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Absorption and compression heat pump systems for space heating and DHW in European buildings: Energy, environmental and economic analysis

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ABSTRACT

The selection of the proper device for space heating and domestic hot water for a building is crucial to achieve good energy and economic performances. For a single-family house, the most common heating device is the condensing boiler. Solar systems, electric heat pumps and gas driven sorption heat pumps represent suitable alternatives for improving the efficiency. Although the performances of each technology are well known, their ability to operate efficiently in bivalent heating plants depends on several variables and the choice of the most suitable heating system for a specific building is not straight-forward. The aim of this paper is to compare, under conditions typical of the European region, the seasonal performances of six system configurations that are obtained by combining the most commonly used heating technologies. The comparison is carried out in terms of primary energy consumption for three climatic conditions, changing the quality of the building envelope and the emission system typology. Although the results are sensitive to the primary energy factor for electricity, electric heat pumps generally result the most promising technology for conditions with low thermal lift, while gas heat pumps have the higher performances at high lift. Additionally, the systems are compared in terms of yearly CO_2 emissions and economic feasibility, finding scattered results among countries, due to large differences in the local energy mix and energy prices.

1. Introduction

It is estimated that in 2013 European buildings used 644 Mtoe of final energy, corresponding to about 41% of the overall consumption in EU28 [1]. Of this share, two thirds were used in households [2], where about 80% of the energy was dedicated to Space Heating (SH) and Domestic Hot Water (DHW), while cooking, lighting, electrical appliances and cooling together accounted for the remaining 20% [3]. Therefore, to be effective, any strategy aimed at significantly reduce the energy consumption and the related emissions in Europe must include space heating and DHW production in residential buildings. On this path, the EU has introduced several measures to ensure the progressive reduction of energy consumption in buildings [4–6].

Actions aiming at the reduction of the energy need for space heating can be focused on the overall performances of the building or on the efficiency of system components [7]. These measures will assure relevant results only in the long-term, since in Europe the new buildings share is about 1% of the actual stock every year and the renovation rate of existing buildings ranges between 0.5% and 1.2%, according to the country [8]. While building renovation takes place slowly, the heating system renovation may represent an option to fasten the reduction of the energy consumption in the residential sector, thanks to its higher renovation rate, estimated at about 3.6% [9]. Moreover, the renovation of the system may be less costly and less impacting on occupied buildings, especially if the renovation action does not imply the replacement of the existing emission system.

Since gas boilers are capable to provide SH and DHW without a storage and auxiliaries, they usually represent the cheapest solution in terms of first costs, given the low price of the appliance itself and the simple system required. Currently, in Europe a relevant share of heating systems is based on gas boilers. In the last decades, the need of reducing the energy consumption for space heating, driven by environmental and economic issues, promoted the improvement of the boilers efficiency, with the introduction of modulating boilers and condensing boilers. In

Abbreviations: AEH, Auxiliary Electric Heater; AEF, Auxiliary Energy Factor; COP, Coefficient Of Performance (heating mode); CR, Capacity Ratio; CFR, Compressor Frequency Ratio; CB, gas Condensing Boiler; DHW, Domestic How Water; EHP, Electric Heat Pump; EU, European Union; GCV, Gross Calorific Value; GHP, Gas absorption Heat Pump; GUE, Gas Utilization Efficiency (heating mode); HS, Hydraulic Separator; IAM, Incidence Angle Modifier; LF, Load Factor; MCHP, Micro Combined Heat and Power; N, New building; NZEB, Net Zero Energy Building; O, Old building; PEF, Primary Energy Factor; PER, Primary Energy Ratio; R, Refurbished building; SFH, Single Family House; SH, Space Heating; SS, Solar thermal System; y, year

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Nomenclature

I_T	global radiation Incident on the solar collector, W/m^2
Q_{DHW}	domestic hot water heat demand, kWh or kWh/m^2
$\dot{Q}_h$	heating capacity, kW
Q_{gas}	gas input energy referred to the Gross Calorific Value, kWh or kWh/m^2
$\dot{Q}_{gas}$	gas input power referred to the Gross Calorific Value, kW
Q_{SH}	space heating energy, kWh or kWh/m^2
T	temperature, °C
W_{in}	electric energy input, kWh or kWh/m^2
$\dot{W}_{in}$	electric power input, kW

Greek symbols

η	thermal efficiency

Subscripts

air	outdoor air
aux	auxiliaries' electric devices
avg	average
ext	external ambient conditions
gen	generator
h	heating mode
in	inlet water
max	maximum
out	outlet water
min	minimum
nom	nominal conditions
th	thermal

2015 condensing boilers accounted for the 78% of the European gas boiler market [9], with a growing trend, fostered by the Ecodesign Directive [7], come into force in September 2015, which banned from the market the less efficient heating devices.

The need for further reduction of the energy consumption for space heating and DHW promoted the diffusion of more efficient technologies, as heat pumps, or combinations of two or more heating devices. These options still represent a small fraction of the installed appliances, but they experience a growing trend.

The vapour compression heat pumps market reached in 2015 a size of 900,000 units, of which about one third is made of air-to-water heat pumps for space heating purposes [10]. Heat pumps may have lower primary energy consumption on seasonal basis, especially if ground sourced and coupled with an underfloor heating system [11], but they require a more cost intensive and complex system. Additionally, their efficiency is strongly influenced by the sizing of the appliance, the heating system features and the adopted control strategy.

Hybrid systems, i.e. a combination of a gas fired unit and a vapour compression heat pump, are considered a possible solution to cope with the drawbacks of heat pumps [12–14]. The market share of these systems is currently low, although it is expected to gain relevance after 2020 [9].

Gas driven absorption heat pumps (GHP) represent a further alternative to gas boilers, without some of the drawbacks of vapour compression units. In particular, GHP are capable to operate with high temperature emitters, as radiators, with good efficiency [15]. Moreover, for countries with a capillary gas distribution infrastructure, they offer the possibility to exploit renewable energy in the heating sector without imposing a load shift from gas to the electricity and, thus, affecting the operation of the electricity grid.

For what concerns solar systems, they can be effectively used for the DHW production. However, when applied for SH, their output is in counter phase with building demand, especially in cold climates. Thus, it is possible to cover a significant share of the heating needs only oversizing the system, with the related additional costs.

The high number of variables affecting the system performances makes the choice of the best technology strongly dependent on the characteristics of the application, such as the climatic conditions, the type of emission system, the space heating and DHW load profiles.

Several studies about the development, optimization and integration of efficient heating devices are available in the literature [16], often addressing buildings with very low heating demand or NZEB [17–19]. Most of these works deal with electric heat pumps [20], often ground sourced to improve the seasonal efficiency [21,22]. Advanced system configurations are also investigated, with heat pumps coupled with other technologies, as internal combustion engines [23], solar thermal or photovoltaic-thermal systems [24].

However, only few works provide an exhaustive comparison among heating systems based on different technologies. In [25] gas condensing boiler, wood pellet boiler, micro-combined heat and power (MCHP), air-to-water electric heat pump, air-to-water gas absorption heat pump and exhaust air-to-air electric heat pump are compared on primary energy consumption for heating low energy houses. The comparison is made for various climatic conditions and considering for each appliance the size available on the market.

