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All-air system and radiant floor for heating and cooling in residential buildings: a simulation-based analysis

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Ventilation systems are necessary to ensure a good indoor air quality, especially in buildings characterized by high energy performance. An all-air system could be more flexible than radiant systems that need to be combined with other systems to provide fresh air and control latent loads. In this work, a multi-zone all-air system has been compared with a radiant floor integrated with mechanical controlled ventilation and an isothermal dehumidifier. All the systems have been modelled in TRNSYS considering a residential building and three different locations (Helsinki, Milan and Rome). The comparison has been performed in terms of both indoor conditions and energy consumption. The results outline that the two systems are able to maintain the desired indoor conditions. For the energy consumption, an air-to-water heat pump and all the auxiliaries have been considered. Under the same indoor conditions, in heating the electrical energy consumption of the all-air system is lower than that of the radiant floor (-19% in Helsinki, -32% in Milan and -74% in Rome); also in cooling the electrical energy consumed by the all-air system is lower in Milan and Rome (-14% and -29% respectively), whereas in Helsinki the difference is less than 5%.

Keywords: all-air system, radiant floor, TRNSYS, thermal comfort, energy efficiency.

INTRODUCTION

Buildings account for around 40% of the total world energy use (IEA 2017) and improving their air-conditioning systems will lead to important increases in energy efficiency and reduction of CO_2 emissions. Also in Europe, buildings are responsible for approximately 40% of energy consumption and 36% of CO_2 emissions (European Parliament 2018). This great energy consumption is due to high energy inefficiency of the building envelope and systems. In fact, 35% of the EU's buildings are over 50 years old and consequently a great part of the building stock presents low energy performance, of both building structures with low thermal insulation and HVAC systems (Martinopoulos et al., 2018). Residential buildings represents surely the great part of the entire building sector; for example, in Italy residential buildings account for about 30% of final energy consumption and around 70% of their energy consumption is due to heating and cooling (ENEA 2018).

In order to decrease the high energy consumption in residential buildings, several new standards and regulations were developed. European Commission started with the Energy Performance of Buildings Directives-EPBD (European Parliament 2002, 2010, 2018), Renewable Energy Directives (European Parliament 2009) and Energy Efficiency Directive (European Parliament 2012) that look at improving energy efficiency of building envelopes and of HVAC systems as well as the development of renewable energy technologies. In fact, there are two fundamental options to reach the result: the first one looks at the building envelope in order to limit the thermal losses with high thermal insulation levels and also to control the solar gains in cooling period; the second option consists in efficient systems for heating and cooling, e.g. radiant systems, heat pumps and renewable energy technologies. All these assumptions move to the low energy buildings (e.g. nearly or net zero-energy buildings) that represent one of the greatest opportunities to increase energy savings in this sector. In Europe, new buildings have to be nearly zero-energy buildings from 2021 (2019 for public buildings) according to the last EPBDs.

Another important challenge regards ensuring a thermally comfortable indoor environment and maintaining high air quality in buildings; an insufficient ventilation rate is associated with health problems such as inflammation, respiratory infections, asthma, allergies and sick building syndrome (Sundell et al. 2011) (Khovalyg et al. 2020). In this context, controlled mechanical ventilation systems (CMV) offer several advantages but they consume energy and can be noisy. HVAC systems have to be able to control the convective, radiant and latent loads in order to maintain the comfort conditions. Radiant systems are surely very efficient since they can work with a fluid temperature near the air temperature and, consequently, they can be coupled with heat pumps and other renewable energy technologies (Kilkis, 2012). However, the radiant systems need to be coupled to a dehumidification device to remove the latent load and also to a mechanical ventilation system to guarantee a good air quality in the building (Zhang et al., 2020).

From this point of view, a suitable air-system can be more flexible than radiant systems since it can satisfy thermal and latent loads at the same time, consequently the user has to manage one only system. In addition, when the heating and cooling loads of the dwelling is relatively low, the air volume to maintain the thermal comfort conditions can be decreased without involving local discomfort conditions. In literature, some researchers have focused the attention on the comparison between radiant and all-air systems, especially in cooling mode. Imanari et al. (1999) compared the radiant ceiling and all air-conditioning system via experiments and numerical simulations in terms of thermal comfort, energy consumption and cost efficiency considering an office room in Tokyo. They found the radiant ceiling competitive, but it had to be coupled to a control humidity system and the response to rapid change of the room condition had to be improved. Salvalai et al. (2013) carried out computer simulations with the software TRNSYS (Solar Energy Laboratory 2012) to analyse the energy consumption and thermal comfort conditions in an office building with different climates of Northern, Central and Southern Europe. They investigated a number of cooling technologies and found that radiant cooling reduced energy consumption and improved thermal comfort; however, they found that a suitable ventilation control strategy was fundamental to reach that objective.

Oxizidis and Papadopoulos (2013) compared radiant and convective systems in terms of energy consumption and thermal comfort by carrying out computer simulations of a single office in Thessalonica, Greece. They considered the floor, ceiling and wall radiant surfaces, concluding that hybrid solutions (e.g. a radiant system and an air system) in warm and humid climates ensure low energy consumption and good thermal comfort. Zarrella et al. (2014) investigated different dehumidification systems coupled to a radiant floor in a common apartment via computer simulations with in-house numerical model; they concluded that radiant floor and mechanical ventilation system allowed to achieve better comfort conditions and indoor air quality, together with reasonable primary energy use in three Italian locations (Venice, Rome and Bari). Khan et al. (2015) investigated both radiant floor and ceiling in a commercial building in India using computer simulations with EnergyPlus (U.S. Department of Energy 2020) and also computational fluid-dynamic analysis. They compared the radiant systems with a common all-air system in cooling mode investigating also the control strategies. Their results show a 30% energy savings with radiant systems coupled with a dedicated dehumidification system.

As regards radiant systems, in many recent papers attention has been focused either on heating or cooling aspects. Fabrizio et al. (2012) compared radiant heating and cooling floors/ceilings with two air systems (all-air and fan-coils) for various European climates in an office building via computer simulations in EnergyPlus (U.S. Department of Energy 2020); their results show that the energy consumption of radiant systems coupled with a suitable primary energy system is lower (from 20-30% in heating dominant locations, up to 60% in cooling dominant locations) than the all-air system. Magni et al. (2019) developed a numerical model to evaluate the 3D distribution of the mean radiant temperature in a room in which a radiant (floor, ceiling or wall) or a convective (radiators or all-air) heating system is used. Comparing the different systems in heating mode, they concluded that in buildings with very low transmission thermal losses the differences among the analysed emitters are strongly attenuated in terms of spatial distribution of the operative temperature. Moreover, they demonstrated that in presence of variable thermal loads all-air systems are faster than radiant systems in following the heating need of the building. Also Alessio et al. (2018) investigated different types of radiant floor and a radiant ceiling in heating mode using the DigiTHon tool (De Carli et al. 2012) and simulating one only room in heating mode with the weather data of Venice; they found that the performance of radiant systems is correlated to the building envelope (i.e. the better the quality of the envelope the better the overall performance of the radiant system) and the variable supply fluid temperature leads to lower consumptions compared to fixed value over the season. Wang et al. (2018) compared four typical space heating systems (all-air, radiator, in-slab floor heating, and lightweight floor heating systems) in intermittent operating conditions under different climates in China by means of a numerical model based on the electrical analogy. Their simulation results outline that the heating load of radiator is 2–20% more than convective heating systems, whereas the heating load of the lightweight and in-slab floor heating is 15–40% and 20–67% more than all-air systems respectively. In addition, they confirm that convective systems are more suitable for intermittent heating operation. Other researchers recently looked at the cooling design of radiant system. Ning et al. (2020) investigated numerically how to split the cooling load of a radiant ceiling and a dedicated outdoor air system used to supply fresh air and to dehumidify the air room. Krajčík and Šikula (2020) analysed four types of radiant wall to satisfy the cooling load in new and retrofitted existing buildings; their analysis did not consider the dynamics of the entire room, but was focused only on the single radiant structures using both steady state and transient simulations.

