Energy and economic analysis of different heat pump systems for space heating

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Abstract

The consumption of natural gas as a primary energy source in Italy has increased during recent years, mainly due to more widespread use of modern natural gas-fired combined cycle power plants. It is generally accepted that such an increased use of natural gas is beneficial, particularly in summer, due to the 'take-or-pay' contracts that often regulate energy supply. Conversely, the use of electrical energy should be decreased, in order to limit the 'peak demand' problem that has become prevalent in Italy. Therefore, besides electrically driven heat pumps (EHPs) that achieve good efficiencies, it is interesting to also consider the option of combustion engine-driven GEHPs for space heating purposes. In the latter type of HPs, losses attributed to the production and transport of electricity are eliminated and, in addition, there is the possibility to re-use the heat from the combustion engine. This article presents an assessment of the annual economic and energy profiles of electric and internal combustion engine HPs for space heating purposes. Due to the dependency of the performance of such technology on the source and sink (heating circuit) temperature levels, a comparison is performed of air-to-water HP systems (the most widely used) in two cases of maximum flow temperatures. The calculations show that natural gas-driven HPs can achieve approximately the same efficiency as electrically driven HPs that are powered with electricity from modern natural gas-fired combined cycle power plants. Within this study, the efficiencies of such systems are also compared with those that utilize conventional boiler technologies.

Keywords: electrical heat pump; gas engine heat pump; space heating; energy efficiency

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1 INTRODUCTION

The energy consumed in residential and tertiary sectors for space heating and domestic hot water production comprises ~20% of the total energy demand in Italy. The publication of the 93/76/EEC Directive (relating to the energy analysis of buildings and energy performance certification), the 2002/91/EU and the newest 2010/31/EU Directive on the energy performance of buildings (recast) has resulted in a greater awareness of the need to explore potential ways to make large energy savings. In Italy, a number of legislative acts have been issued during recent years, but heat pumps (HPs) have not been an attractive option for the market until now. In the past, limited uptake may have been due to unfavourable economic conditions with regard to the electrical tariff (only recently resolved by the Italian mains electricity distributor with the introduction of a dedicated tariff), and to the undeveloped technical knowledge of Heating, Ventilation, Air Conditioning (HVAC) designers and installers.

Presently, conditions are more favourable for a number of reasons given below.

Increased electrical energy production by distributed technologies (i.e. by feed-in tariff or net metering) could encourage the use and uptake of HPs, as they can be used in conjunction with the latter.

The energy performance of HPs will be improved by means of standardized test methodology (UNI EN 255-3, UNI EN 14511-2/3/4) [1–4].

Plant design tools and numerical procedures for energy analysis have now been defined for HPs (UNI EN 15316-4-2, BS EN14825) [5, 6].

It is interesting to also consider the use of combustion engine-driven HPs, as well as electrically driven HPs, for space heating purposes.

In Italy, the UNI/TS 11300-1-2-3-4 standards have been developed, which outline the calculation methods to be used

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for evaluating the energy performance of buildings. In particular, Part 4 [7], due to be published during 2012, concerns the replacement of conventional means of space heating and domestic hot water production (i.e. boilers) with the use of renewable and other technologies. It proposes the UNI EN 15316-4-2 calculation method for quantifying energy loads and efficiencies of electrical HP-based heating plants.

In order to extend the analysis, this article reports on the energy comparison of different air-water HPs (electrical—EHP, and gas engine HPs—GEHPs), as applied to different heating plants for the residential sector. It is only in this sector that heating loads definitely exceed cooling loads for the climates considered. The assessment is extended to consider CO_2 emissions, and the economic implications of the various technologies. It is undertaken for three different Italian climates (Milan, Florence and Rome), in order to provide information to HVAC designers regarding the energy and economic performance of EHP and GEHP plants, when compared with more traditional heating systems.

2 CHARACTERIZATION OF HPS

In Italy, all compact electrically driven HP systems approaching the market are tested by the manufacturers using the European standards UNI EN 255-3 and UNI EN 14511-2/3/4. This article is concerned with the analysis of scroll-type compressor HPs, which are operated at a given constant speed (on/off operation). Based on the performance data of electrically driven HPs, a model has been developed which describes the performance that would be expected when the compressor within the HP is directly driven by an internal combustion engine [8]. The assessment has been undertaken for three different Italian moderate climates (Milan, Florence and Rome, respectively, MI, FL and RM) by means of monthly mean dry bulb temperature and data relating to solar radiation on the horizontal plane [9].

The maximum coefficient of performance of an HP provides an appropriate platform from which to develop the model. It is dependent on both the heat sink temperature (water to heating plant, $T_{w,out}$) and the heat source temperature (external air, T_e):

$$\mathrm{COP}_{max} = \frac{T_{w,out} + 273.15}{T_{w,out} - T_e} \tag{1}$$

It is intended that Equation (1) is valid only in the case where there is an absence of intermediate thermal vectors between the heat tanks and the HP refrigerant, as assumed by prUNI/TS 11300-4. COP values of real systems are always lower than theoretical COP_{max}, a situation which is accounted for by consideration of the calculated exergy efficiency:

$$\eta_{ex} = \frac{\mathrm{COP}}{\mathrm{COP}_{max}} \tag{2}$$

HP performance is much more sensitive to operating conditions than other conventional heating systems. Therefore, in order to undertake energy analysis during the heating season, it is important that the HVAC designer has access to performance data for the HP at full load for different temperatures of the heat tank.