Purpose of this work is to provide a wide and in-depth view of the performance of different heating systems in Europe, addressing both space heating and DHW production. The analysis focuses on the residential market and on single-family houses, which represent 66% of the residential buildings in EU28, in terms of floor area [1]. To provide results of general applicability, the comparison is performed for several cases, changing two of the variables, which mostly can affect the performances of heating systems, i.e. the climatic condition and the quality of the building envelope. Care is given to simulate the part load behaviour of the appliances, which

As for the selection of the heating systems the criteria of low cost, simple installation, low maintenance and large potential market have been followed. The six resulting heating system layouts are the following:

- Condensing boiler (CB);
- Condensing boiler with solar system for DHW production (CB + SS);
- Electric heat pump with electric back-up (EHP + AEH);
- Hybrid system with electric heat pump and gas back-up (EHP + CB);
- Gas absorption heat pump monovalent (GHP);
- Gas absorption heat pump with gas fired back-up (GHP + CB).

With the purpose of providing relevant and exhaustive results, the comparison has been carried out based on three indicators: primary energy, CO_2 emissions and economics.

By means of numerical simulations, the seasonal performances of the six alternative systems are compared under nine different conditions, obtained by the combination of three climatic conditions and three building standards. Unlike in [25], the heating devices have been sized according to the building requirement, assuming that appliances of different capacity will be available on the market as soon as each technology increase its market share. Moreover, this choice allows the scalability of the results to larger or smaller buildings than the one used for the presented calculations.

Since the different technologies have a different level of maturity, a direct comparison of their life cycle costs is not meaningful. Therefore, the seasonal operating costs are calculated for different countries and are used to provide an indication of the affordable investment cost, defined as the investment cost of each technology that would ensure economic feasibility with respect to the reference technology.

2. Methodological approach

The comparison among the different heating system has been performed for nine cases defined as the combination of three climatic conditions, identified by the main European standards on this topic [26,27] (average, cold and warm), and three building standards, namely “old”, “refurbished” and “new” buildings representative for each climate. The nine resulting buildings have been modelled in TRNSYS 16, using Type 56 [28].

The heat demand for space heating has been calculated through six minutes time-step simulations and coupled with the DHW demand to generate the heating demand for the system model. The choice of performing separate simulations for building and system has been considered the most suitable for the type of investigation carried out within this work. On the one hand, this approach has the drawback of neglecting the interaction between system and envelope. This implies that no indoor temperature fluctuations due to the hysteresis of the control system are allowed. Moreover, the response of the emission system at the different boundary conditions (ambient temperature, supply temperature, mass flow rate) is not considered. On the other hand, the simulation of a specific control and emission systems would have prevented a more dedicated focus on the heating technology, as it would have introduced additional variables, like the control strategy and the emission system features. Moreover, separate simulations of space heat demand and plant operation make the calculation simpler and reduces significantly convergence issues.

With the purpose of obtaining realistic results, the system has been sized using the same method usually applied by system designers. The maximum steady-state heat demand has been calculated, by means of the “Maximum heat load calculation” method already implemented in the tool, which sets the ambient temperature at the minimum of the given location and switches off the internal and solar gains.

3. Building modelling

The selection of the building features has been done considering two antithetic needs. On the one hand, in order to obtain relevant results, the buildings models have to be as more representative as possible of the typical buildings of each climatic condition. On the other hand, the degrees of freedom for the definition of each building has to be minimized, in order to have comparable results. Consequently, the nine cases share common building geometry, occupancy and internal loads, while they differ based on U-values and envelope permeability.

3.1. Climate data

Three different climate zones were selected, corresponding with the three different climates defined by the European ERP Directive. For the weather data, the following Meteonorm [29] weather files have been used:

- Helsinki (cold climate): FI-Helsinki-Kaisani-29980
- Strasbourg (average climate): FR-Strasbourg-71900
- Athens (warm climate): GR-Athinai-167140

3.2. Buildings description

The heating load profiles were defined for a single-family house (further referred to as SFH), based on the outcome of Task 44 “Solar and Heat Pump Systems” of the IEA Solar Heating and Cooling program [30]. The building consists of two levels, with a floor area of 70 m^2 each. The simulation is carried out considering the buildings as a single thermal zone.

Table 1
Building selected from the TABULA database as reference for the modelled buildings.

Climate	Building condition	Building ID
average	Existing state	DE.N.SFH.05. Gen – 1958–68
	Usual refurbishment	DE.N.SFH.05. Gen – 1958–68
	Improved standard	DE.N.SFH.12. Gen – after 2016
warm	Existing state	GR.ZoneB.SFH.02. Gen – before 1980
	Usual refurbishment	GR.ZoneB.SFH.02. Gen – before 1980
	Ambitious standard	GR.ZoneB.SFH.04. Gen – after 2011
cold	Existing state	NO.N.SFH.02. Gen – 1956–1970
	Usual refurbishment	NO.N.SFH.02. Gen – 1956–1970
	Improved standard	NO.N.SFH.07. Gen – after 2011

The features of the envelope have been selected coherently with the building location. For each climate, the envelope of the old, refurbished and new buildings have been defined based on the results of the European projects TABULA and EPISCOPE [31], which provided details about the typical buildings of different European regions, differentiated by period of construction, the renovation measures usually implemented and the envelope features for each location and each construction period.

The selected buildings are reported in Table 1, identified according to the nomenclature used in the TABULA database. In Table 2, the resulting U-values for the different surfaces and for each building are summarized. Additionally, it is reported the U-value variation due to the thermal bridges and the infiltration rate. The required ventilation rate is obtained by the difference between the air change for hygienic purposes, set to 0.4 h^{-1} for all the building, and the infiltration rate. When the air change is provided by a mechanical ventilation system, a heat recovery system with an efficiency of 60% is considered.

The reference buildings have been selected according to the following criteria:

- The year of construction of the old building is antecedent the implementation of energy efficiency regulation. Moreover, among different possible periods, the one with the higher number of construction was chosen, in order to use a building typology that is representative for the given location. For the warm climate, the database contains a single typology for all buildings before 1980, which has been selected as representative of the old building.
- The refurbished building corresponds to the same building typology of the old building, but refurbished. In the TABULA project, two possible renovation measures, called “usual refurbishment” and “advanced refurbishment”, are reported for each existing building. Within this work, the usual refurbishment, which is more likely to be implemented with reasonable costs and payback time, has been selected.
- For the new building, three standards are available: national standard, improved standard and ambitious standard. For what this work concerns, the improved standard has been considered for the average and the cold climate. The ambitious standard has been chosen for the warm climate because the features of the improved standard would have been very similar to the refurbished buildings.

3.3. Internal gains

Internal gains due to the presence of inhabitants and to the use of equipment and lighting are considered in the calculations. One person is associated with 20 W of convective and 40 W of radiative gains. The latent heat, usually about 40 W, is not considered in the simulation, as humidity is not controlled by the system.

Both the profiles for the occupation and equipment and lighting are described by an hourly schedule, as reported in [30] and assumed identical for each day. The corresponding yearly energy amounts to 3.0 kWh/m^2/y of convective and 6.0 kWh/m^2/y of radiative gains for

Table 2
Main building features for the nine cases.

