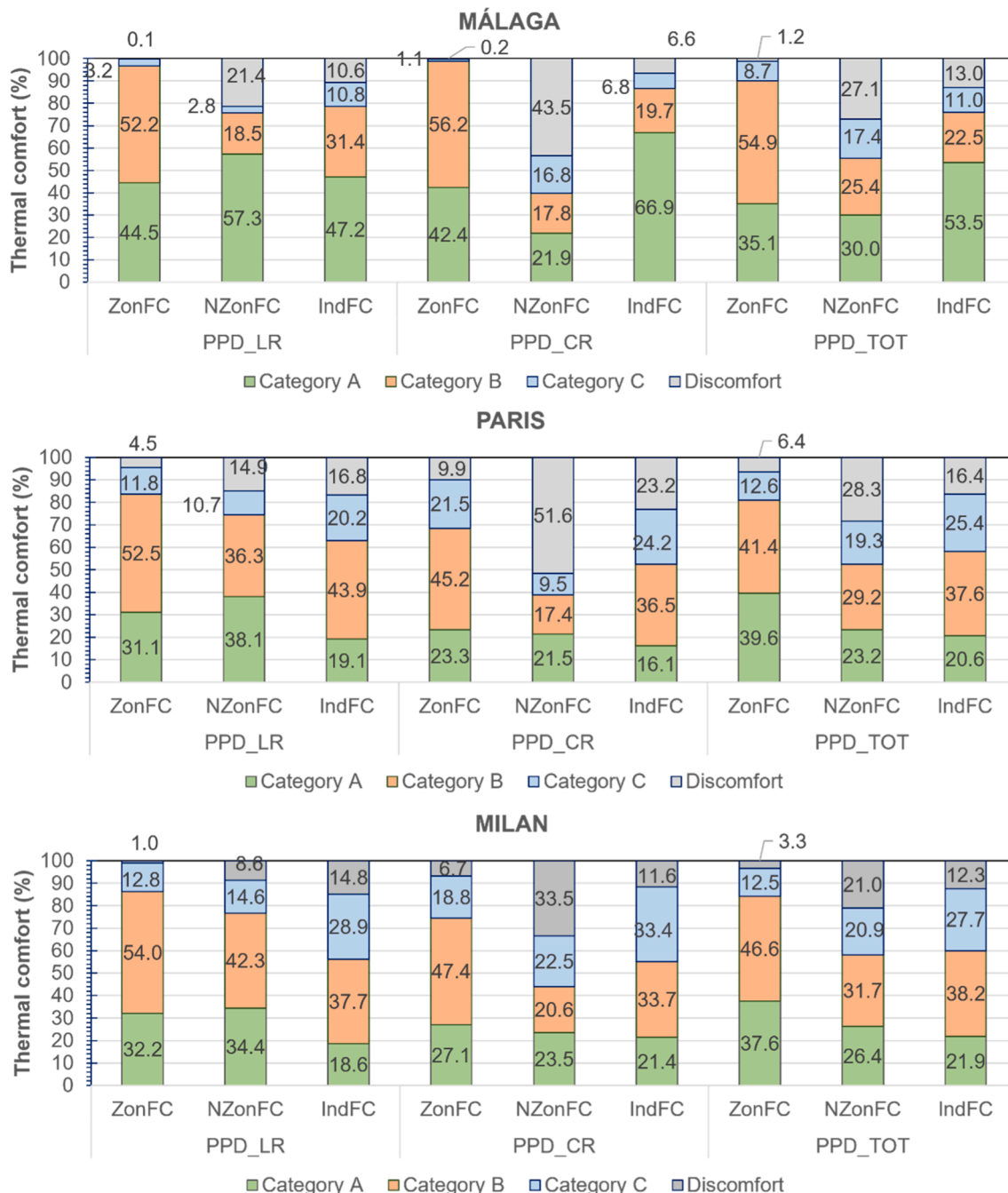

Fig. 11. Percentage of hours in thermal comfort for the LR, CR and the building in the three cities proposed.

5.3. Heat pump energy efficiency and energy consumption

To complete the evaluation, it is fundamental to compare the control systems by analyzing the energy consumption which includes the electrical consumption of the heat pump and the motor of the fans. Fig. 12 presents a comparison for the cases presented in Table 5, in terms of total energy consumption in heating and cooling seasons.

Overall, it is observed that HPT control according to the proposed heating and cooling curves (NZonFC2 and IndFC2) has achieved significant improvements in energy consumption with respect to the conventional case with fixed HPT (NZonFC and IndFC), reaching ranges from 5.9 to 8.8% in Milan to 12.2–20.4% in Málaga. So, this basic strategy which only requires the intervention of an expert user allows obtaining important energy savings, which are less significant in less severe climates.