As the literature points out, space heating and cooling of residential buildings account for a great part of world energy use. Besides moving to low energy buildings, a great challenge concerns also ensuring a thermally comfortable indoor environment while maintaining a high indoor air quality. Summarizing the main results of the studies comparing all-air systems with radiant systems, it can be noticed that they generally take into account office buildings and many of them focus only on cooling aspects. Residential buildings are quite different from offices, in terms of user occupation, internal gains, ventilation requirements, and therefore they deserve dedicated analysis. Moreover, only some studies consider the effect of different climates in the analysis. This work aims to compare both in heating and cooling mode an all-air system and a radiant floor system in a residential dwelling with high energy performance of the envelope. The investigated all air-system is able to satisfy the heating and cooling demand of residential buildings with low energy consumption without other backup devices, whereas a radiant floor needs to be coupled with a system providing fresh air and controlling indoor humidity in summer. The comparison is performed by means of integrated dynamic simulations of the building and HVAC systems, considering also the auxiliaries and paying particular attention to ensure the same indoor conditions. In order to understand if the system is suitable and convenient for different climate conditions, the comparison is carried out in terms of both indoor conditions and energy consumption in three locations: Helsinki (i.e. with a dominant heating climate), Rome (i.e. with a dominant cooling climate), and Milan (i.e. with an almost balanced heating and cooling climate). The building considered as case study for the simulations is an existing building where the investigated all-air system is actually installed with a dedicated monitoring system (Lazzarini et al. 2019a, Lazzarini et al. 2019b) in order to test its control strategies.

METHOD

Overview of the two systems

The investigated all-air system is able to provide the heating and cooling demand of a small residential building with a high-quality envelope. The system (Figure 1) is equipped with a heat recovery unit, an air-water finned heat exchanger and fans and can provide both fresh air and air recirculation depending on the operating conditions. Fresh air is supplied according to a user's schedule and sensors placed in the controlled zones to improve the indoor air quality and to control relative humidity and/or CO_2 levels. The system needs to be coupled to a heat pump which produces hot and cold water.

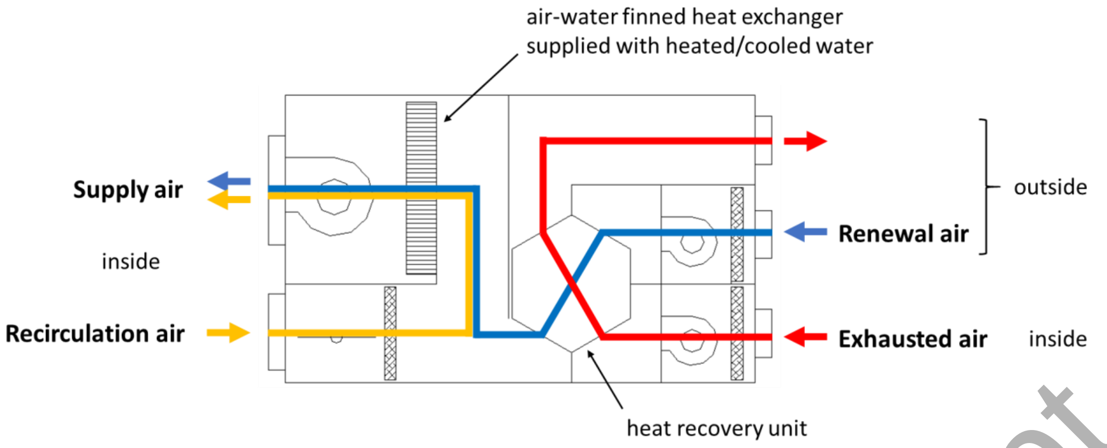


Figure 1. Scheme of the investigated all-air system.

The operating modes of the all-air system are the following:

- *renewal*: the system provides only fresh air, which flows through the heat recovery unit where it exchanges heat with the exhausted air, while the air-water finned heat exchanger is not supplied with heated/cooled water; the heat recovery unit can be also by-passed to ensure free cooling;
- *recirculation with thermal integration*: the system heats or cools (and dehumidifies) the indoor air flowing through the air-water finned heat exchanger, without providing fresh air;
- *renewal and recirculation with thermal integration*: renewal air from the heat recovery unit and recirculation air are mixed and heated or cooled (and dehumidified) flowing through the air-water finned heat exchanger.

The main technical data of the all-air system are shown in Table 1. The size of the chosen machine is consistent with the heating and cooling loads of the case study presented afterward.

Table 1. Technical data of the investigated all-air system.

Heating thermal capacity	4.0 kW
Total cooling thermal capacity	3.2 kW
Sensible cooling thermal capacity	2.4 kW
Water volume flow rate	550 L/h
Energy efficiency of the heat recovery in heating mode (ambient air at 20°C and 50%relative humidity; external air at -5°C and 80% relative humidity)	90%
Energy efficiency of the heat recovery in cooling mode (ambient air at 27°C and 50%relative humidity; external air at 35°C and 50% relative humidity)	85%
Nominal external air volume flow rate	200 m^3/h
Nominal air volume flow rate (renewal + recirculation)	500 m^3/h

Besides the model of the presented all-air system, another system has been modelled in order to compare the indoor conditions and the energy performance. For this purpose, a radiant floor system, coupled with an isothermal dehumidifier in summer and with controlled mechanical ventilation (CMV) ensuring fresh air, has been considered. A schematic overview of the two systems is shown in Figure 2.

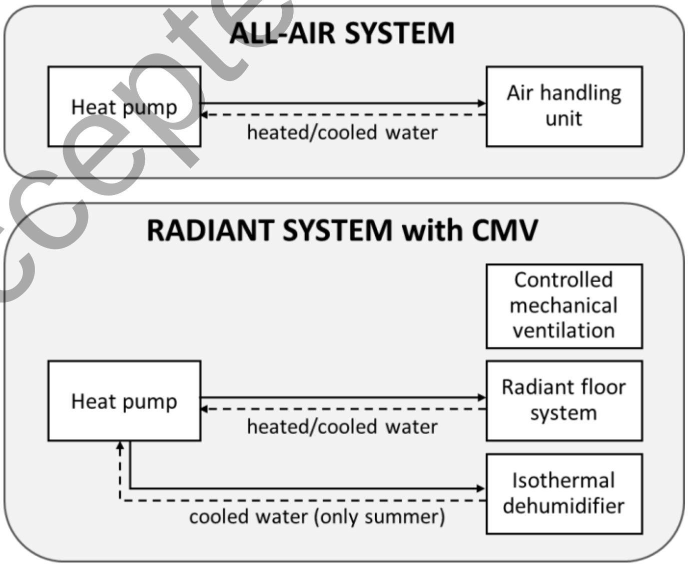


Figure 2. Schematic overview of the two systems under comparison.

The two HVAC systems have been modelled using the Simulation Studio tool of TRNSYS (Solar Energy Laboratory 2012), which is a software to simulate the behaviour of transient systems. To analyse and compare their performance, a small multizone building has been considered as case study and has been modelled in detail using the TRNBuild tool. The behaviour of the building and of the HVAC system are strongly linked and for this reason they are simultaneously simulated, using a calculation time step of 15 minutes to properly consider their dynamics.

Model of the all-air system

An overview of the model built using the Simulation Studio tool is shown in Figure 3. The all-air system model takes as input variables set or calculated in other components (called *Types*), like weather data from *Type 15.3* (external air temperature and relative humidity), seasonal parameters from *Type 9e* (indicators for heating-cooling and mid-seasons, heat recovery unit efficiency in the different seasons, design recirculation air flows) and thermostat states from *Types 2b*. Moreover, the all-air system model exchanges data also with the building model (*Type 56*); in detail, it takes as input the air temperature and the humidity ratio of the rooms from which recirculated and exhaust air are extracted and it provides as output the mass flow rates, the temperature and the humidity ratio of the air to be supplied to each room. The all-air system has been modelled employing *Type 9e* to set the renewal air flows (according to a fixed time schedule), *Type 760* for the heat recovery unit and different calculators, used for the finned coil, for the air extraction from the rooms and for the mixing of renewal and recirculation air. Calculators are very flexible modules, since they can be used to define variables and set their values according to the outputs of other *Types*. In each calculator, equations using also Boolean expressions are used, enabling to reproduce the behaviour of the component as well as controllers.