		Average climate			Cold climate			Warm climate		
		Old	Refurb.	New	Old	Refurb.	New	Old	Refurb.	New
U_{wall}	$W/(m^2 K)$	1.10	0.23	0.15	0.41	0.29	0.10	2.20	0.41	0.35
U_{roof}		0.80	0.41	0.13	0.36	0.21	0.08	3.70	0.40	0.30
U_{floor}		1.00	0.31	0.15	0.90	0.90	0.15	0.95	0.95	0.75
$U_{windows}$		2.80	1.30	1.10	2.80	1.90	0.80	4.70	3.00	1.82
U_{door}		3.00	1.30	1.30	3.00	1.30	1.30	3.00	1.30	1.30
$\Delta U_{t.\ brid.}$		0.10	0.10	0.05	0.10	0.05	0.02	0.15	0.10	0.05
Infiltr.	h^{-1}	0.40	0.20	0.10	0.40	0.20	0.05	0.40	0.10	0.05
Vent. System	–	No	Yes	Yes	No	Yes	Yes	No	No	No

occupation and 13.4 kWh/m^2/y for equipment.

3.4. Heating set points

The heating set point is 20 °C between 6:00 and 22:00 while it is lowered at 16 °C for the remaining hours. For each building, the heating season has been defined according to the building location (see Table 3).

3.5. Domestic hot water

For what concerns the DHW needs, the tapping profiles defined in the Commission Delegated Regulation (EU) No 812/2013 [32], concerning water heaters, have been considered. The cycles define a DHW demand over a period of 24 h, specifying for each tapping the typology, the beginning time and the amount of energy in the hot water, according to the typology. In the present work, coherently with the building size, the tapping cycle “L” has been used, corresponding to a household with four inhabitants. According to the Standard, the inlet water temperature has been set a 10 °C for all the buildings. Over the 24 h of the cycle, an amount of water equivalent to 200 l at 60 °C is drawn, corresponding to 11.7 kWh per day.

For what the system layout for the DHW production concerns, a storage tank of 80 l has been used, heated up by means of an internal coil. The set point temperature in the tank is 60 °C, with a death band of 5 °C.

4. System modelling

In this section, criteria for the system dimensioning are presented and it is described how the different heating devices are modelled.

4.1. System layout and sizing criteria

A general layout of the hydraulic schemes is reported in Fig. 1. The back-up, when present, is installed in series with the main heating device. The DHW is stored in a tank of 80 l, which is heated up deviating the flow rate leaving the heating device with a three-way valve, when the water temperature falls below 55 °C. The hydraulic separator (HS) has a volume of 50 l and the mass flow rate on the primary and secondary circuit are constant and set according to the maximum space heating demand of each building. A variable flow approach has not been adopted to avoid the introduction of arbitrary choices on the control, which could have differently affected the heating devices efficiency.

As anticipated in Section 2, at the design temperature the main heating device and the back-up should be able to provide a heating capacity corresponding to the maximum steady-state heat load of the building. When a back-up system is installed, the main heating device is able to meet the demand only above a certain external temperature, called bivalent temperature. Above the bivalent temperature, the heating device modulates because its heating capacity exceeds the building load; below the delivered heat has to be supplemented by the back-up system.

For the different system configurations, the following dimensioning criteria have been followed:

- CB, CB + SOL and GHP: the size of the monovalent appliances is chosen to provide the maximum building load at the design air and water outlet temperatures. As the solar system is used for DHW only, the CB operates as a monovalent appliance for space heating.
- GHP + CB: lacking more detailed studies about the proper set of the bivalent temperature of a GHP with a CB as back-up, a preliminary investigation on the optimal sizing has shown that, as a rule of thumb, the GHP design conditions shall cover about 50% of the maximum building load.
- EHP + AEH: as the EHP is always more efficient than an electric resistance, the EHP should run whenever possible. Thus, a bivalent temperature equal to the minimum operating temperature has been chosen. Below the minimum operating temperature, the load is fully covered by the electric heater.
- EHP + CB: the heat pump capacity is lower than the maximum building demand. The bivalent temperature can be either chosen based on economic considerations or by driven by the low COP at low ambient temperature. In the present work, the bivalent temperature for the warm and cold climate is set according to the limits proposed by the Standard UNI EN 14825 [27], i.e. 7 °C for warm climate and −7 °C for cold climate. In both the cases, the ambient temperature is lower than the bivalent temperature during about 9% of the hours in the heating season. For the average climate, a different approach has been used. In this case, the limit proposed is 2 °C and the number of hours in the heating season with a lower ambient temperature is about 25%, significantly higher than for the cold and warm climate. Thus, it has been decided to lower the bivalent temperature to −2 °C, in order to maintain a coherent approach with the warm and average climates. Moreover, this choice is also in agreement with the findings of Naldi et al. [33], who carried out a more detailed investigation on the selection of the bivalent temperature for EHP.

The resulting heating capacity for each system is reported in Table 4. When two numbers are displayed, the first refers to the main heating device, the second to the back-up. The reported capacities are related to the rating conditions defined by the Standards [26,34], i.e. the external air temperature of 7 °C and water temperature 40/45 °C

Table 3
Heating season limits for the different climates.

	Helsinki	Athens	Strasbourg
From	01-sep	01-nov	01-oct
To	31-may	30-apr	30-apr

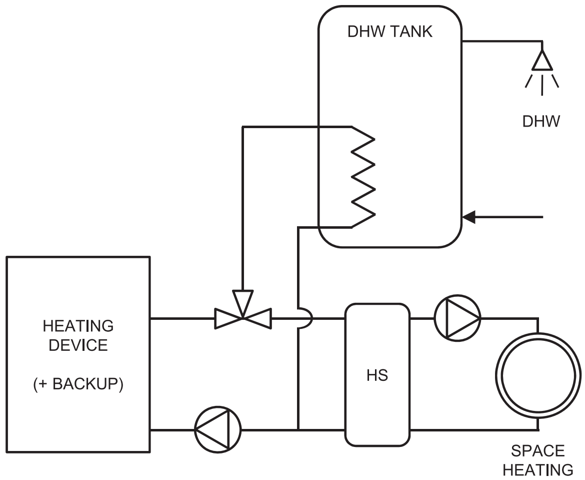

Fig. 1. Generic scheme of the systems layout.

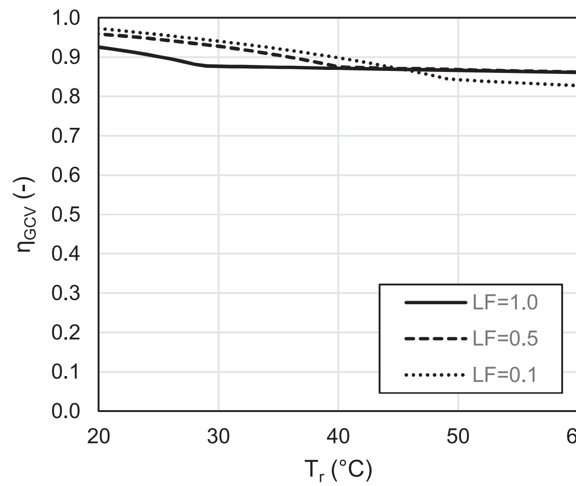

Fig. 2. Boiler thermal efficiency on the GCV against the return water temperature for three LF.

and 41.3/55 °C for the EHP and GHP respectively.

It has been decided to maintain the appliance size resulting from the calculation even when a corresponding product is not available on the market. This choice ensures the scalability of the results to buildings of different size. Moreover, it includes the possibility that systems with lower capacity will enter the market, especially for the most recent products (GHP).

4.2. Emission system and set point temperature

The emission system of the old building is based on radiators, thus, the water supply temperature at the design conditions is 65 °C. For what concerns the refurbished building, it is assumed that the emission system is not changed with the renovation. However, the supply temperature is set at 55 °C, under the hypothesis that the refurbishment of the envelope, reducing the heating demand, allows lower radiators temperature. For the new building, it is assumed that an underfloor heating system is used, with a supply temperature of 35 °C.