Despite this, in all the cases, the ZonFC is the system with the lowest energy consumption. Focusing on the comparison between ZonFC and IndFC or IndFC2, both modes are thermally zoned, but remarkable ZonFC energy savings are achieved. The main reasons are two. Firstly, the total energy consumption of the fans in the IndFC is higher because the five fan coils operating lead to higher energy consumption than the fan of the ducted fan coil. Besides, the IndFC imposes a high-speed fan when the temperature difference between zone air and set point temperature is high, so there is a high period in different zones operating in this unfavorable condition. Secondly, the heat pump consumption in the ZonFC is also lower than IndFC or IndFC2 because the control system algorithm manages the HPT according to both, the thermal needs of the zones and the fan speed selected. In short, energy savings of ZonFC compared to IndFC or IndFC2 are around 31–45% and 18–38%, respectively.

When considering the comparison between ZonFC and NZonFC, thermal zoning shows a very good solution to obtain important energy savings due to different influencing factors on the heat pump performance. The heating or cooling thermal energy provided by the heat pump is different according to the control system. ZonFC and IndFC can adapt this thermal energy either by turning off the motorized dampers or closing the corresponding valve of the FC of the zones which are either unoccupied or in thermal comfort, whereas the NZonFC operation is based on the demand of the main zone. In addition, the model of the selected heat pump is influenced by the thermal zoning control system as described in section 5.1. Thus, to have a better understanding, the influence of the thermal zoning on the performance of the heat pump is analysed in Fig. 13 with the representation of COP/EER against the outside air temperature for cooling and heating, in summer in Málaga and in winter in Paris, respectively.

As it can be seen, the heat pump efficiency coefficient highly depends on part-load performance. In both situations, the NZonFC efficiency coefficient is lower, first due to the efficient management of the thermal energy distribution from the heat pump to each of the building's zones and secondly due to the sizing of the unit taking into account the influence of the zoned control system. For example, in Málaga, the maximum EER for ZonFC with outside air temperature of 28 °C is 3 and for NZonFC is 2.7. The highest differences occur when the heat pump operates in part-load performance, with lower temperatures in summer and higher in winter. The improvement in the heat pump performance results in reduced energy consumption with savings ranging from 20 to 31% with respect to NZonFC and from 13 to 16% with respect to NZonFC2.

5.4. Economic analysis. Life cycle and payback period

The economic feasibility of the different HVAC configurations considered is evaluated through the calculation of the payback period (PB) of the ZonFC system with respect to the other systems, which depends on the initial cost (IC) and the operating cost (OC), as indicated in equations.

$$PB(years)_{ZonFC-NZonFC/ZonFC-IndFC} = \frac{\left(IC_{ZonFC} - IC_{NZonFC/IndFC}\right)}{\left(OC_{ZonFC} - OC_{NZonFC/IndFC}\right)} \qquad (15)$$

where the PB is calculated with respect to the NZonFC or IndFC.

The initial costs of the heat pump, terminal units (ducted and individual fan coils), air diffusion equipment (dampers and motorization) and zoned control system (control boards, communication gateways) have been obtained from manufacturers' catalogues [42,43], as resumed in Table 8.

As it can be seen, the initial cost of the ZonFC system is higher than the NZonFC and IndFC despite the reduction of the heat pump unit model, because of the costs of the Zoned control system and the dampers' motorization. The IC difference between the ZonFC and the others is indicated at the end of the table.

The OC is calculated from the energy consumption results from Fig. 12 and the electricity cost, which is recently updated [44]. The electricity cost is not constant, it fluctuates during the day and depends on different factors, mainly the electricity rate contracted by the user. However, a conservative value of 0.25 €/kWh is assumed, which is the same for all the configurations. Finally, the return of the investment period is calculated from the years of the payback period as the ratio between the differences between initial costs and operational costs, as indicated in Eq. (15). The economic analysis is presented in Table 9.