There are two possible methods that could be used to characterize the performance of EHPs and also GEHPs, as will be described further in this article [10].

The first technique would be to follow the prUNI/TS 11300-4 calculation method. In this case, it is possible to calculate the nominal exergy efficiency [Equation (3)] where there is knowledge of nominal data ($Q_{HP,nom}$ and COP_{nom}) at nominal temperatures ($T_{w,out,nom}$ and $T_{e,nom}$). Supposing this value is constant, it is then possible to assess the useful thermal power produced and the COP at various temperatures of the heat tanks [Equation (4)]:

$$\eta_{ex,nom} = \frac{\mathrm{COP}_{nom}}{\mathrm{COP}_{max\,,nom}} = \mathrm{COP}_{nom} \times \frac{T_{w,out,nom} - T_{e,nom}}{T_{w,out,nom} + 273.15} \tag{3}$$

$$\begin{aligned} Q_{HP} &= Q_{HP,nom} \times \frac{T_{w,out} - T_{e,nom}}{T_{w,out} - T_e} \\ \mathrm{COP} &= \eta_{ex,nom} \times \frac{T_{w,out} + 273.15}{T_{w,out} - T_e} \end{aligned} \tag{4}$$

An alternative method would be to use the HP manufacturer, Q_{HP} and COP data for a range of temperatures of the heat tanks. From this, it is possible to calculate the performance for operating conditions through the use of a double linear interpolation of the data. This second method of HPs characterization is more accurate than the first, and is utilized within this study. The performance data for the two different EHP models selected are shown in Tables 1 and 2.

Useful thermal power produced by EHP at given T_e and $T_{w,out}$ (Q_{CR}) may be lower than the Q_{HP} given by the capacity ratio CR:

$$\mathrm{CR} = \left.\frac{Q_{CR}}{Q}\right|_{T_e, T_{w,out}} \tag{5}$$

CR is a function of the climate (as building thermal loads vary with temperature, humidity and solar radiation) and also the rated capacity of the HP. The COP varies with CR by the correction factor f_{CR}:

$$f_{CR} = \left.\frac{\mathrm{COP}_{CR}}{\mathrm{COP}}\right|_{T_e, T_{w,out}} \tag{6}$$

f_{CR} is a characteristic of the HP, and it can be calculated through the use of manufacturer's data, or by means of simplified formulas suggested by the prEN 14825 standard. For example, for on–off air–water EHP [10, 11], the f_{CR} could be

Table 1. *Performance data of the lower capacity EHP [case (a)] ($Q_{HP,nom}$ = 29.1 kW_{th}, $P_{comp,HP,nom}$ = 7.3 kW_{el} at nominal conditions A7/W45).*

T_e (°C)	$T_{w,out}$ (°C) 35			40			45			50			55			60		
	kW_{th}	kW_e	COP	kW_{th}	kW_e	COP	kW_{th}	kW_e	COP	kW_{th}	kW_e	COP	kW_{th}	kW_e	COP	kW_{th}	kW_e	COP
−15	16.1	5.5	2.9	16.2	5.9	2.7	16.6	6.4	2.6	17	7.1	2.4	17.7	7.93	2.2	17.9	8.9	2.0
−10	19	5.7	3.3	19.1	6.1	3.1	19.3	6.6	2.9	19.6	7.29	2.7	20	8.12	2.5	20.5	9.1	2.3
−7	20.4	5.8	3.5	20.5	6.2	3.3	20.7	6.7	3.1	20.9	7.39	2.8	21.2	8.22	2.6	21.6	9.2	2.3
−5	22	5.8	3.8	22.1	6.3	3.5	22.2	6.8	3.3	22.3	7.52	3.0	22.5	8.34	2.7	22.8	9.3	2.4
2	26	6.0	4.3	26	6.5	4.0	26	7.1	3.7	26.1	7.81	3.3	26.2	8.66	3.0	26.2	9.6	2.7
7	29.1	6.2	4.7	29.1	6.7	4.3	29.1	7.3	4.0	29.1	8.06	3.6	29.1	8.93	3.3	28.6	9.9	2.9
10	30.5	6.2	4.9	30.4	6.8	4.5	30.4	7.4	4.1	30.5	8.18	3.7	30.5	9.06	3.4	30.0	10.0	3.0

Table 2. *Performance data of the higher capacity EHP [case (b)] ($Q_{HP,nom}$ = 40.9 kW_{th}, $P_{comp,HP,nom}$ = 10.8 kW_{el} at nominal conditions A7/W45).*