A climatic curve is used to modulate the supply temperature according to the external air temperature. In particular, the climatic curves have been shaped based on the test conditions of the Standards for heat pumps seasonal performance assessment [26,27], dependent on the climate and on the nominal supply temperature. The supply temperature is set at the highest value at the design conditions and reduced linearly as the outdoor air temperature increases.

4.3. Condensing boiler model

The gas condensing boiler model is based on the European Standard CEN EN 15316-4-1 [35]. Among the three calculation methods proposed in the Standard, the "boiler cycling method" has been selected as it distinguishes more explicitly the losses due to cycling.

The model includes the calculations for chimney and envelope heat losses with burner on and off as well as the auxiliary electric energy consumption. The impact of these effects on the boiler efficiency depends mainly on: type of heat generator and its location; load factor; operating conditions (e.g. return water temperature); control strategy (on/off, multistage, modulating, cascading).

Within this work, for all the simulations a high efficiency modulating gas condensing boiler has been considered, using the model parameters suggested by the standard. The resulting efficiency is mainly influenced by the return water temperature and the load factor (LF). The load factor is defined as the ratio between the actual delivered heat and the nominal capacity of the boiler. Fig. 2 shows the resulting thermal efficiency (η_{GCV}) against the return water temperature, calculated for three different load factors.

The shape of the curves can be justified considering that a lower

Table 4
Capacity of main and back-up heating devices.

		Old		Ren.		New	
		Q_{main_nom}	Q_{backup}	Q_{main_nom}	Q_{backup}	Q_{main_nom}	Q_{backup}
Cold climate	CB	13.0	–	10.0	–	6.0	–
	CB + SOL	13.0	–	10.0	–	6.0	–
	EHP + E	19.3	12.2	17.8	9.9	9.5	6.0
	EHP + CB	10.2	13.0	7.9	10.0	3.9	6.0
	GHP	17.2	–	13.5	–	6.6	–
	GHP + CB	8.7	7.0	6.7	5.0	3.3	3.0
Average climate	CB	14.3	–	6.3	–	4.5	–
	CB + SOL	14.3	–	6.3	–	4.5	–
	EHP + E	15.4	4.2	9.4	0.0	6.3	0.0
	EHP +CB	14.1	5.0	5.8	3.0	3.7	2.0
	GHP	18.6	–	7.7	–	4.9	–
	GHP + CB	9.4	8.0	4.0	4.0	2.5	3.0
Warm climate	CB	20.0	–	6.0	–	5.0	–
	CB + SOL	20.0	–	6.0	–	5.0	–
	EHP + E	23.2	6.0	7.9	0.0	5.3	0.0
	EHP + CB	12.3	11.0	3.8	4.0	2.7	3.0
	GHP	22.3	–	6.6	–	4.5	–
	GHP + CB	11.2	10.0	3.3	3.0	2.2	3.0

return water temperature allows higher condensation in the flue gases. Additionally, since the model considers that the efficiency of the flue gases heat exchanger decreases linearly with the gas mass flow rate, a lower load factor gives higher heat recovery efficiency.

4.4. Electric heat pump model

The electric heat pump model is based on the data included in a performance map provided by a manufacturer [36], reporting the appliance capacity and COP with air temperature ranging from −20 °C to 40 °C and compressor frequency ratio (CFR) of 33%, 66% and 100%. In Fig. 3, the COP is plotted as function of the supply and outdoor air temperatures for CFR of 100% and 33%. Additionally, the ratio between the actual and the nominal capacity is also reported. In the charts, the cross marks indicate the position of the available data used to create the performance map.

The selected appliance is an air-to-water heat pump for residential application, with an inverter driven compressor and with a nominal heating capacity of 15 kW at air 7 °C and water 40/45 °C. It has been verified that the appliance is representative of the units available on the market by comparing its efficiency with the efficiency of appliances from the Eurovent database [37]. The results of this comparison are reported in Fig. 4 at six different working conditions, given by the combination of three outdoor air temperatures (−7 °C, 2 °C and 7 °C) and two water temperatures (30/35 °C and 50/55 °C). For each working condition the bar identifies the COP range of the Eurovent population, the blue square represents the average COP and the red circle is the COP of the selected appliance. The chart shows that the appliance efficiency is not far from the average, being very close at 2 °C, slightly lower at 7 °C and slightly higher at −7 °C. Since for each working conditions more than 50 units were available in the database, this comparison can be considered rather representative of the market.

Additionally, analyzing the models available in the Eurovent database, it has been observed that the COP at air 7 °C and water 40/45 °C is independent on the appliance capacity in the range between 4 and 16 kW. This allows the scalability of the appliance size based on the reference performance map.

4.5. Gas absorption heat pump model

The GHP model, which details can be found in [15], is based on experimental data carried out on an ammonia/water gas driven heat pump prototype designed for residential applications. The data have

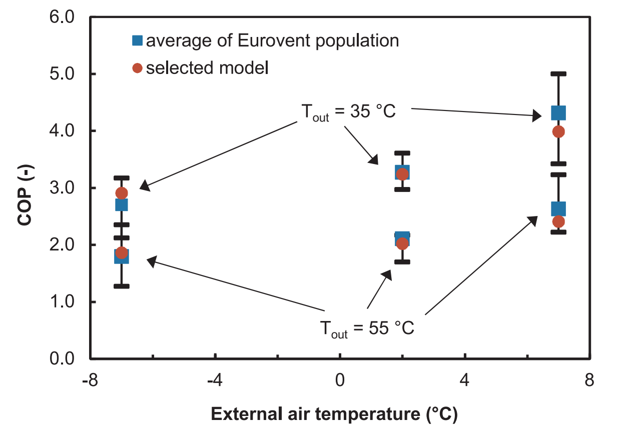

Fig. 4. COP at maximum compressor frequency of the appliances available in the Eurovent database compared with the COP of the selected appliance at two different supply water temperatures (T_{out}). (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article).

been collected according to the test procedure defined in the EN 12309-4 [22] and cover a wide range of operating conditions. In particular, the GHP performance have been measured over more than 50 working condition, obtained by the combination of various air temperature, outlet temperature and capacity ratio (CR), i.e. the ratio between the actual output and the maximum heating capacity at the same working conditions.

The model consists of a set of algebraic equations that allow calculating the Gas Utilization Efficiency (GUE) and the Auxiliary Energy Factor (AEF), defined as in Eqs. [1] and [2], based on the external air temperature, the return water temperature and the CR.

$$GUE = \frac{\dot{Q}_h}{\dot{Q}_{gas}} \tag{1}$$

$$AEF = \frac{\dot{Q}_h}{\dot{W}} \tag{2}$$

Once the GUE and AEF are known, the gas ($\dot{Q}_{gas}$) and electrical ($\dot{W}$) inputs can be calculated from the required heating capacity ($\dot{Q}_h$).