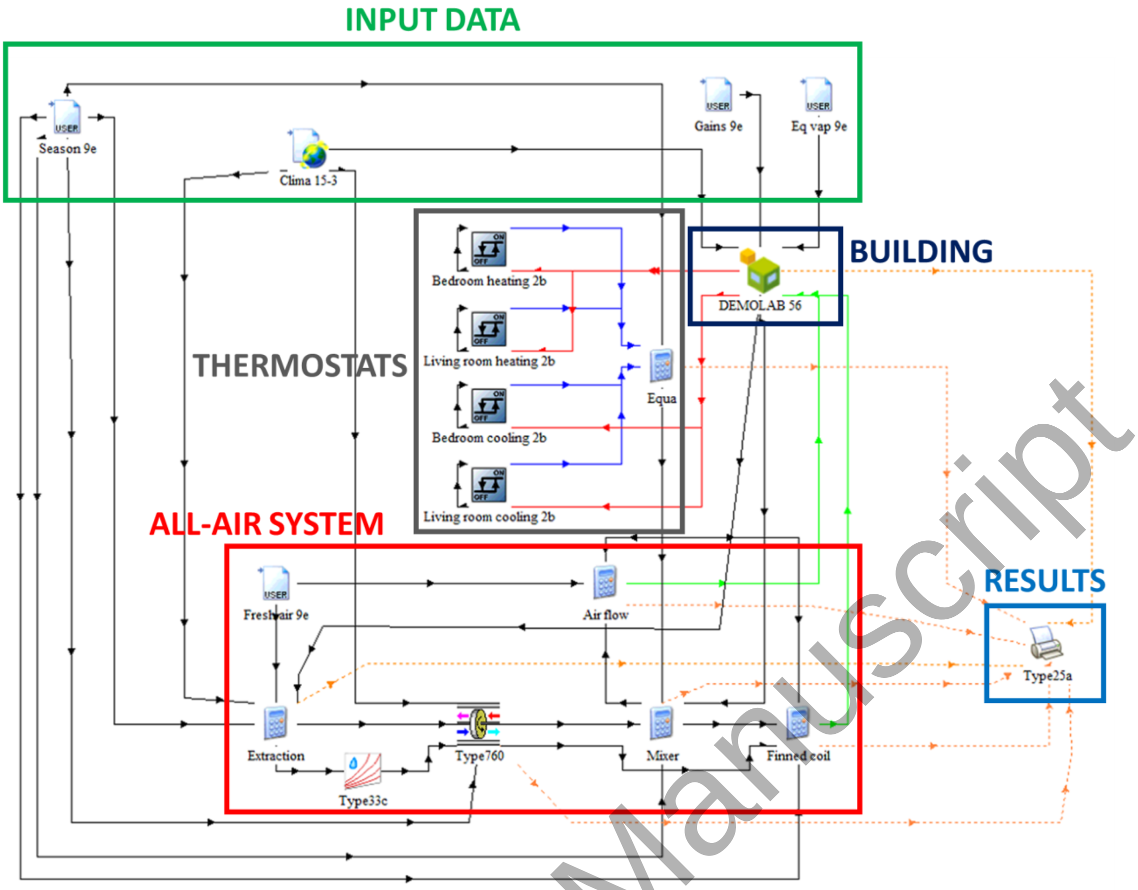


Figure 3. The model of the all-air system as it appears in Simulation Studio.

The workflow of the all-air system and the fundamental equations (basically mass and energy balances) used to describe the main components are shown in Figure 4. When renewal is set (according to a time schedule), fresh air moves through a cross-flow heat recovery unit, where it exchanges heat with the exhaust air (i.e. the mixed air extracted from the rooms) considering a set thermal efficiency ε. The case when condensation occurs in the fresh air stream flowing in the heat recovery unit is also considered, resulting in a reduction of the humidity ratio. Moreover, in summer the heat recovery unit can be by-passed to provide free cooling if the external air temperature is lower than the extracted air temperature. When heating, cooling or dehumidification is needed (according to the thermostats' states), the recirculation mode takes an indoor air flow and the eventual fresh air through an air-water finned heat exchanger, which heats or cools (and dehumidifies at the same time) the air supplied to the rooms. The air exits the finned coil to be supplied to the rooms at a fixed temperature (defined according to the season). In summer latent cooling is also provided by the finned coil. If only renewal is needed, the finned coil is not supplied with hot/cold water and the air is supplied to the building at the same temperature it exits from the heat recovery unit.

The operating mode of the all-air system (only renewal, only recirculation, renewal and recirculation) is defined by control variables set according to the renewal time schedule, to the thermostats' states and to seasonal parameters. Control variables are used also to manage the air flows supplied to the two thermal zones.

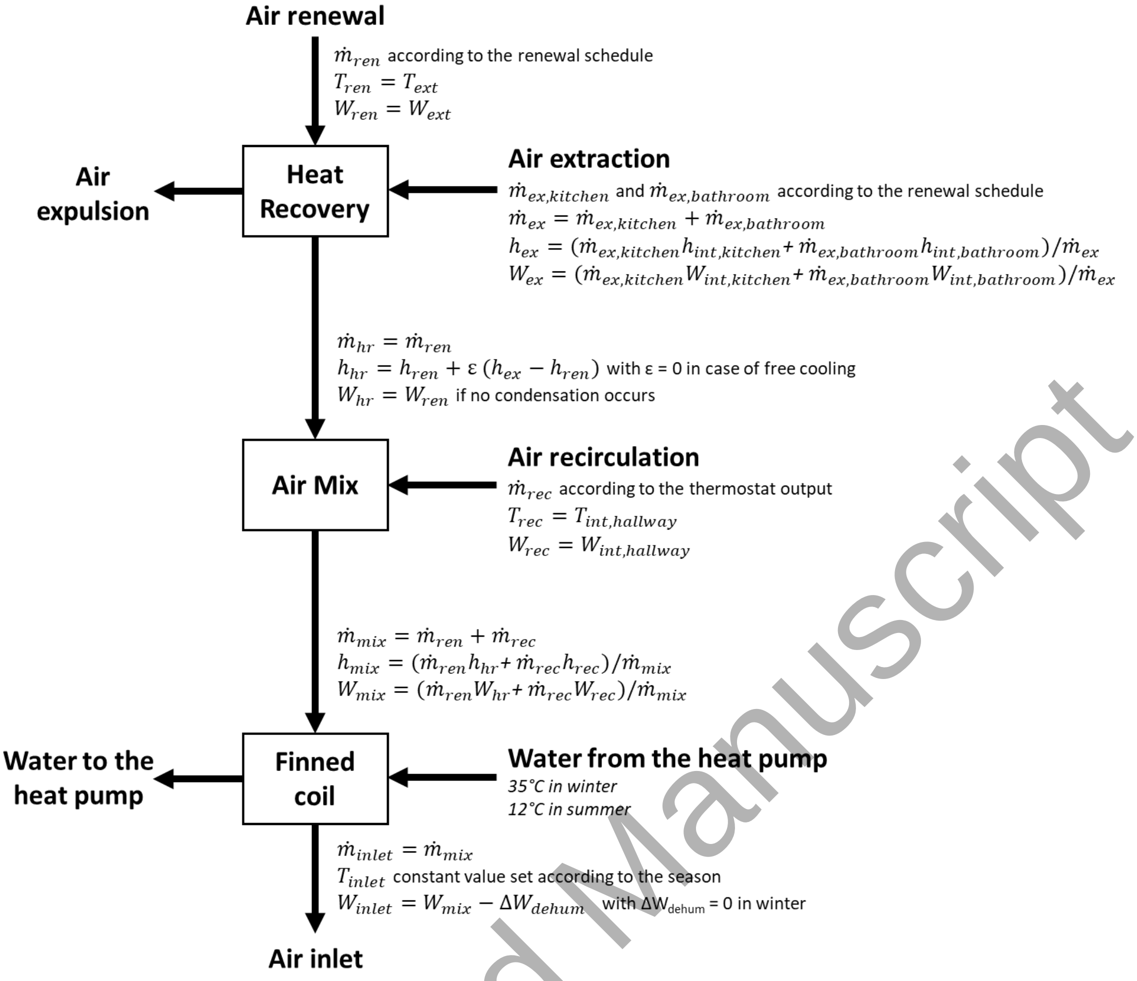
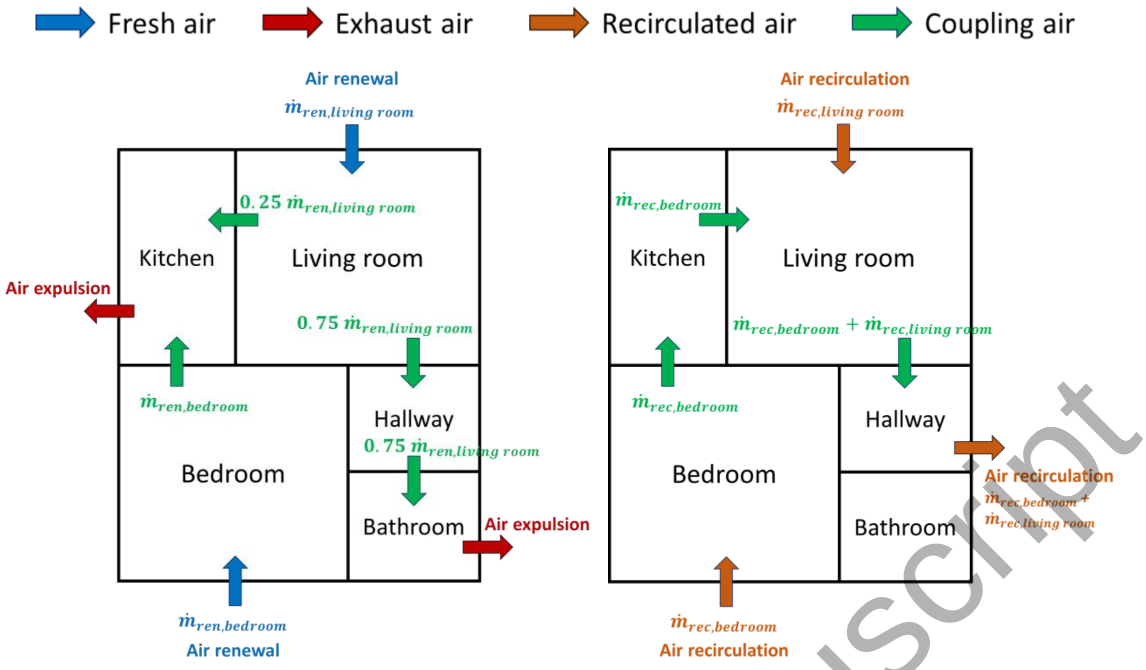