T_e (°C)	$T_{w,out}$ (°C) 35			40			45			50			55			60		
	kW_{th}	kW_e	COP	kW_{th}	kW_e	COP	kW_{th}	kW_e	COP	kW_{th}	kW_e	COP	kW_{th}	kW_e	COP	kW_{th}	kW_e	COP
−15	24.2	7.8	3.1	24.6	8.5	2.9	25	9.2	2.7	25.4	10.1	2.5	25.7	11.1	2.3	24.8	12.0	2.1
−10	27.3	8.0	3.4	27.5	8.7	3.1	27.8	9.5	2.9	28	10.4	2.7	28.1	11.4	2.5	28.3	12.5	2.3
−7	29.1	8.2	3.6	29.2	8.9	3.3	29.4	9.7	3.0	29.5	10.6	2.8	29.6	11.6	2.6	29.8	12.6	2.4
−5	31.1	8.3	3.7	31.1	9.1	3.4	31.2	9.9	3.2	31.2	10.8	2.9	31.3	11.8	2.7	31.4	12.8	2.5
2	36.6	8.7	4.2	36.5	9.5	3.8	36.4	10.4	3.5	36.3	11.3	3.2	36.2	12.3	2.9	36.1	13.4	2.7
7	41.4	9.1	4.5	41.1	9.9	4.1	40.9	10.8	3.8	40.7	11.8	3.4	40.5	12.8	3.2	39.3	13.8	2.9
10	43.7	9.3	4.7	43.4	10.1	4.3	43.1	11.0	3.9	42.8	12	3.6	42.6	13	3.3	41.3	14.0	3.0

found by the following equation:

$$f_{CR} = \frac{CR}{0.1 + 0.9 \times CR} \tag{7}$$

Once f_{CR} has been calculated for given values of T_e and $T_{w,out}$, it may be assumed that the trend observed as a function of CR is valid for all of the temperatures.

3 DEFINITION OF THE UTILIZATION

Assuming that the heat demand of a building is proportional to the difference between internal and external temperatures, and discounting any heat storage capacity within the structure of the building, the heating demand profile can be calculated in line with this proportional relationship.

It is also assumed that the power required to meet the design point of the heating system $T_{e,design}$ (i.e. the minimum ambient temperature where desired heating can be guaranteed) corresponds exactly with the maximum heating need of the building. The heating curve is defined using the heating system design point data and the heating limit of the building—$T_{H,off}$ (i.e. the ambient temperature where no heating is needed) [8, 10].

This study also considers two levels of heating circuit temperature: one with 40–30°C flow-return temperatures (typically used in modern buildings with large radiant surfaces heating plants) and the other with 60–50°C flow-return temperatures (older buildings with small area radiators). It is assumed that the flow and return temperatures for both cases are set to 18°C at an ambient temperature of 18°C ($T_{H,off}$), and the values for ambient temperatures in between are interpolated linearly to produce a linear heating 'curve'. Table 3 shows the data relating to the determination of the heating curve for the three climates under consideration.

4 ENERGY ANALYSIS

Several other factors, in addition to the COP and Q_{HP} characteristics, will influence the energy performance of the HP heating plant. The capacity of the HP with respect to building thermal load, and the type of operation, will also affect the performance. There are a number of different types of operations, including:

- monovalent plant: the HP capacity is chosen to be equal to building thermal load at $T_{e,design}$;
- alternative bivalent plant: HP capacity is chosen to meet building thermal load until $T_e > T_{H,cut\text{-}off}$, when $T_e < T_{H,cut\text{-}off}$ HP is off and thermal load is met by the back-up system;
- parallel bivalent plant: HP capacity is chosen to meet building thermal load until $T_e > T_{BV}$, when $T_e < T_{BV}$ HP is assisted by the back-up system.

Table 3. *Space heating plant characteristics.*

	Radiant surfaces (RS)		Radiators (R)	
	T_{flow} [°C]	T_{return} [°C]	T_{flow} [°C]	T_{return} [°C]
$T_{e,design}$ (°C)				
−5 MI, 0 FL, 0 RM	40	30	60	50
$T_{H,off}$ (°C)	18	18	18	18

While the performance of traditional boiler systems may be calculated on a monthly basis, seasonal energy analysis of air–water HP plant should take into account the variation of heat source (air) and heat sink (water) temperatures throughout the calculation period (typically 1 h). In such cases, the T_e range is divided into bins, as suggested by the prUNI/TS 11300-4 standard. This document describes a calculation method for the determination of bins by a normal distribution of T_e based on monthly mean external temperature (UNI 10349) [9], $T_{e,design}$ (UNI EN 12831) [12] and the sum of diffuse and beam horizontal monthly mean daily solar radiation values. It is suitable for use in situations where specific climate data are not available.

In this article, the bin distribution of the three resorts for each month of the heating season (October–April) has been calculated by use of the prUNI/TS 11300-4 method.

The building energy requirement at the heat distribution system inlet (that is at the HP outlet), E_{month}, is then calculated, as the sum of thermal energy needs per bin:

$$E_{month}(\text{kWh}) = \sum_{\text{bin}=-10}^{T_{H,off}} \text{UA} \times (T_{set,in} - T_{bin}) \times t_{bin,month} \qquad (8)$$

Table 4 includes information relating to the building characteristics used for the energy calculations. Considering an operating time of $\Delta t_{ON} = 12$ h/day and $n_g = 26$ days/month, it is possible to calculate the monthly mean thermal power that will be produced by the heating plant [$Q_{month} = E_{month}/(n_g \times \Delta t_{ON})$]. These values may be presented as a function of external monthly mean temperature (energy signature, UNI EN 15603 [13]) (Figure 1). The value for the energy signature will be the same for all three buildings, as they each have identical values for global heat transmission UA and $T_{H,off}$ (Table 4). The calculated value of UA is typical of a small residential new (2011) building of four flats. From Figure 1, it is possible to obtain the design thermal loads as 62, 49 and 49 kW_{th} for Milan, Florence and Rome, respectively.