An overview of the resulting GHP efficiency is given in Fig. 5. The first and the second charts provide the GUE in the relevant ranges of supply water temperature (T_h) and external air temperature (T_{ext}) for two Gas Input Ratio (GIR), where the GIR is defined as the ratio

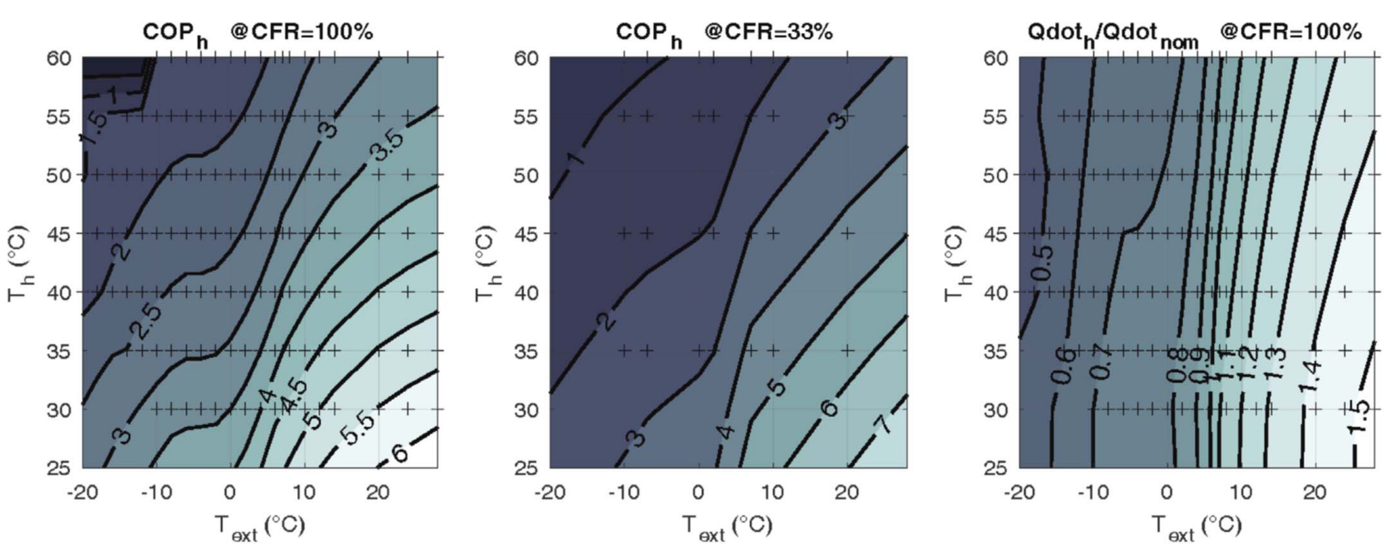

Fig. 3. COP at CFR = 100%, COP at CFR = 33%, and ratio between actual and nominal capacity of the EHP in the relevant ranges of external temperature (T_{ext}) and supply water temperature (T_h).

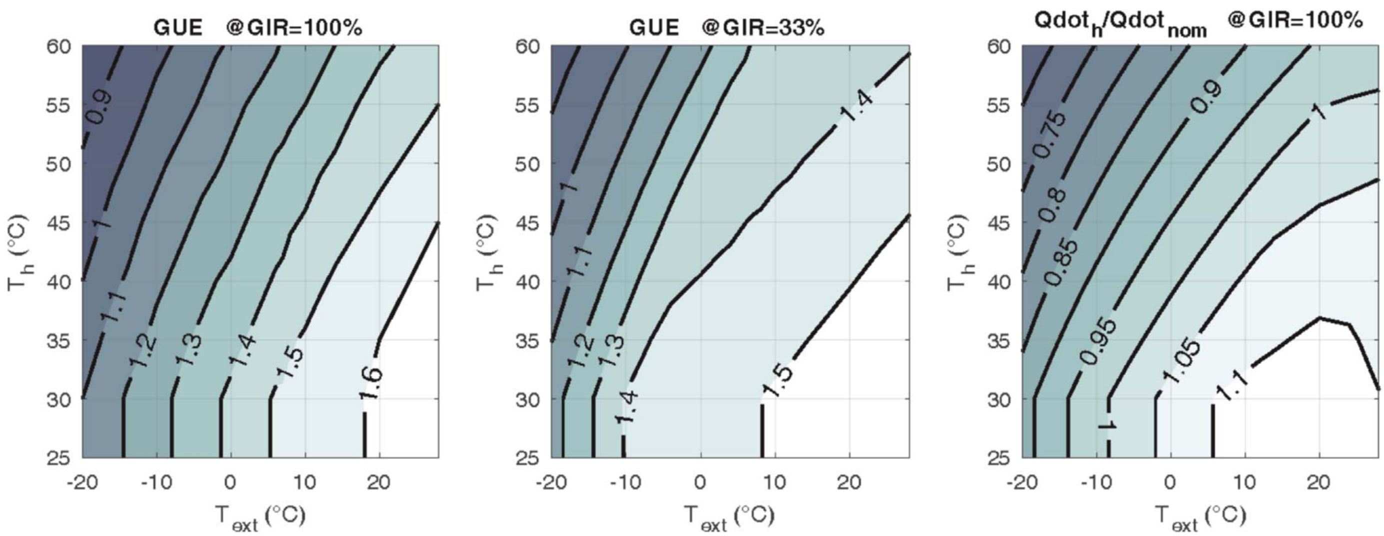

Fig. 5. GUE at GIR = 100%, COP at GIR = 33%, and ratio between actual and nominal capacity of the GHP in the relevant ranges of external temperature and supply water temperature.

between the actual and the maximum gas input. The third chart reports the ratio between the heating capacity and the nominal heating capacity at the maximum GIR.

4.6. Solar system model

The solar system is composed by a field of flat plate solar thermal panels, a circulation pump and a heat exchanger immersed in the DHW tank (Fig. 6).

The flat plate solar thermal panels thermal performance has been assessed using the Trnsys Type 1. This model assumes that the solar collectors efficiency versus a ratio of fluid temperature minus ambient temperature to radiation ($\Delta T/I_T$) can be modelled as a quadratic equation. The used efficiency equation parameters are: 0.8 for the intercept value, 3.6 W/m^2/K for the slope and 0.014 W/m^2/K^2 for the curvature. Lastly, about the effects of off-normal solar incidence there are 5 possibilities for considering it using the Trnsys Type 1. In this instance, a linear function has been used to compute the incidence angle modifier (IAM). The coefficient of the function used is 0.2 for the 1st order IAM.

The circulation pump guarantees a fixed flow rate. It is activated by a differential controller with hysteresis. The controller monitors the solar panel input and output temperatures and operates with a deadband between 2 and 10 °C, meaning that the pump is activated once the temperature difference is above 10 °C and it is kept running until it drops below 2 °C. Additionally, for safety reasons, the controller stops the pump when the temperature ion the DHW tank reaches 90 °C. For what concerns the energy balances, it is assumed that no pump power is converted to fluid thermal energy.

5. Results discussion

5.1. Heating need

The space heating demand is calculated with a time-step of six minutes to provide the load file to the system simulations. The yearly SH demand for the nine buildings are reported in Table 5. The DHW demand, equal for the nine cases, is 30.5 kWh/m^2/y, obtained repeating the daily tapping profile described in Sec. 3.6 along the year. The monthly details can be found in Figs. 7–9, for the warm, average and cold climate respectively. The vertical bars display the monthly SH and DHW demand profiles for the old (O), refurbished (R) and new (N) buildings, while the lines show the trend of the maximum, average and minimum monthly mean air temperatures.