Figure 4. Scheme of the workflow of the modelled all-air system when both renewal and recirculation are operating.

The all-air system processes and distributes to the thermal zones of the building the renewal and recirculation air flows, which are defined as variables inside the building model; at each simulation time step the proper values of the variables are set and provided as input to the building model (*Type 56*). The possibility to control more than one thermal zone has been implemented in the model. A thermostat has been defined for each of the main rooms and the total recirculation air flow is set depending on the thermostats' states. When all the controlled thermal zones require heating/cooling, the all-air system processes the total recirculation air flow (together with the design renewal air flow, when needed), otherwise it processes only the recirculation air flow of the zone requiring air conditioning.

An important issue is the proper modelling of the resulting air flows inside the building, which are set inside TRNBuild tool (Solar Energy Laboratory 2012). Fresh air is supplied to the living room and to the bedroom, while exhaust air is extracted from the kitchen and the bathroom; the resulting air flows through the doors of the adjacent thermal zones, which in TRNSYS are called *coupling air flows*, are shown in Figure 5a. The air flows shown in the figure are calculated considering the design values of the supply and extraction air flow rates of the specific case study presented afterwards. When recirculation mode is on, the air is extracted from the hallway and, after conditioning, it is supplied to the living room and to the bedroom; the resulting air flows are shown in Figure 5b. When both renewal and recirculation modes are on, the superimposition principle is applied: the resulting *coupling air flow* between each couple of rooms is the sum of the *coupling air flow* due to renewal and the *coupling air flow* due to recirculation. When neither renewal nor recirculation air are needed, no coupling air flow is considered, since in new buildings infiltrations are low (particular attention is paid on the envelope airtightness) and therefore air movements between adjacent rooms are negligible.



(a) Air flows due to renewal (b) Air flows due to recirculation

Figure 5. Air flows modelling of renewal (a) and recirculation (b) modes.

As regards the auxiliaries of the modelled all-air system, the electric consumption of the renewal and recirculation fans has been evaluated for all the possible operating conditions using the fan performance curves with the operating renewal and recirculation air flow rates and the design pressure head needed.

Model of the radiant floor system with controlled mechanical ventilation

The considered heating and cooling system is a water based embedded floor system defined using TRNBuild inside the multi-zone building model (Solar Energy Laboratory 2012). An “active layer” is used for this purpose, placed between an insulation layer and a cement screed. The layer is called “active” because it contains fluid filled pipes that either add or remove heat from the surface. The following parameters of the active layer are specified inside each thermal zone: fluid specific heat capacity, pipe spacing, internal and external pipe diameter, pipe conductivity and number of loops. The heat carrier fluid mass flow rate of each zone is provided as input to the building model (*Type 56*) at each simulation time step, since it is controlled by the thermostat state. Also the inlet fluid temperature is provided as input: for heating purposes it depends on the external air temperature (climatic control), while it is set constant during the cooling season. As regards the auxiliaries of the radiant floor system, the electric consumption of the circulating pump has been evaluated on the basis of the design mass flow rate and pressure head using manufacturers' performance curves.

The CMV system is provided with a cross-flow heat recovery unit with the same energy performance as the one in all-air system and with the possibility to be by-passed to get free cooling in summer. This system has been modelled in TRNSYS in the same way of the renewal mode of the all-air system. Accordingly, the building model has been modified considering only the coupling air flows shown in Figure 5a.

As regards dehumidification, an isothermal system has been considered. Inside the isothermal dehumidifier the indoor air is first pre-cooled by an air-water finned heat exchanger, then it is dehumidified in the evaporator of a vapor-compression refrigeration cycle and finally heated in the condenser: the air temperature is the same at the beginning and at the end of the process. The isothermal dehumidifier was modelled as a system which reduces the humidity ratio of a specified air flow rate. To avoid condensation on the cooling radiant floor system, the system turns on when the relative humidity exceeds a specified set-point in at least one of the thermal zones. The air flow rate, the amount of humidity which condensates and the temperature level of the refrigerated water are set according to the technical data of a proper machine for residential use.

Heating and cooling generation system

In order to perform a comprehensive energy comparison of the all-air system and of the radiant floor system with controlled mechanical ventilation, an air-to-water heat pump has been considered to provide heated/cooled fluid to the all-air system and to the radiant system. The heat pump has not been included as component in the TRNSYS model; its electric consumption has been calculated from the simulation results using the energy efficiency curves (i.e. COP in heating mode, EER in cooling mode) shown in Figure 6. The curves, which are function of the air source-sink temperature and supply water temperature, are taken from the datasheet of a commercial air-to-water heat pump in compliance with EN 14511 (CEN 2018), therefore these values consider also energy consumption due to the defrost cycles and auxiliaries. Second-order functions resulting applying polynomial regression have been considered in the analysis for different temperature of water production (T_w).

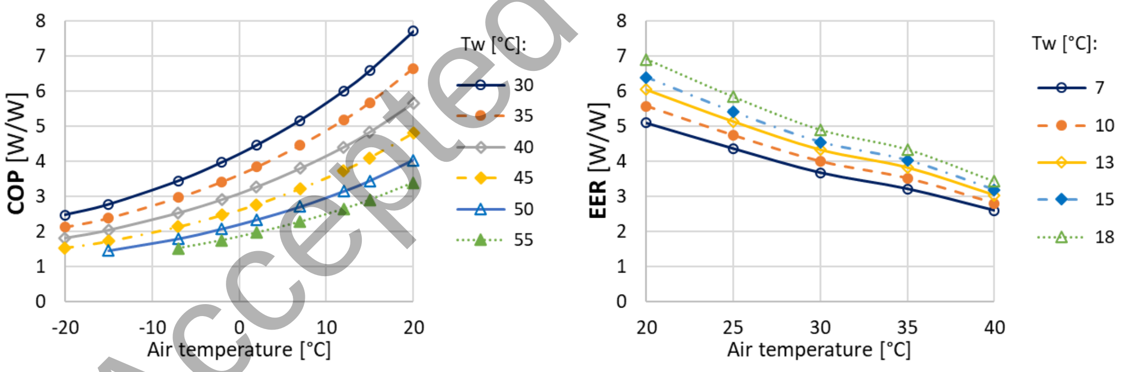


Figure 6. COP and EER curves of the air-to-water heat pump considered in the analysis for different temperature of water production T_w.