It is important to note that the calculated design loads are quite high when compared with ordinary operating conditions of the buildings. This suggests that the selection of HP nominal capacity on the basis of the design conditions may not optimize heating plant energy performances. For this reason, further analysis has been undertaken using two types of HP (Tables 1 and 2). In both cases, the capacity of the HP is lower than the building design thermal load, and therefore a

Table 4. *Building characteristics.*

UA (W/K)	1200
$\triangle t_{on}$ (h/day)	12
n_g (days/month)	26

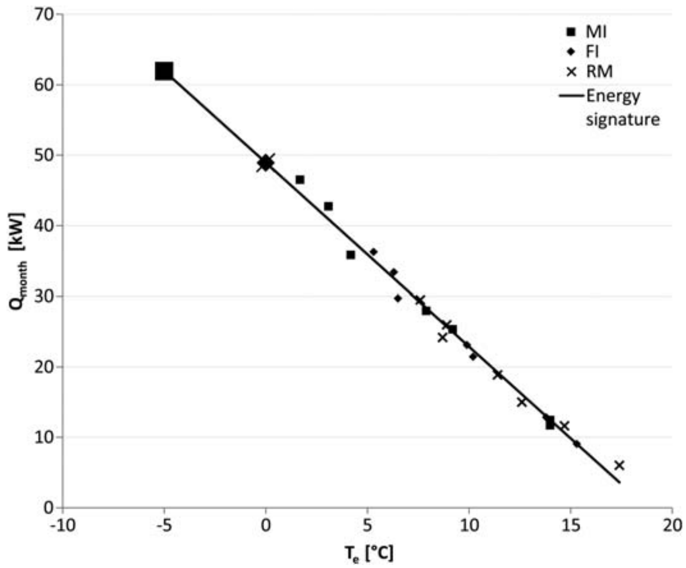

Figure 1. *Energy signature and thermal loads of the building in the three locations.*

natural gas boiler with a thermal mean efficiency of $\eta_{th} = 95\%$ is used as a back-up measure, resulting in the provision of a parallel bivalent system.

The monthly thermal energy needs can be divided into bins using the equation:

$$E_{bin,month}(\text{kWh}) = E_{month} \frac{(\Delta t_{ON}/24) \times t_{bin,month} \times (T_{H,off} - T_{bin})}{\sum_{bin} (\Delta t_{ON}/24) \times t_{bin,month} \times (T_{H,off} - T_{bin})} \qquad (9)$$

where $\Delta t_{ON}/24$ is the HP daily fraction operating time.

The mean thermal power to be produced and supplied to the building per bin per month is given by the ratio between $E_{bin,month}$ and HP operating time:

$$Q_{bin,month}(\text{kW}) = \frac{E_{bin,month}}{(\Delta t_{ON}/24) \times t_{bin,month}} \qquad (10)$$

This power has to be compared with the thermal power produced by the HP at full load, which can be calculated by interpolation of the manufacturer's data (Tables 1 and 2), in order to calculate the capacity ratio per bin per month:

$$\text{CR}_{bin,month} = \begin{cases} 1 & \text{if } Q_{bin,month} > Q_{HP,bin} \\ \dfrac{Q_{bin,month}}{Q_{HP,bin}} & \text{if } Q_{bin,month} < Q_{HP,bin} \end{cases} \qquad (11)$$

Using Equation (7), it is possible to calculate f_{CR}. Due to the nature of air source HPs, it is necessary to take into account a defrost penalization factor (f_{frost}), which is calculated as a function of the temperature and relative humidity of external air [14]. The COP in operating conditions, per bin per month, can be found by:

$$\mathrm{COP}_{CR} = \mathrm{COP} \times f_{CR} \times f_{frost} \tag{12}$$

The following equation will assess the thermal energy that is effectively produced by the HP per bin per month:

$$E_{HP,bin,month}(\mathrm{kWh}) = Q_{HP,bin,month} \times \left(\frac{\Delta t_{ON}}{24}\right) \times t_{bin,month} \times \mathrm{CR}_{bin,month} \tag{13}$$

Finally, the electrical (EHP) or mechanical (GEHP) energy that is consumed by the HP per bin per month can be found using:

$$W_{HP,bin,month}(\mathrm{kWh}) = \frac{E_{HP,bin,month}}{\mathrm{COP}_{CR}} \tag{14}$$

The difference between the values will give a value for the thermal energy produced by the back-up boiler:

$$E_{back-up,bin,month}(\mathrm{kWh}) = E_{bin,month} - E_{HP,bin,month} \tag{15}$$

Therefore, the primary energies consumed by the systems are:

$$\begin{aligned} E_{prim,EHP,bin,month}(\mathrm{kWh}) &= \frac{W_{EHP,bin,month}}{\eta_{el}} \\ E_{prim,GEHP,bin,month}(\mathrm{kWh}) &= \frac{W_{GEHP,bin,month}}{\eta_{ICE}} \\ E_{prim,back-up,bin,month}(\mathrm{kWh}) &= \frac{E_{back-up,bin,month}}{\eta_{th}} \end{aligned} \tag{16}$$

An example of the energy trends calculated for one of the case studies can be found in Figure 2, and it shows the thermal energy produced by the HPs [$E_{(G)EHP,season,bin,RS}$] and the back-up systems [$E_{backup,(G)EHP,season,bin,RS}$], when compared with the thermal energy need ($E_{bin,season}$). It can be seen that the value for $T_{H,cut-off}$ is -5°C, while T_{BV} is lower for GEHP than for EHP. T_{BV} relates to bivalent temperature, which is the external temperature until which the HP fully meets the thermal load of the building. This means that the T_e operating range of GEHP is extended when compared with EHP. Therefore, GEHP has an advantage in terms of total energy use as there is reduced requirement for the use of the secondary back-up heating system.