5.2. Primary energy consumption

The performances of the different systems are summarized in Figs. 10–12 for the old, refurbished and new buildings respectively. In the charts, the light grey bars represent the gas yearly input and the dark grey bars are the required electrical input. The considered electrical input is limited to the auxiliaries of the heating device, while the consumption of circulation pumps or system controls has not been accounted in the comparison. In fact, since these contributions depend on the building and not on the heating device, they just represent a constant offset on the analysis. Thus, for condensing boiler the amount of electrical energy consumption is barely visible, due to the negligible auxiliaries, and for gas driven heat pumps it is rather small, being associated to the fan and solution pump operation only. The electric heat pump with electrical back-up does not require any gas input, while for the system with electrical heat pump and back-up boiler, inputs of both gas and electricity are required. The dots represent the Primary Energy Ratio (PER), calculated according to Eq. (3) with different values of the Primary Energy Factor (PEF). In fact, the calculation of the primary energy consumption for each case depend on the PEF used for the conversion of electrical energy into primary energy. The value to be adopted depends on the calculation method, on the time the electricity is used and on whether it is considered the average or the marginal

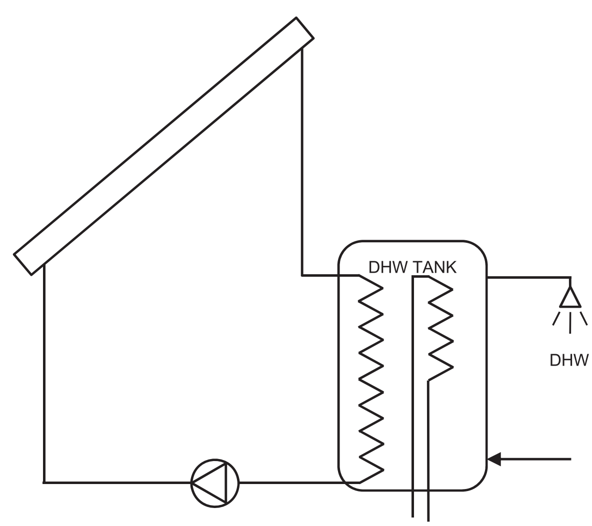

Fig. 6. Scheme of the solar system.

Table 5
Building heating needs for the various cases.

Climate	Building	SH needs (kW h/m²/y)
Warm	Old	208.1
	Refurbished	58.5
	New	42.5
Average	Old	230.6
	Refurbished	65.6
	New	25.0
Cold	Old	202.8
	Refurbished	146.1
	New	29.0

electricity generation. Since 2012 in the EU a conventional value of 2.5 has been used, as established by the Directive 2012/27/EU [5]. Recent studies [38] suggest that the reference PEF value for EU should be decreased at 2.0 to take into account the increasing contribution of renewable energy sources. This suggestion is already included in the latest proposal of the EC on energy efficiency [6]. For this work, the calculations have been performed for PEF values of 2.5, 2.0 and 1.8, i.e. the actual, the newly proposed and an even lower value, which may become the reference in a future scenario with higher renewable energy penetration. As expected, the variation of the PER with the PEF is almost negligible for the systems with condensing boiler or GHP, while it becomes significant when an electrical heat pump is used.

$$PER = \frac{Q_h}{Q_{gas} + W_{in} PEF} \tag{3}$$

Using the system based on a CB as the reference, the performances of the alternative systems can be commented:

- the primary energy savings given by the addition of a solar system to a condensing boiler depend on the climate and on the relative magnitude of space heating and DHW needs. The higher primary energy savings given by the solar system are found in the new buildings, where the DHW demand of about 28 kWh/m^2/y, represent about 40–50% of the energy need. High savings are also found for the refurbished building in the warm climate, where the share of energy need for DHW is about 30%, but the plant benefits of the high solar radiation. Lower savings are found in the refurbished building with average climate and even lower in the cold climate and in the old buildings, where the share of energy need for DHW is around 10%.
- The use of an electric heat pump with auxiliary electric heater is strongly dependent on the fraction of energy delivered by the electric back-up, which depends on the limit of the EHP to operate at high lift. Thus, in the cold climate there is a relevant amount of hours where the required heating capacity is provided by the AHE. Additionally, the PER of EHP decreases when the number of operating hours at high lift, i.e. with high supply temperature and low outdoor air temperature, is high, as in the cases of the old and refurbished building of the average climate. On the contrary, for systems which operate at low lift, as in the warm climate, high primary energy savings can be achieved, especially if low PEF are considered.
- Similar considerations can be done for the EHP with a CB as back-up system. The gas input to the CB decreases when moving from old building to refurbished and further to new: in the old building, the emission system (radiators) requires high water temperature, which can be achieved only with the contribution of the auxiliary system. With lower water temperatures, as with the underfloor heating system of the new building, the load for the back-up system is lower. Additionally, more insulated buildings have a space heating demand less dependent on the external air temperature, thus the difference between required power at bivalent temperature and the design temperature is smaller than for building with a less performing envelope.
- The performance in terms of PER of a monovalent GHP is rather constant with the building typology and climatic conditions: it increases slightly moving to a warmer climate, while almost no differences are found among old, refurbished and new buildings.
- The possible advantage of the use of a bivalent system with a GHP and a CB is to reduce the GHP operation at part load conditions. However, the results show that, with the sizing criteria applied within this work, the bivalent system is always less efficient than the monovalent. This implies that the disadvantage of using the CB for a certain number of hours exceeds the benefit of the higher load factor of the GHP. The difference between bivalent and monovalent systems increases in the new building and in the warmer climate where, unlike the CB, the monovalent GHP benefits from the low thermal lift.

Comparing the different systems, in the cold climate, the results suggest that the monovalent GHP is the system with the higher PER for all the buildings, regardless of the PEF. This result can be explained with the capability of GHP to maintain a high GUE even at high thermal lifts, both given by low ambient temperature and high water temperature. The gap between GHP and EHP increases in the new building, even if the underfloor heating system allows relatively low thermal lift for space heating, because the average lift along the year is increased by the high share of heating energy delivered for the DHW production. With a PEF for electrical energy of 2.0 or below the systems based on an EHP have a PER higher than the condensing boiler, except for the system with electrical back-up in the cold climate.

Considering the average climate, the monovalent GHP remains the solution with the highest PER in both the old and refurbished building, while both the systems with EHP perform better in the new building. This is explained considering that EHP have a steeper relation between thermal lift and performances. Thus, even if their efficiency is lower at high lift, it becomes higher when climatic conditions or supply temperature allow low lifts.

Therefore, the PER of EHP-based systems results higher also for the warm climate, except for the new building in the warm climate, where the fraction of heating power at high temperature required for the DHW

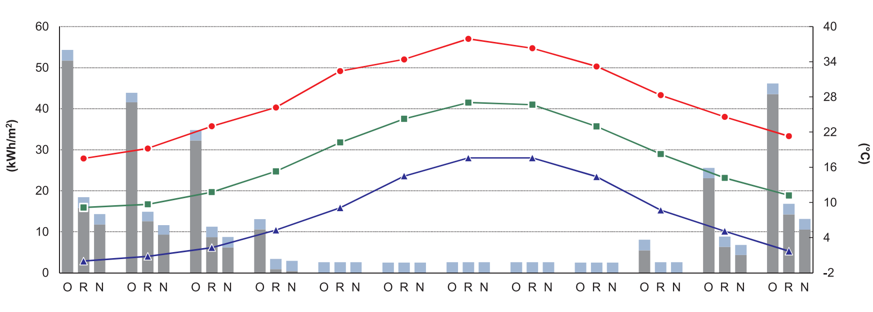

Fig. 7. Monthly SH and DHW demand and external mean air temperatures for the warm climate.