CASE STUDY

The planimetry of the case study is shown in Figure 7; it represents a real building where the modelled all-air system is installed (Lazzarini et al. 2019a, Lazzarini et al. 2019b) and it has been chosen for the possibility of future on site measurements. The dimensions of the building (net floor area 57.2 m^2, net volume 154.6 m^3) are representative of a small apartment with a living room, a kitchen, a bedroom, a bathroom and a technical room. The East and the West side of the building are external, while the North and the South walls as well as the ceiling and the floor are adiabatic surfaces. The thermal transmittance of the external walls, equal to 0.27 W/(m^2 K), is typical of a high energy performance building. The thermal transmittance of the windows is 1.4 W/(m^2 K) and an external shading of 50% is considered.

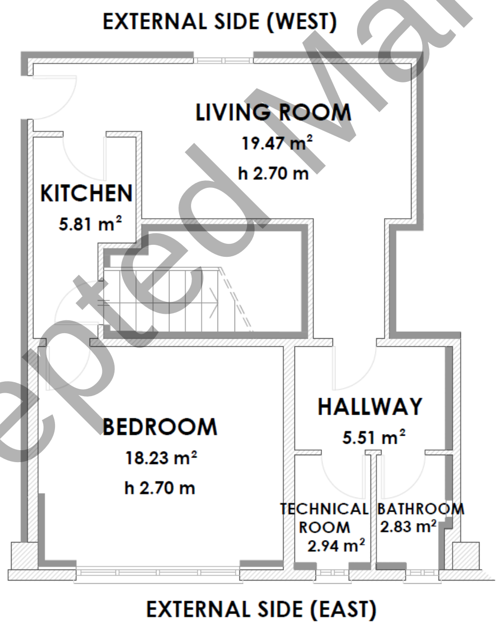


Figure 7. Planimetry of the case study.

Yearly computer simulations of the all-air system and of the radiant floor system with CMV have been performed using a calculation time step of 15 minutes. Three locations have been considered in the analysis: Helsinki, Milan and Rome; IWEC weather data (ASHRAE 2001) has been used. According to Köppen-Geiger climate classification (Kottek et al. 2006), Milan is *Cfa* (warm temperate, fully humid, hot summer), Rome is *Csa* (warm temperate, summer dry, hot summer), Helsinki *Dfb* (boreal, fully humid, warm summer). The seasons set in the simulations are shown in Table 2: heating is generally not necessary for the entire period defined as *heating season* and the heat recovery on the renewal air could be sufficient to guarantee comfortable indoor temperature, while during the period defined as *cooling season* the renewal air is supplied in free cooling mode when the external air temperature is lower than the extracted air temperature.

Table 2. Locations considered in the analysis: minimum and maximum external air temperature, humidity ratio during the hottest day, annual global horizontal insolation and seasons set in the simulations.

Location	**T_{min} [°C]**	**T_{max} [°C]**	**W* [g_v/kg_{da}]**	**GHI [kWh/m^2]**	**Heating season**			**Cooling season**		
Helsinki	-21.6	28.7	8.5	947	1st September	-	30th April	1st July	-	15th August
Milan	-10.9	32.6	12.5	1292	15th October	-	15th April	1st June	-	15th September
Rome	-4.0	31.7	14.2	1462	1st November	-	31st March	15th May	-	15th October
* mean value during the hottest day of the year, i.e. when T_{max} occurs										

The objective of the present work is the comparison of the energy consumption of the all-air system with that of a radiant floor system with CMV. While all-air systems have an effect mainly on air temperature, radiant systems act on both air temperature and mean radiant temperature. Human thermal comfort responds to operative temperature, therefore this is the most suitable parameter to be controlled to be sure that the systems analyzed provide similar comfort conditions to the occupants. Two thermal zones have been defined in the building model, corresponding with the two main rooms (i.e., the living room and the bedroom), where the operative temperature is set to 20.5 °C in winter and 25.5 °C in summer, with an on-off band of ± 0.5°C.

Hourly profiles for internal sensible and latent gains have been defined for each room according to current standards (EN 16798-1, CEN 2019) and typical occupants' presence. The hourly profiles for the different rooms give a mean value of sensible heat gains equal to 5 W/m^2 and an overall vapour load equal to 3.7 kg/day (1.0 kg/day in the living room and in the kitchen, 1.2 kg/day in the bedroom and 0.5 kg/day in the bathroom).

Since controlled mechanical ventilation is provided and the envelope performance is typical of a new building, where particular attention is paid on air tightness, 0.1 Vol/h has been set as constant value for air infiltrations. According to Annex C of EN 16798-1 (CEN 2019), during occupied hours minimum 0.5 L/(s m^2) of fresh air should be considered for energy calculation. The renewal air flow rate is therefore set to 110 m^3/h (1.9 Vol/h) during occupied hours (from 6 pm to 8 am); a reduced renewal air flow rate has been set in the two hours before and in the 2 hours after the occupied period (75% and 50% of the design air flow rate respectively). The renewal air is equally supplied to the living room and to the bedroom. The efficiency of the heat recovery unit and the electrical consumption of the fans are shown in Table 3.

Table 3. Data for the model of the renewal section of the all-air system and for the CMV system used with the radiant system.

Air flow rate during occupied hours	110 m^3/h
Fan electrical power (110 m^3/h, 150 Pa)	2 x 30 W
Heat recovery energy efficiency, heating	90%
Heat recovery energy efficiency, cooling	85%

As regards the all-air system, the recirculation air flow rate has been set to 1.5 Vol/h in winter and 2.0 Vol/h in summer, with a constant supply air temperature of

30°C and 20°C respectively; the temperature of supply air has been chosen to ensure comfort to the occupants. The air flow rates and the electric consumption of the fan are shown in Table 4.

Table 4. Data for the model of the recirculation section of the all-air system.

	Supply air temperature [°C]	**Air flow rates [m^3/h]**		**Fan electric power [W]**	
		2 zones	**1 zone**	**2 zones**	**1 zone**
Heating	30	150	75	33	17
Cooling	20	200	100	43	23

As regards the radiant system, it has been sized according to the heating load; the following parameters have been used for the *active layer*: 0.15 m pipe spacing, pipes with 20 mm outer diameter and 2 mm thickness with a thermal conductivity of 0.35 W/(m K), water as heat-carrier fluid. The stratigraphy of the radiant floor system and the properties of its layers are shown in Figure 8. The supply water temperature, the total flow rates and the electric consumption of the pump are shown in Table 5. For heating purposes, this climatic control is used: when the external air temperature is below 0°C the maximum supply temperature is used, then it varies linearly up to 15°C, when the minimum supply temperature is used. For cooling purposes, the supply water temperature is constantly set to 18°C. To avoid condensation on the cooled floor surface in summer and to ensure comfortable indoor conditions, the maximum relative humidity of the two thermal zones is set to 55%. The data used in the model of the dehumidifier are shown in Table 6.

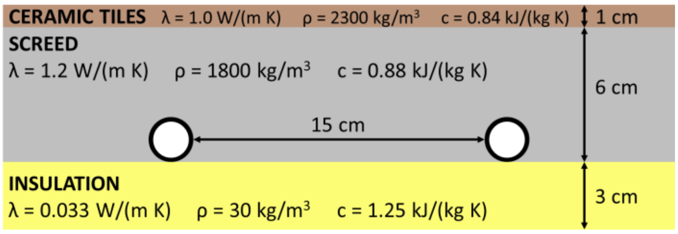


Figure 8. Stratigraphy of the radiant floor system and properties of its layers.

Table 5. Data for the model of the radiant floor system.