4.1 GEHP characterization

The test data for EHPs can be adjusted in order to consider the efficiency of electric motors and possible auxiliary drives associated with the mechanical energy demand of GEHPs.

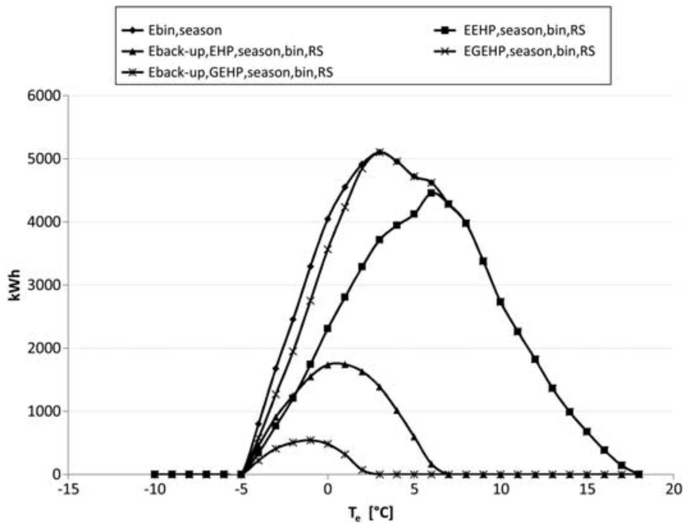

Figure 2. *Energies as a function of external air temperature [for the case MI(a) and the radiant surfaces heating plant] for the parallel bivalent systems under consideration. Note that bivalent temperatures (temperatures until which HPs are able to meet the whole heating load) are ~7 and 2°C, respectively, for EHP and GEHP, and the consequent trends of* $E_{back-up}$.

When the compressor is driven by a natural gas engine, the efficiency behaviour of the engine allows a straightforward calculation of the fuel energy input. Available heat flows, and the use of these for space heating, can be evaluated through assessment of the energy balance of the engine.

In order to estimate the coefficient of performance (COP_{GEHP}) relating to the mechanical work (W_{GEHP}) at the compressor's shaft from measured electrically driven HP systems, the efficiency ($\eta_{em} = W_{GEHP}/P_{comp,HP}$) of the electric motor has to be known (here it is assumed to be $\eta_{em} = 95\%$).

For air-to-water systems, the pumping energy for the ambient heat has to be taken into account. This is assessed through evaluation of relative pumping effort when compared with the energy demand of the compressor ($h = P_{vent,HP}/P_{comp,HP}$). Values can be derived from the manufacturer's data, and in this study, values of $h = 0.14$ and 0.09 were used for Table 1[case (a)] and for Table 2 [case (b)], respectively. Thus, the mechanical COP_{GEHP} can be calculated from its known electrical COP_{EHP}:

$$\begin{aligned} \mathrm{COP}_{GEHP} &= \frac{Q_{HP}}{W_{GEHP}} \\ &= \frac{Q_{HP}}{P_{comp,HP} + P_{vent,HP}} \times \frac{(P_{comp,HP} + P_{vent,HP})/P_{comp,HP}}{W_{GEHP}/P_{comp,HP}} \\ &= \mathrm{COP}_{EHP} \times \frac{1+h}{\eta_{em}} \end{aligned} \tag{17}$$

In the case of GEHPs, the waste heat generated by the engine is available for heating purposes. In this study, it is assumed that the heat from both the engine cooling circuit and the exhaust gas is combined to provide a single engine waste heat value Q_{ICE}.

Furthermore, it is considered that the HP and then the engine heat exchangers are connected in series. This allows the engine waste heat to be utilized to its full potential. In simple terms, the temperature level and the temperature difference which have to be provided by the HP circuit are reduced and the efficiency of the HP is increased [8] (Figure 3).

The heating power of the HP circuit Q_{GEHP} is a function of the engine mechanical power multiplied by the HP COP_{GEHP}:

$$\begin{aligned} Q_{GEHP} &= W_{GEHP} \times COP_{CR,GEHP} \\ &= \eta_{em} \times P_{comp,HP} \times COP_{CR,GEHP} \end{aligned} \tag{18}$$

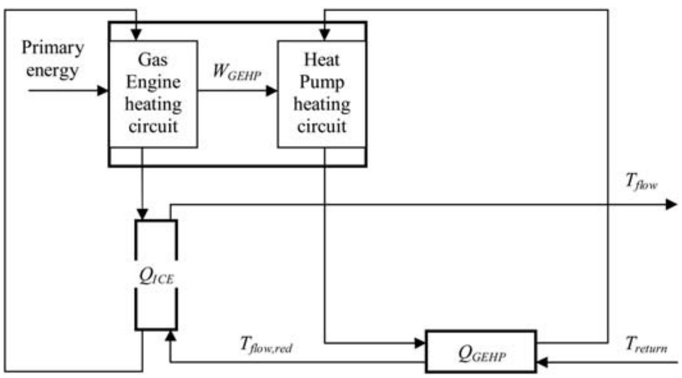

Figure 3. *Block diagram of the GEHP.*

Assuming unchanged flow in the heating circuit of the building, the energy balance leads to the reduced flow temperature that is desired from the HP $T_{flow,red}$ (Figure 3):

$$T_{flow,red} = T_{flow} - k \times (T_{flow} - T_{return}) \tag{19}$$

In this equation, k describes the fraction of the engine's heating power Q_{ICE} in relation to the total heating power ($Q_{ICE} + Q_{GEHP}$). The iterative numerical process that allows determination of the correct engine operating point is described in Brenn *et al.* [8]. When the value of k for each bin per month is known, it enables the calculation of Q_{ICE}:

$$Q_{ICE} = \frac{k}{1-k} \times Q_{GEHP} \tag{20}$$

while Q_{GEHP} is calculated by means of Equation (18).