The economic analysis shows how the ZonFC investment cost, despite the reduction of the thermal capacity of the heat pump unit, is higher due to the cost of the control system. However, the reduction of the energy consumption results in a lower operating cost and the payback periods in each city are from 1.7 to 2.3 years with respect to the NZonFC and from 0.5 to 1 year with respect to the IndFC. Regarding the cases of NZonFC2 and IndFC2, the PB increases with respect to the others, but positive results are obtained, with a maximum of 4.9 years in Malaga with respect to NZonFC2. Considering that the lifespan of the different elements of the HVAC installation is around 15 years, the payback period obtained is acceptable.
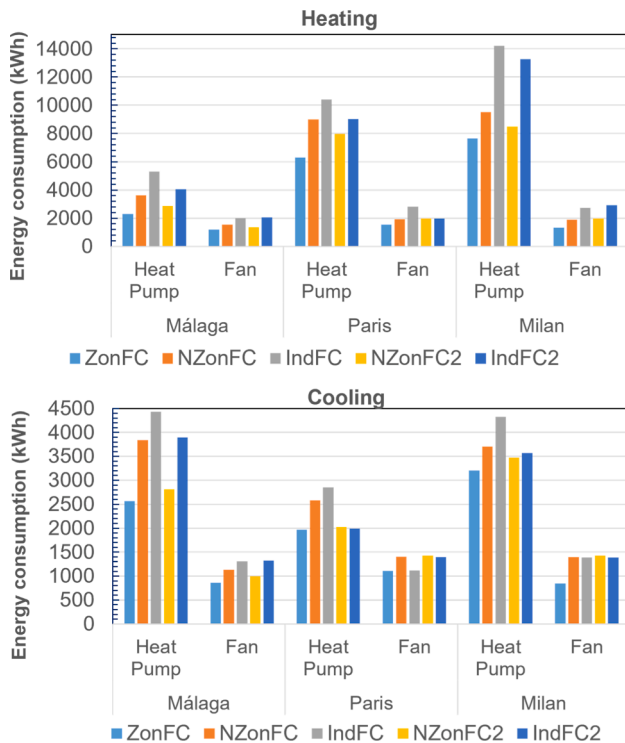

Fig. 12. Comparison of energy consumption in each city for heating and cooling seasons.
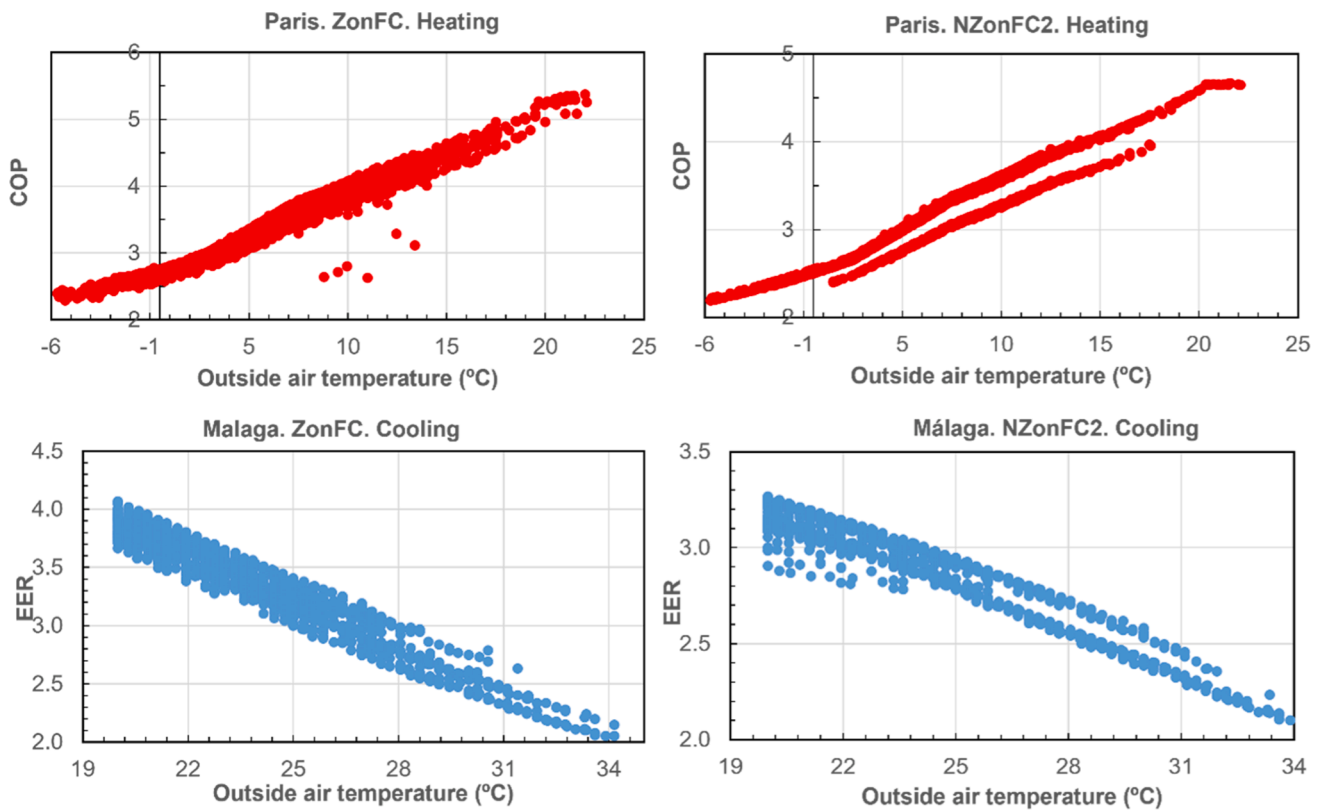
Fig. 13. Comparison of heat pump's efficiency coefficient for heating and cooling seasons.