	Supply water temperature [°C]			**Water flow rate [kg/h]**	**Pump electric power [W]**
Locations	**Heating**		**Cooling**		
	max	**min**			
Helsinki	30	25	18	720	11
Milan, Rome	28	25	18	450	9

Table 6. Isothermal dehumidifier technical data.

Air flow rate		200 m^3/h
Dehumidification capacity	*	21.0 L/24h
Latent cooling power	*	609 W
Required refrigerator power (pre-cooling)	*	828 W
Electrical power	*	249 W
* refrigerated water 12°C, ambient air 26°C and 55% relative humidity		

As regards the heat pump, the COP and EER curves shown in Figure 6 are used to calculate the electric energy consumption at each time step of the simulation. For the all-air system, the finned heat exchanger is supplied with water at 35°C in heating and 12°C in cooling. For the radiant system with CMV, water is assumed to be supplied at 30°C in heating in all the climates. In summer, the radiant floor system is supplied with water at 18°C, while the dehumidifier requires water at 12°C. Two options are analysed to verify if an energy saving can be obtained controlling the operating conditions of the air-to-water heat pump: (i) in the first case the heat pump supplies water at 12°C, (ii) in the second one it provides water at 12°C when dehumidification is needed and at 18°C when only the radiant floor system is supplied.

RESULTS AND DISCUSSION

Indoor conditions

The all-air system and the radiant floor system coupled with CMV have been simulated setting the same operative temperature to ensure comparable comfort conditions. The distribution of the running operative temperature has been carefully analysed during the heating and cooling seasons; to this purpose, the actual heating/cooling period is set between the first time the heating/cooling system switches on and the last time it switches off. In general, with the all-air system the operative temperature is inside the set temperature band (20-21 °C in winter, 25-26 °C in summer) for more time than that with the radiant floor system.

During the heating season, the all-air system presents a very good control of the operative temperature, which is inside the set-band for almost all the time (Figure 9). In the living room, the operative temperature is above the set-band for a higher amount of time with the radiant floor system than with the all-air system, as effect of the higher variability of the internal gains in that room combined with the inertia of the system. The operative temperature falls below the desired set band only in the case with the all-air system in the coldest of the three locations, i.e. Helsinki: this happens in the living room for 93 hours (less than 2% of the heating season).

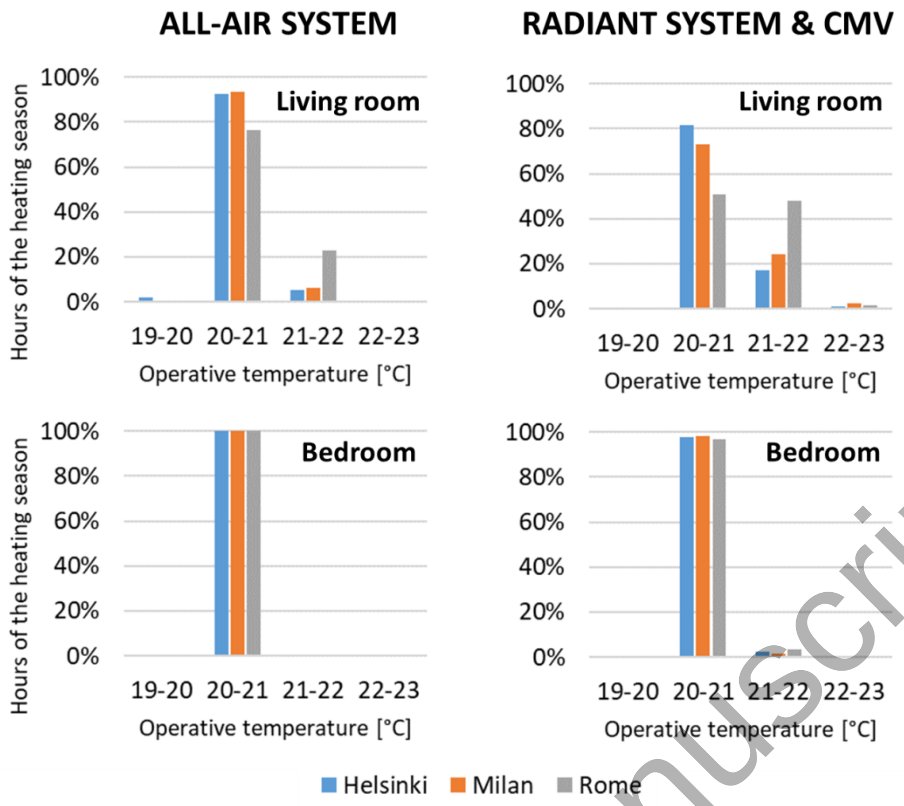


Figure 9. Operative temperature distribution during the heating season for the living room and the bedroom. The set temperature band is 20-21 °C.

During the cooling season the operative temperature is below the set-band for a consistent amount of time with both the systems (Figure 10), as effect of free cooling due to renewal during the night-time; this can be noticed especially in the living room, where internal heat gains are significantly lower than in the bedroom during the night. The thermal inertia of the radiant floor system makes it slower in adapting to changes in internal gains and external conditions and this makes the temperature distribution more spread. The operative temperature remains above the desired set band only in the case with the all-air system: this happens in the living room for 134 hours in Milan and 123 hours in Rome (about 5% and 4% of the cooling season respectively).

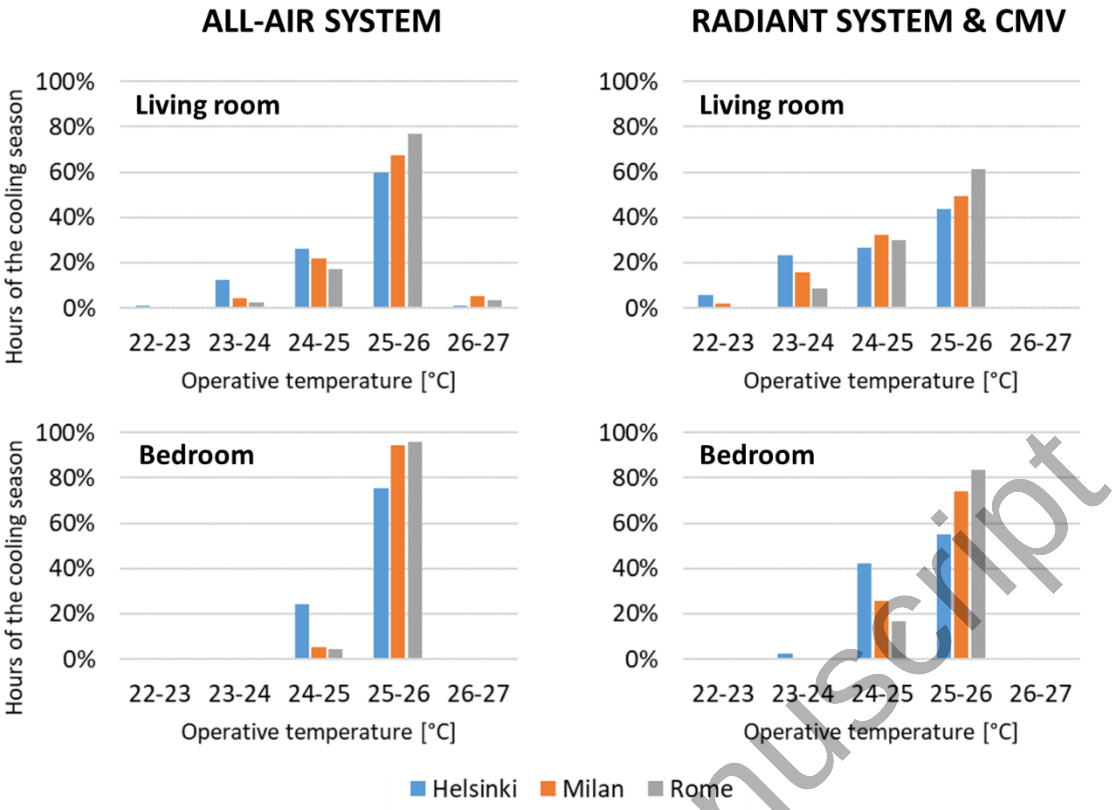


Figure 10. Operative temperature distribution during the cooling season for the living room and the bedroom. The set temperature band is 25-26 °C.