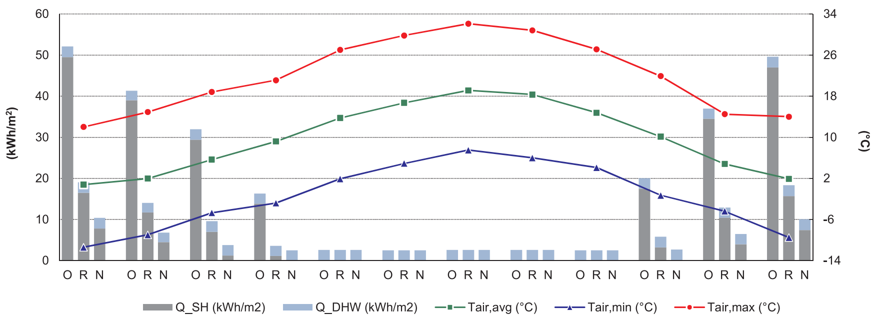

Fig. 8. Monthly SH and DHW demand and external mean air temperatures for the average climate.

production is high. In this case the PER of the monovalent GHP and the EHP + CB are very similar, with the value of PEF for electricity above 2.0.

A dependence of the performance by the fraction of delivered energy dedicated to the DHW production is found also on the impact of the solar system on the system performances. Thus, in the old buildings, where the demand for the DHW is about 10% of the overall heating demand, the increase of the PER is small. In the new buildings, where the DHW can reach 50% of the heating demand, the improvement of the PER much more evident. Besides the building typology, the climatic conditions affect the performance of the solar system, with a higher contribution in the warm climate than in the average and cold.

Summarizing, the calculations show that alternative systems to the condensing boiler can provide higher efficiency in terms of primary energy. Additionally, the results suggest that climate and building typology drive the selection of the system configuration. In particular, gas heat pumps result the most suitable technology for retrofitting the heating system of existing buildings when the original high-temperature emission system is maintained. Additionally, they can also be preferred to EHP for new buildings in the cold climate. Electric heat pumps best fit the warm climate and the new buildings in the average climate.

5.3. CO_2 emissions

Besides primary energy, CO_2 emissions represent another useful indicator for policy makers when comparing different heating technologies. The calculations have been performed at UE level, using a CO_2 emission intensity for electricity generation of 275.9 g_{CO2}/kWh, as reported by the European Environmental Agency [39]. For what concerns the natural gas, a value of 205 g_{CO2}/kWh has been used.

The results reported in Table 6 for the different climatic conditions and building typology, show that all the alternative systems guarantee CO_2 savings compared to the condensing boiler. The savings given by the solar system depend only on the climate, as solar energy is used only for DHW production, which is independent on the building typology. On the contrary, EHP and GHP guarantee savings that are dependent on the heating demand, higher in the old buildings and lower in the new ones. Comparing the alternative configurations investigated for the EHP and the GHP, in both cases the solution with a condensing boiler as back-up system gives the least CO_2 savings. In the case of the EHP, this can be explained with the different sizing criteria applied, as discussed in Section 4.1, which reduce significantly the operation of the electric back-up compared to the condensing boiler back-up. For what concerns the cases with the GHP, the CO_2 savings depend only on the seasonal efficiency of the system that, as discussed in Section 5.2, is always lower for the monovalent appliance.

5.4. Cost targets

Prime cost is usually the main factor hindering the diffusion of more efficient heating technologies. Thus, an economic analysis is required to complete the picture provided by this comparison. However, a direct comparison based on actual market prices would not provide meaningful results, because of the differences among the technologies in terms of number of manufacturers and available models and because of the different cumulative volume of production and, consequently, position on the experience curve. Thus, using the condensing boiler as reference technology, the acceptable cost difference has been calculated as economic indicator, i.e. the additional cost that can be accepted for a given technology compared to a condensing boiler. The acceptable cost difference is thus obtained as the difference of the yearly operative costs compared to the costs of the condensing boiler, times the number of year for the payback. The acceptable cost difference can be either a positive or a negative value. When it is positive, i.e. when the running cost is lower than for a CB, the considered technology results more convenient over a five-year period if its additional cost is lower than the calculated acceptable cost difference. Negative values can be found when the operating cost are higher than for a condensing boiler. In this case, the technology should cost less than a condensing boiler to be more convenient

The calculation has been performed for some representative European Countries, using the natural gas and electricity prices provided by Eurostat [40,41], which include taxes and levies, and reported in Table 7. Unlike for primary energy and CO_2 emissions, for what costs are concerned, it makes less sense to carry out an analysis at European level, since the economic and boundary conditions of the space heating and DHW markets vary significantly from country to country.

The results of the economic analysis can be found in Table 8, where the considered climatic condition is reported within brackets for each

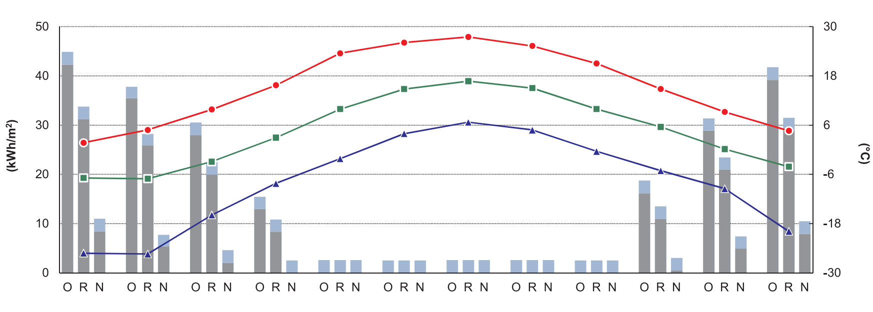

Fig. 9. Monthly SH and DHW demand and external mean air temperatures for the cold climate.

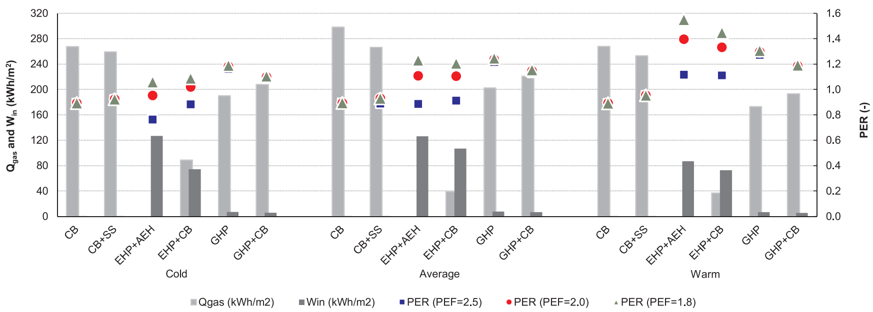

Fig. 10. Yearly gas and electricity demand and PER for the old building in the three climates.