4.2 Seasonal energy performances

It is interesting to evaluate three seasonal performance indexes for the bivalent systems considered in this study: $COP_{HP,season}$, $PER_{HP,season}$ and $PER_{tot,season}$.

The first index concerns the ratio between useful thermal energy produced and electrical (EHP) or mechanical (GEHP) energy consumed by the HP during the heating season:

$$COP_{HP,season} = \frac{\sum_{month=October}^{April} E_{HP,month}}{\sum_{month=October}^{April} W_{HP,month}} \tag{21}$$

The second factor comprises the ratio between useful thermal energy produced and the primary energy consumed by the HP

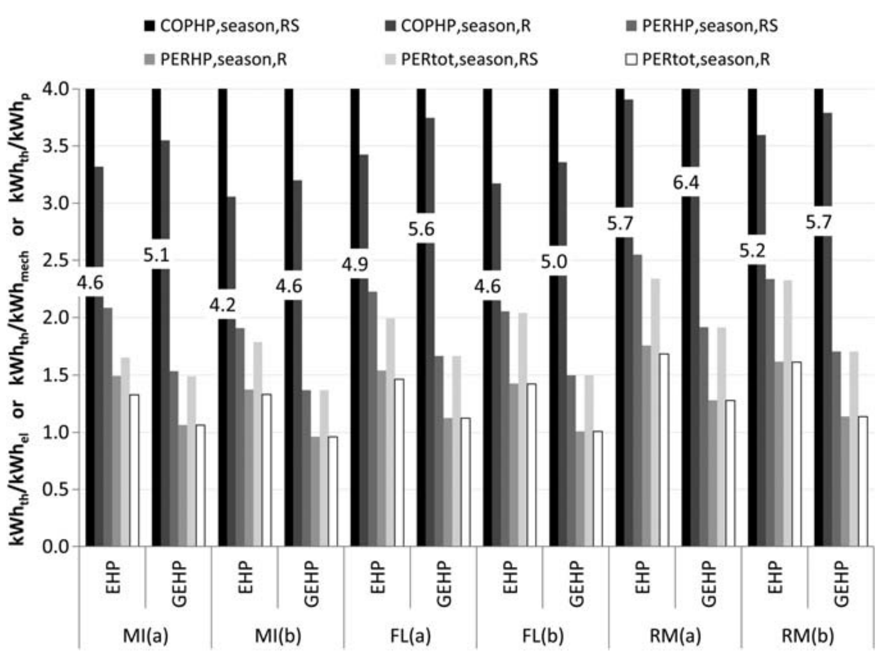

Figure 4. *Seasonal performance indexes of Equations (21–23) for the electrical and GEHP systems, for both the HP models [model (a) refers to Table 1, model (b) to Table 2], the two heating circuit types (RS, radiant surfaces, R, radiators), and for the three climates.*

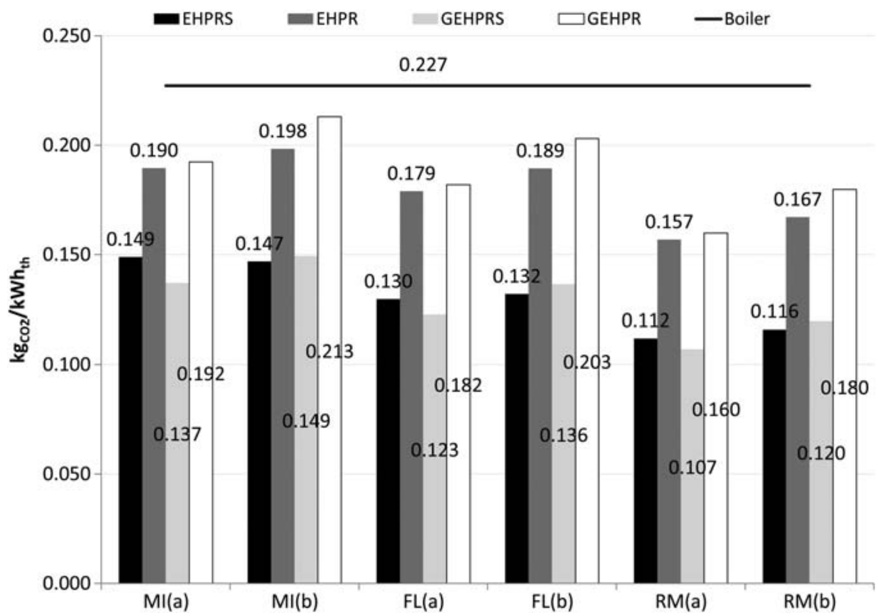

Figure 5. *CO_2 emissions per unit of useful thermal energy.*

during the heating season, taking into account typical values of global thermoelectric efficiency, $\eta_{el} = 45\%$, and engine mechanical efficiency, $\eta_{ICE} = 30\%$, respectively, for EHPs and GEHPs:

$$\mathrm{PER}_{(G)EHP,season} = \frac{\sum_{month=October}^{April} E_{(G)EHP,month}}{\sum_{month=October}^{April} W_{(G)EHP,month}/\eta_{el(ICE)}} \quad (22)$$