Figure 11 shows the temperature trend in the living room during a winter day in Milan. The heat flow P_w exchanged by the water inside the pipes of the radiant floor system and in the finned coil of the all-air system is also shown, along with the renewal air flow rate, which is in heat-recovery mode. The set-point band for the operative temperature is also highlighted in the figure: in the considered day, the two heating systems switch on almost at the same time during the night starting from similar air and mean radiant temperature values. In the case of the radiant floor system, the mean radiant temperature and the air temperature rise slowly, with a similar trend and are very close, even after the system switches off; with the all-air system, the air temperature rises very fast up to about 1°C above the mean radiant temperature, which keeps almost unchanged when the system is working. From 6:30 to 10:00 the temperatures keep quite stable with both the systems (the thermal losses due to air renewal are balanced by the internal heat gains). From 10:00 to 16:00, the temperatures decrease in the case with the radiant system, while in the case with the all-air system they keep stable because in the bedroom heating is on and the recirculation air flows back to the all-air system through the living room. From 19:00 to 23:00 the internal heat gains make air temperature to rise faster than mean radiant temperature in both the cases.

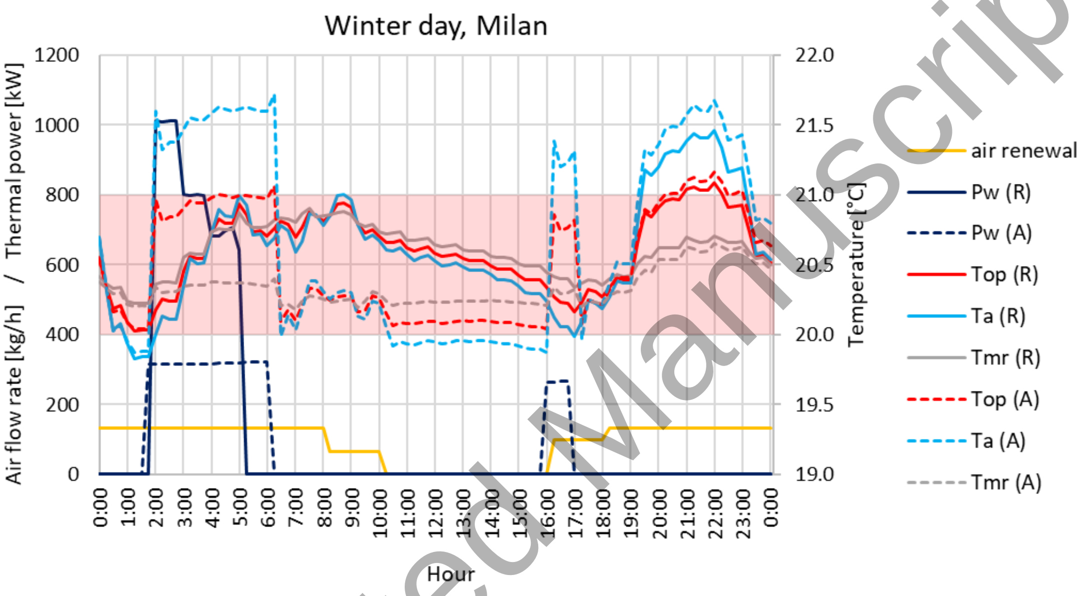


Figure 11. Example of operative T_{op}, air T_a and mean radiant T_{mr} temperature trend inside the living room during a winter day (20th December) for the all-air system (A) and for the radiant floor system with CMV (R).

Figure 12 similarly shows the temperature trend in the living room during a summer day in Milan. In the two hours after the systems switch on the temperature trend is very similar, then temperature decreases faster in the case with the radiant system. Air renewal is operating in free cooling mode from 18:00 or 19:00 (in the case with the all-air system and radiant system respectively) to 10:00 and for this reason the indoor air temperature continues to decrease until 6:00. From 6:00 to 9:00 internal heat gains and external temperature increase and the temperatures rise with the same rate for the two systems, equal to 1.9°C and 0.8°C in 3 hours for air temperature and mean radiant temperature respectively.

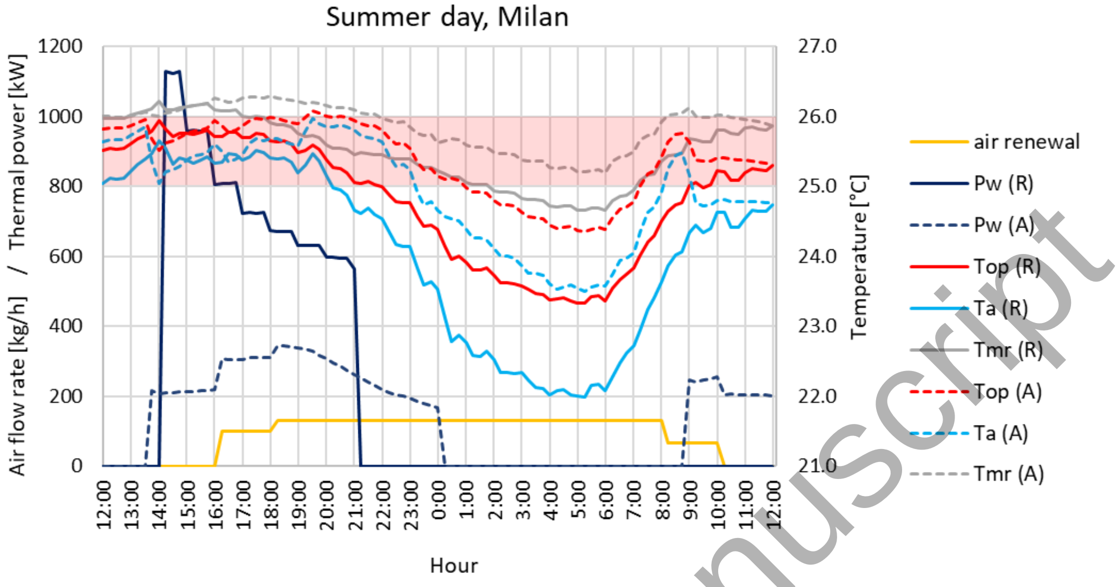


Figure 12. Example of operative T_{op}, air T_a and mean radiant T_{mr} temperature trend inside the living room during a summer day (4th-5th July) for the all-air system (A) and for the radiant floor system with CMV (R).

As regards indoor humidity in summer, in the case of the radiant system a dehumidifier constantly keeps relative humidity under 55%. In the case with the all-air system, sensible cooling and dehumidification of recirculated air are interdependent: it is not possible to dehumidify without cooling the rooms, since there is only one heat exchanger. Nevertheless, the resulting indoor humidity is never above 65% in Helsinki, while in Milan and Rome humidity slightly exceeds 70% for about 5% and 9% of the length of the cooling season respectively. This happens only in the first hour of the morning, when renewal is not able to significantly reduce the internal latent gains of the bedroom. Free cooling occurs during these hours and the indoor air temperature is often below 24°C.

Energy analysis

The results of the simulations of the all-air system (*system A*) and of the radiant system with controlled mechanical ventilation (*system R*) are presented in this subsection for heating, cooling and ventilation.

Table 7 summarizes the results for the heating season. The thermal energy Q_{th} represents the energy exchanged by the finned coil of the all-air system or by the water flowing inside the radiant floor. The all-air system requires 38% less thermal energy than the radiant system coupled with CMV in Helsinki, 47% in Milan and 80% in Rome. As it can be seen, the heat recovery unit allows a consistent energy saving (Q_{hr}); with the all-air system Q_{hr} value is 30% higher than that with the radiant system in Helsinki and Milan, 35% higher in Rome. This difference is due to the different heat transfer mechanism of the two systems: the all-air system acts on the air temperature, while the radiant floor acts also on the mean radiant temperature, and therefore ensures the same operative temperature with a lower air temperature. The electric energy consumption of the air-to-water heat pump is also shown, along with the seasonal coefficient of performance, which is calculated considering also the auxiliaries (recirculation fan of the all-air system, circulation pump of the radiant floor). The seasonal energy efficiency in heating mode SCOP, calculated as the ratio between the thermal energy Q_{th} provided by the heat pump and the total electrical energy consumption $E_{el,tot}$, is lower for the all-air system than for the radiant system, because both the water temperature level and the consumption of the auxiliaries are higher. Nevertheless, the total electric energy consumption of the all-air system is always lower than that of the radiant floor: the difference is 19% in Helsinki, 32% in Milan and 74% in Rome.