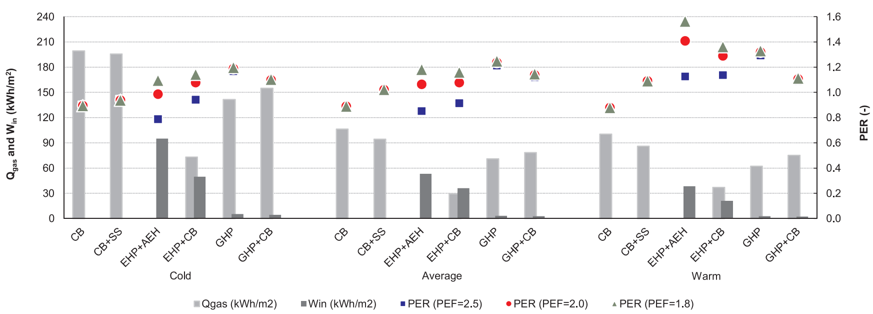

Fig. 11. Yearly gas and electricity demand and PER for the refurbished building in the three climates.

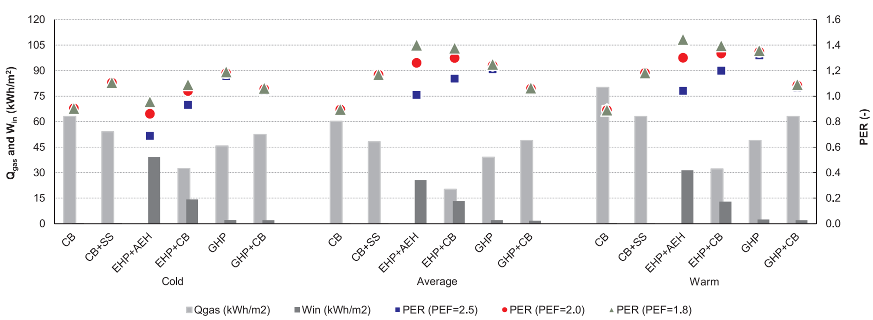

Fig. 12. Yearly gas and electricity demand and PER for the new building in the three climates.

Table 6
Yearly CO_2 emission savings (kg_{CO2}/y) and relative variation compared to condensing boiler.

		Cold		Average		Warm	
Old	CB + SS	263	(−3%)	327	(−4%)	501	(−7%)
	EHP + AEH	2 844	(−37%)	3 914	(−46%)	4 391	(−57%)
	EHP + CB	2 335	(−30%)	3 550	(−41%)	3 892	(−50%)
	GHP	2 010	(−26%)	2 511	(−29%)	2 525	(−33%)
	GHP + CB	1 549	(−20%)	2 011	(−23%)	2 003	(−26%)
Renovated	CB + SS	271	(−5%)	397	(−13%)	564	(−19%)
	EHP + AEH	2 234	(−39%)	1 337	(−44%)	1 678	(−58%)
	EHP + CB	1 868	(−32%)	1 120	(−37%)	1 264	(−44%)
	GHP	1 517	(−26%)	917	(−30%)	1 016	(−35%)
	GHP + CB	1 142	(−20%)	717	(−23%)	635	(−22%)
New	CB + SS	334	(−18%)	408	(−23%)	571	(−25%)
	EHP + AEH	530	(−29%)	907	(−52%)	1 244	(−54%)
	EHP + CB	475	(−26%)	763	(−44%)	990	(−43%)
	GHP	460	(−25%)	518	(−30%)	822	(−36%)
	GHP + CB	290	(−16%)	294	(−17%)	443	(−19%)

Table 7
Electricity and natural gas prices in € and ration between the two prices.

	Sweden	France	Germany	Italy	Netherlands	UK	Greece
Electricity price	0.19	0.17	0.30	0.24	0.18	0.19	0.18
Gas price	0.12	0.07	0.07	0.08	0.08	0.06	0.07
Ratio	1.7	2.3	4.4	2.8	2.3	3.1	2.4

country. To improve readability, the results have been rounded to the nearest hundredth.

The acceptable additional cost is strongly dependent on the heating demand of the building, thus it is usually higher in the old buildings and lower in the new ones. Additionally, it depends on the gas and electricity prices and on the ratio between the two prices. Looking at two extreme situations, in Sweden, with high gas price and low ratio between electricity and gas prices the acceptable additional cost is high for all the technologies, especially for EHP. On the contrary, in Germany the price of electricity is high, while the ratio between electricity and the gas prices are rather low. Thus, EHP are not economically convenient and also the technologies using gas as main energy source have a lower acceptable additional cost than in other countries.

6. Conclusions

In this paper, the seasonal energy efficiency of six different heating systems for space heating and DHW production in a single-family house has been compared numerically in terms of primary energy. To include the main variables affecting the results, the analysis has been carried out for three different representative European climates and for three different building standards for each climate.

The results are coherent with the features of the different technologies. The benefit given by the solar systems are higher in warm climates and in new buildings, where the energy need for DHW is comparable with the need for space heating. The thermal lift impacts on the performances of the heat pumps, but more significantly on vapour compression than on absorption appliances. Consequently, if GHP will

Table 8
Additional acceptable cost compared to condensing boiler for a simple pay-back time of 5 years for different European countries (in €).

		Sweden (cold)	France (average)	Germany (average)	Italy (average)	Netherlands (average)	UK (average)	Greece (warm)
Old	CB + SS	700	600	500	700	600	500	900
	EHP + AEH	5 000	1 000	−10 700	−2 500	1 300	−3 300	3 300
	EHP + CB	4 800	1 200	−8 700	−1 600	1 600	−2 400	3 100
	GHP	5 500	4 000	3 200	4 600	4 500	3 200	4 200
	GHP + CB	4 300	3 200	2 500	3 600	3 600	2 600	3 300
Refurbished	CB + SS	800	700	700	800	800	600	1 000
	EHP + AEH	4 100	200	−4 100	−1 100	300	−1 400	1 300
	EHP + CB	4 100	400	−2 600	−400	600	−700	1 300
	GHP	4 200	1 500	1 100	1700	1 600	1 200	1 700
	GHP + CB	3 100	1 000	700	1 100	1 100	800	1 000
New	CB + SS	900	700	700	800	800	600	1 000
	EHP + AEH	700	500	−1 500	0	700	−300	800
	EHP + CB	1 000	800	−400	600	900	300	1 100
	GHP	1 300	800	600	900	900	600	1 400
	GHP + CB	800	400	300	500	500	300	700

confirm the performances observed for the first prototypes, it would result a promising option for high-lift applications, i.e. in the cold climate and for existing buildings, with radiator-based emission system. On the contrary, in new buildings with underfloor heating and in the warm climate, the EHP results the option with the lowest primary energy consumption, especially if low PER are considered.

The distinction of the most suitable heating device for each building becomes less obvious if the comparison is made in terms of CO_2 emissions or costs.

From a comparison at EU level based on CO_2 savings, all the considered technologies provide an emission reduction compared to the condensing boiler. In particular, systems with an EHP result the less impacting, especially if an auxiliary electric heater is used.

A comparison on economic basis has been carried out evaluating the maximum acceptable additional cost compared to the cost of a condensing boiler. The prices of natural gas and electricity influenced significantly the results. In particular, in old buildings, in countries with high electricity prices as Germany, Italy or UK, the EHP have to be less expensive than condensing boilers to be economically convenient. In Germany this is the case for refurbished and new buildings too. On the contrary, in countries with higher gas prices or lower electricity prices, the additional prices for EHP and GHP are comparable. Moreover, additional costs up to some thousands Euros can be accepted in old buildings and in some refurbished buildings, where the high energy needs the more efficient technologies to provide high savings. On the contrary, in the most performing buildings, because of the smaller economic savings the price of an alternative technology should not exceed about one thousands Euros or less the price of a condensing boiler to be competitive.

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