The final parameter is the ratio between useful thermal energy produced and the primary energy consumed by the whole system (HPs + boiler) during the heating season:

$$\mathrm{PER}_{tot,season} = \frac{\sum_{month=October}^{April} E_{(G)EHP,month} + E_{back-up,month}}{\sum_{month=October}^{April} (W_{(G)EHP,month}/\eta_{el(ICE)}) + E_{primary,back-up,month}} \quad (23)$$

The results of this analysis are shown in Figure 4. In terms of the COP, GEHP performs better than EHP due to the heating power of the engine Q_{ICE} and the reduced flow temperature that is produced by the HP [Equation (19)]. This observation does not vary considerably with climate, but is more important for lower rated capacity HPs [case (a) in Figure 4] and for low-temperature heating circuits (RS).

With regard to the primary energy ratio indexes, it is interesting to observe that, with the hypotheses considered, EHP allows higher efficiency than GEHP mainly in milder climates. Other parameters do not seem to exert significant influence on this aspect. The increasing value of global thermoelectric efficiency η_{el} in Italy during recent years [15], associated with a relative low internal combustion engine efficiency η_{ICE}, supports such observations. Finally, it can be seen that the back-up system has more impact on the results in colder climates (PER_{tot} is lower) and for (b) cases, because the higher HP capacity means that HPs operate within the range CR < 1 for a longer period of time.

5 CO_2 AND ECONOMIC ANALYSIS

In relation to CO_2 emissions, Figure 5 reports the specific values calculated as follows [16] for boilers and GEHPs:

$$\frac{2.75(\mathrm{kg_{CO_2}/kg_{CH_4}}) \times 0.7139(\mathrm{kg_{CH_4}/m^3_{CH_4}})}{9.6(\mathrm{kWh_{CH_4}/m^3_{CH_4}})} \times \left(\frac{(1-\alpha)}{\mathrm{PER_{GEHP}}(\mathrm{kWh_{th}/kWh_{CH_4}})} + \frac{\alpha}{\eta_{th}(\mathrm{kWh_{th}/kWh_{CH_4}})}\right) \quad (24)$$

And as below for EHPs.

$$\frac{(1-\alpha)\times(0.6)(\mathrm{kg_{CO_2}/kWh_{el}})}{\mathrm{COP_{EHP}}(\mathrm{kWh_{th}/kWh_{el}})} + \left(\frac{2.75(\mathrm{kg_{CO_2}/kg_{CH_4}}) \times 0.7139(\mathrm{kg_{CH_4}/m^3_{CH_4}})}{9.6(\mathrm{kWh_{CH_4}/m^3_{CH_4}})} \times \frac{\alpha}{\eta_{th}[\mathrm{kWh_{th}/kWh_{CH_4}}]}\right) \quad (25)$$

The 2.75 factor is obtained through use of a stoichiometric CH_4 burning equation. The α factor is the percentage of heat load supplied by integrative boilers. CO_2-specific emissions for

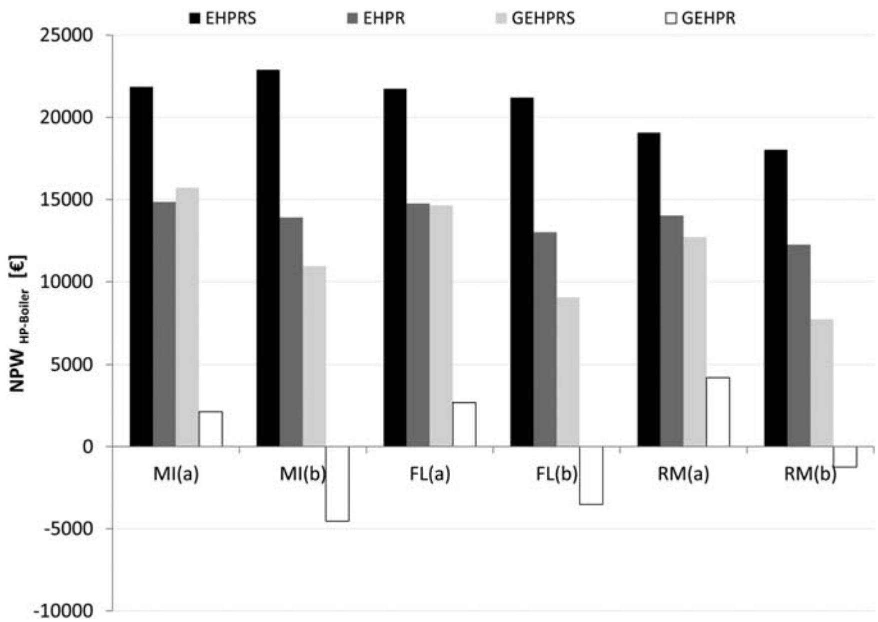

Figure 6. *Differential NPWs of the scenarios under consideration (electrical and GEHPs, coupled to radiant surfaces and radiators heating plants, for two heat pumps capacities, and for three different climates) when compared with a conventional system (natural gas boiler).*