Table 7. Results of the simulations for heating.

	Q_{th}	Q_{hr}	$E_{el,hp}$	$E_{el,aux}$	$E_{el,tot}$	ΔE_{el}	SCOP
	[kWh]	**[kWh]**	**[kWh]**	**[kWh]**	**[kWh]**	**(A-R)/R**	
Helsinki, A	1504	2821	483	65	548	-19%	2.75
Helsinki, R	2427	2160	661	18	679		3.57
Milan, A	626	1802	177	26	203	-32%	3.08
Milan, R	1182	1379	291	10	300		3.94
Rome, A	54	1147	14	2	16	-74%	3.48
Rome, R	265	845	57	2	59		4.49
A: all-air system; R: radiant system with CMV							

The results for cooling are shown in Table 8. For the radiant floor with CMV the possibility in which the heat pump can work at two different supply temperatures is also shown (i.e., *case R**). In the case of the radiant floor the thermal energy Q_{th} includes the energy removed by the water flowing inside the pipes and the latent load removed by the dehumidifier. The all-air system removes 47% more thermal energy than the radiant system in Helsinki, 30% in Milan and 17% in Rome. The free cooling ensured by renewal air bypassing the heat recovery unit (Q_{fc}) ensures a significant energy saving in all the locations. The electric energy consumption of the heat pump (which feeds also the dehumidifier), the dehumidifier and the auxiliaries are also presented: the total electric energy consumption of the two systems is almost the same in Helsinki, while in Milan and Rome the all-air system consumes respectively 14% and 29% less than the radiant floor system. The seasonal energy efficiency in cooling mode SEER, calculated as the ratio between the sensible and latent thermal energy removed by the refrigerated water and by the dehumidifier (Q_{th}) and the total electrical energy consumption ($E_{el,tot}$,), is significantly lower for the radiant system with dehumidifier (2.9 in Helsinki, 2.5 in Milan, 2.3 in Rome) than for the all-air system (4.1 in Helsinki, 3.8 in Milan and Rome) because of the low performance of the dehumidifier. In fact, if the contribution of the dehumidifier is not considered, the seasonal energy efficiency value $SEER_{rad}$ is higher than the previous one (4.9 in Helsinki, 4.6 in Milan and Rome). Producing refrigerated water at a higher temperature when the isothermal dehumidifier is not working (*case R**) has no significant impact on the total electric consumption, since the latent load in summer is dominant for most of the time.

More details about the energy balance of the isothermal dehumidifier are shown in Table 9: the latent energy removed by the air stream ($Q_{th,lat}$), the electricity consumed by the compressor and the fan ($E_{el,dehum}$), the refrigerated energy supplied by the heat pump feeding the dehumidifier ($Q_{w,hp}$) and the related electric energy consumed by the heat pump ($E_{el,hp}$).

Table 8. Results of the simulations for cooling.

	Q_{th}	**Q_{fc}**	**$E_{el,hp}$**	**$E_{el,dehum}$**	**$E_{el,aux}$**	**$E_{el,tot}$**	**ΔE_{el}**	**SEER**	**$SEER_{rad}$**
	[kWh]	**[kWh]**	**[kWh]**	**[kWh]**	**[kWh]**	**[kWh]**	**(A-R)/R**		
Helsinki, A	-254	-294	53	-	9	62	+4%	4.08	-
Helsinki, R	-172	-290	34	24	1	60		2.88	4.92
Helsinki, R*	-172	-290	31	24	1	56		3.04	5.70
Milan, A	-1367	-446	308	-	48	356	-14%	3.84	-
Milan, R	-1052	-442	234	174	5	414		2.54	4.56
Milan, R*	-1052	-442	219	174	5	398		2.64	5.14
Rome, A	-2237	-549	505	-	79	584	-29%	3.83	-
Rome, R	-1918	-558	436	374	9	820		2.34	4.63
Rome, R*	-1918	-558	417	374	9	800		2.40	5.08

A: all-air system; R: radiant system with CMV
*: refrigerated water produced at a higher temperature when the isothermal dehumidifier is not working

Table 9. Results of the simulations for cooling: details on the isothermal dehumidifier.

	$Q_{th,lat}$	**$E_{el,dehum}$**	**$Q_{w,hp}$**	**$E_{el,hp}$**	**$E_{el,tot}$**
	[kWh]	**[kWh]**	**[kWh]**	**[kWh]**	**[kWh]**
Helsinki, R	-59	24	-80	13	37
Milan, R	-425	174	-578	102	276
Rome, R	-915	374	-1244	229	603

The yearly electric consumption of the two systems for heating, cooling and ventilation are shown in Table 10. The all-air system ensures lower electrical consumption than the radiant floor system coupled with CMV in all the considered locations: the energy saving is 12% in Helsinki, 14% in Milan and 23% in Rome.

Table 10. Overall results in terms of yearly electric consumption.

	Electric consumption [kWh/year]				**ΔE_{el}**
	Heating	**Cooling**	**Ventilation**	**Total**	**(A-R)/R**
Helsinki, A	548	62	361	971	-12%
Helsinki, R	679	60	361	1100	
Milan, A	203	356	361	920	-14%
Milan, R	300	414	361	1075	
Rome, A	16	584	361	961	-23%
Rome, R	59	820	361	1240	
A: all-air system; R: radiant system with CMV					

CONCLUSIONS

The use of proper ventilation systems is necessary to ensure a good indoor air quality, especially in buildings characterized by high-energy performance. A suitable all-air system could be more flexible than radiant systems, since a single system provides fresh air and satisfies thermal and latent loads at the same time. Moreover, when the heating and cooling loads are relatively low, thermal comfort can be maintained with low air flow rates, thus avoiding local discomfort conditions.

In this work, a multi-zone system based on a small all-air system and a radiant floor system coupled with mechanical controlled ventilation were compared both in heating and cooling mode. The analysis was carried out in three locations (Helsinki, Milan and Rome) by means of TRNSYS simulations of the two systems, considering as case study a small flat of 57 m^2 characterized by low energy demand. To ensure comparable comfort conditions with the two systems, the simulations were performed setting the same operative temperature and, then, analysing its distribution along the heating and cooling seasons. Both the systems are able to maintain the desired thermal comfort conditions, but the operative temperature is inside the set band for a higher amount of time with the all-air system than with the radiant system.

The energy performance of the two systems was compared considering also the consumption of an air-to-water heat pump and the auxiliaries. When the building envelope is designed to control heat losses and solar gains, the all-air system has proved to be an interesting solution in the considered climates not only in cooling conditions but especially in heating operation. Despite the higher water temperature needed, during the heating season the total electric energy consumption of the all-air system is always lower than that of the radiant floor system (-19% in Helsinki, -32% in Milan and -74% in Rome). During the cooling season, the total electric energy consumption of the two systems is almost the same in Helsinki, while in Milan and Rome the all-air system consumes respectively 14% and 29% less than radiant floor with controlled mechanical ventilation; this is due to the consumption of the dehumidifier coupled with the radiant floor system. Air renewal with free cooling ensures a significant energy saving in all the locations. Considering the yearly electric consumption for heating, cooling and ventilation, the energy saving ensured by the all-air system compared to the radiant floor system with controlled mechanical ventilation is 12% in Helsinki, 14% in Milan and 23% in Rome. In conclusion, the results outline that the analysed all-air system is suitable and convenient for all the investigated locations.

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NOMENCLATURE

E_{el}	electric energy [kWh]
GHI	annual global horizontal insolation [kWh/m^2]
h	entalpy [J/kg]
$\dot{m}$	mass flow rate [kg/h]
Q_{th}	thermal energy [kWh]
T	temperature [°C]
W	absolute humidity [kg$_v$/kg$_{da}$]
ε	efficiency of the heat recovery unit

Subscripts

a	air
aux	auxiliary
dehum	dehumidifier
ex	extraction
ext	external
hp	heat pump
hr	heat recovery
lat	latent
min	minimum
max	maximum
mr	mean radiant
op	operative
rec	recirculation
ren	renewal
tot	total
w	water

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