Table 8
Initial Costs of the ZonFC, NZonFC and IndFC elements (in €).

City	Malaga			Paris			Milan		
Element	ZonFC	NZonFC	IndFC	ZonFC	NZonFC	IndFC	ZonFC	NZonFC	IndFC
Heat Pump	5224	5866	5224	4439	5402	4439	7772	7772	7772
Ducted FC	744	880	–	880	1013	–	880	1013	–
Individual FC	–	–	1685	–	–	1711	–	–	1768
Manufacturer control	–	152	162	–	152	162	–	152	162
Zoned Control	1743	–	–	1743	–	0	1743	–	0
Dampers' motorization	580	–	–	580	–	–	580	–	–
Total	8291	6898	7071	7642	6567	6312	10,975	8937	9702
IC difference	–	1393	1220	–	1075	1330	–	2038	1273

Table 9
Economic Analysis. Payback period calculation.

System/City	OC(€)					PB (years)			
	ZonFC	NZonFC	NZonFC2	IndFC	IndFC2	ZonFC-NZonFC	ZonFC-NZonFC2	ZonFC-IndFC	ZonFC-IndFC2
Malaga	1767	2575	2052	3035	2872	1.7	4.9	1.0	1.1
Paris	2772	3772	3394	4329	3631	1.1	1.7	0.9	1.3
Milan	3300	4168	3885	5702	5319	2.3	3.5	0.5	0.6

6. Conclusions

In this study, a new model of an air to water heat pump with a ducted fan-coil with a zoned control system is developed using available energy analysis software. Among the new HVAC control systems as MPCs, which achieve excellent results but are still not implemented due to some factors such as the complexity or a high cost, the zoned control system presented in this paper is characterized by its practical feasibility, since it is commercialized and operative in different HVAC installation configurations in many European countries and the US. It is based on an algorithm and a bidirectional communications gateway that allows the controls system to act on the heat pump, setting the set-point temperature, and on the fan coil, setting the fan speed. A comparison in terms of thermal comfort and energy consumption with respect to two conventional configurations: a non-zoned ducted fan coil system and individual fan coils, in a residential building, and under different climatic conditions, is presented and evaluated. The main conclusions are:

● When the heat pump is sized, it can be observed that the zoned control system reduces the thermal energy the unit must provide to

meet the thermal load of the zones. In warmer climates the thermal savings are more significant and then, a heat pump with a lower capacity can be installed.

- The zoned system improves the thermal comfort of occupants concerning a non-zoned system. The zoned system adjusts the supply airflow rate to the zones according to their thermal requirements avoiding overheating/overcooling in the zones that can occur when the non-zoned system is operating. Then, using a zoned control system guarantees to achieve the best comfort categories most of the time in all zones, while a non-zoned control system can only assure to reach comfort conditions in the master zone.
- Thermal zoning improves the energy efficiency coefficient of the heat pump, taking advantage of the better sizing procedure that leads to a smaller heat pump, and also because the algorithm can adapt the thermal energy of the heat pump to the thermal needs of the building with the HPT, the fan speed of the FC and motorized dampers control.
- The ZonFC system achieves high energy savings in all the cities considered because of the regulation of the fan-coil fan speed and the HPT of the heat pump according to the thermal requirements of the building and thanks to the communication gateway. It is highlighted a 18–38% of savings in energy consumption with respect to the IndFC2, and from 13 to 16% with respect to NZonFC2. Regarding the influence of the climatic conditions, the control system achieves higher energy savings in Málaga (warmer climate) than Milan (colder climate).
- The economic analysis results in interesting payback periods from 1.7 to 4.9 years with respect to eh NZonFC and from 0.5 to 1.3 years with respect to the IndFC.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgements

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