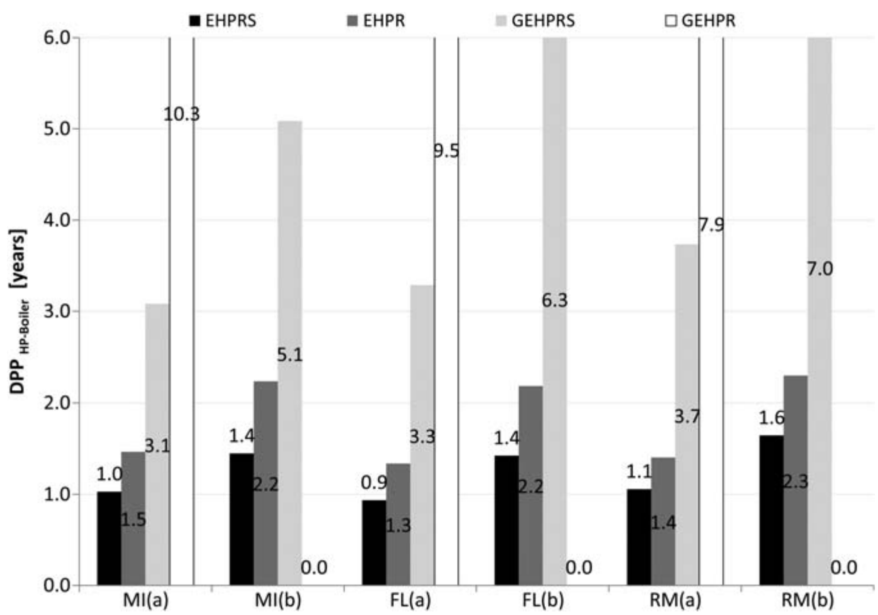

Figure 7. *DPP of the scenarios in Figure 6 when compared with a conventional system (natural gas boiler).*

electricity production have been fixed to the mean value for Italian thermoelectric plants (0.6 kg_{CO_2}/kWh_{el}).

It has been observed that all of the HPs perform better than conventional natural gas boilers. GEHPs result in lower specific emissions in cases (a), where rated capacity is undersized with respect to building thermal load and so there are greater operating times spent within CR = 1. However, this is true only with radiant surface heating plant, while in all other cases EHPs demonstrate better performance.

Finally, an economic analysis has been undertaken in order to complete the comparison, in terms of both net present worth (NPW) (Figure 6) and discounted payback period (DPP) (Figure 7). Assumptions made are included in Table 5, where the interest rate and the time period of the analysis are, respectively, 3% and 15 years.

Electric HP systems appear to be economically more viable when RS heating plants are considered, with NPW always being positive and the DPP lower than 1.5 years. This reduces

Table 5. *Hypotheses of the economic analysis [16].*

	EHP	GEHP	Boiler
Appliance investment cost	500 €/kW_{el}	250 €/kW_{th}	75 €/kW_{th}
Electricity/natural gas cost	14 c€/kWh_{el}	0.6 €/m^3	0.6 €/m^3
Ordinary maintenance cost	10 €/(kW_{el} year)	5 €/(kW_{th} year)	3.5 €/(kW_{th} year)

further to ~1 year when HPs are undersized with respect to building thermal load. High-temperature heating plants such as radiators result in a less favourable economic analysis, particularly when considering GEHPs. In this case, it is possible that conventional natural gas boilers would perform better [i.e. case (b), with negative NPWs].

6 CONCLUSIONS

This article reports on the recent development of standards for the calculation of energy performance of non-conventional heating systems, such as HPs, coupled with integrative boilers. A method to calculate the performance of natural gas engine-driven HPs, derived from known performance data of electrically driven HPs, has also been presented. In the case of GEHPs, it is possible to use the heat from the engine for space heating, which leads to reduced temperature levels and temperature differences for the HPs. Therefore, they are able to achieve very good yearly efficiencies. The energy advantage with respect to EHPs depends strongly on the thermoelectric efficiency η_{el} and the internal combustion engine efficiency η_{ICE}. When considering typical values used at present in Italy, GEHPs seem to be less efficient than EHPs. This is potentially due to the great improvements that have been made in the thermoelectric plant efficiency η_{el}, largely as a result of the increased use of gas turbine combined cycles to power thermoelectric plants.

In all situations, the choice of a low flow temperature level is essential to achieve an energy saving when utilizing HP technologies. It is therefore advisable that new buildings are constructed to utilize large heating surfaces such as heating floors or heating walls, and that in older buildings high temperature radiators are replaced with low-temperature models [17].

The assessment shows that natural gas engine-driven HPs have the potential to reduce primary energy use by ~60% when compared with condensation boilers. This reduction is dependent upon the required temperature level of the heat and the nature of the ambient heat source. Electrical HPs can further improve this performance, and will have a greater impact on CO_2 emissions savings. However, this impressive reduction in primary energy use should be considered against the higher system costs incurred due to greater technical complexity.

When evaluating the hypotheses set at the commencement of this study, it can be seen that the economic analysis reveals that financial advantages for end-users are possible with regard to EHPs. However, in the case of GEHPS, such returns are not guaranteed and are largely dependent on flow temperatures (i.e. the heating plant) and the sizing of the system in place.

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