Air-to-water heat pumps: the impact of climate, compressor technology, water output temperature and sizing on the seasonal coefficient of performance for heating

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Abstract. Air-to-water heat pumps (AWHPs) are devices that will broadly replace heating systems based on fossil fuels. The performance of AWHPs in heating operation is strongly influenced by various parameters such as the climate of the location, the hot water output temperature to the heat emitters, the oversizing or downsizing of the HP with respect to the thermal load of the building, the control system, and the heat pump’s compressor technology (fixed-capacity and inverter-driven HPs). The aim of this work is to study the impact of those parameters on the seasonal efficiency for heating of AWHPs in buildings at representative cities of the four Greek climatic zones. The active mode seasonal coefficient of performance (SCOPon) was estimated in various case studies, with the method proposed in European standard EN 14825, by using climatic data of the four locations. The results show the positive effect of lowering the water supply temperature to the heating system and of using inverter technology and control system with compensation on AWHPs efficiency. Additionally, it is demonstrated that the optimal performance of inverter-driven HPs appears when they are sized at the design outdoor air temperature, while in the case of the fixed-capacity HPs when they are down-sized.

1. Introduction

The building sector is a major energy consumer and one of the most significant contributors to greenhouse gas (GHG) emissions. In 2017, European building stock accounted for approximately 42% of the total final energy consumption [1]. In particular, space heating and domestic hot water in households contributed with 64.1% and 14.8% respectively in the total energy consumption [2]. Several studies have shown that the building sector offers large potentials for energy savings. Constructing and renovating buildings with advanced design techniques and materials, energy upgrade of heating and cooling (H&C) systems and expanding the use of renewable energy technologies, are measures which can minimize the energy consumption, improve energy efficiency and reduce GHG emissions.

Towards 2030 and 2050, the European Union (EU) has put ambitious goals for the reduction of fossil primary energy consumption and the related CO₂ emissions. Firstly, all new buildings must be nearly Zero Energy Buildings (nZEB) by 31 December 2020. The targets for 2030 are at least 40% cuts in GHG emissions from 1990 levels, at least 32% share for renewable energy, and at least 32.5% improvement in energy efficiency. Furthermore, by 2050, the EU has set itself a long-term goal of reducing GHG emissions by 80-95%, when compared to 1990 levels [3]. For achieving these targets, a
series of Directives were introduced by the EU which impose the significant reduction of energy usage in both new and old buildings and promote the integration of high-efficiency heating, ventilating and air-conditioning (HVAC) technologies, based on renewable energy sources.

Heat pumps (HPs) are today efficient and commercially competitive heat generation devices of mature and reliable technology [4], with no harmful emissions given off locally, which can contribute significantly to attain the EU goals. HPs provide either heating or cooling, by operating in reverse, and represent a valid alternative to conventional systems because they use less primary energy. Although they require high-cost and high carbon footprint electricity to operate, the majority of the energy pumped is renewable heat drawn from the environment. Furthermore, with the decarbonization of the electricity sector the GHG from HPs operation will be reduced to very low levels. Also, H&C systems with HPs and water thermal storage may help to balance the differences between the electric energy production from renewable energy sources (RES) and use, playing the role of thermal batteries.

Particularly, air-to-water heat pumps (AWHPs) are expected to replace widely oil or gas boilers, especially in low temperature hydronic heating systems. AWHPs are able to operate at a wide range of outdoor temperatures, have low operating cost in low hot water temperatures [5], their size is small, and they are installed easily with relatively low cost. However, even if AWHPs are characterized by high efficiencies when the temperature difference between the heat source and the heat sink is small, their thermal output and performance factor are reduced when ambient temperatures are low and a high temperature lift is needed.

Usually, monovalent AWHPs are sized with a heating capacity equal to the heat load of the building at design conditions or sometimes are oversized, which means that they operate under full load conditions only for a limited part of the heating period. On the other side, bivalent AWHPs have an auxiliary heat source which is turned on whenever the HP’s capacity is insufficient to satisfy the building heating demand. In HPs with small heating capacity, this heat source is usually an electric resistance.

The performance of AWHPs is varied continuously, due to the continuous changes of ambient temperature on hourly and daily basis, but it is influenced as well by various other factors related to the HP’s technology, operation conditions and control system. Several researches which analyze the performance of AWHPs are available in the literature. Francesco Madonna and Francesca Bazzocchi [6] developed a model that simulates the hourly efficiency of AWHPs with variable speed compressor, based on thermodynamic equations and on monitoring data, and estimated the seasonal performance factors of AWHPs in a set of residential buildings located in different Italian cities. The obtained results showed that the climate is the factor that mainly affects the annual performance especially in the case of no oversizing. If HPs are oversized, a seasonal efficiency reduction may be caused up to 25%. It was also demonstrated that seasonal performances are increased by applying a weather compensation strategy. The HP sizing is also an important aspect which must be considered in order to have maximum performance conditions. Claudia Naldi et al. [7] analyzed the SCOP of on-off mono-compressor, multi-compressor and inverter-driven compressor AWHPs, integrated by electric heaters as back-up system, in several buildings located in three different Italian cities with different climate. They demonstrated that the optimal sizing of an AWHP is related to the compressor technology. Matteo Dongellini et al. [8] calculated the seasonal performance for H&C of a reversible AWHP coupled to an office building, which was placed in three different locations representative of Colder, Average and Warmer heating reference climates indicated by the standard EN 14825 [9]. Mono-compressor with on-off operation, multi-compressor and inverter-driven HPs with electric resistances as back-up heaters were considered in the study. The obtained results pointed out that the correct sizing of the HP is a crucial parameter that influences the energy performance and that the inverter-driven and multi-compressor HPs show larger SCOP than the on-off HPs. In another work, Matteo Dongelini et al. [10] simulated the operation of three different models of AWHPs with similar rated performances at full load, coupled with radiant floor heating system in 11 different buildings in Bologna, and evaluated the SCOP of the system in each case study. They highlighted the importance of the correct sizing of a HP in order to obtain high seasonal efficiency and showed that, for a fixed
thermal load, inverter-driven and multi-compressor heat pumps have to be slightly oversized with respect to mono-compressor ones in order to obtain for the same building the highest SCOP values. Matteo Dongellini and Gian Luca Morini [11] investigated the effect of on-off cycling, during partial load operation, on the SCOP of fixed-capacity (single-stage units and multi-stage units) and inverter-driven AWHPs by means of a series of innovative TRNSYS models not available in the software standard library. They pointed out that the energy losses introduced during the on-off cycling has a significant impact on the overall energy performance only for single-stage units, with a decrease of the SCOP up to 12%. Huchtemann and Müller [12] modelled and simulated a heating system with an AWHP and buffer tank in a model building and examined the influence of the supply water temperature to the heating system on the seasonal performance factor. The results showed that with a control strategy that adapts the supply set temperature according to the actual load, the seasonal performance factor for heating can be increased by up to 7%. Claudia Naldi et al. [13] developed a simulation tool in MATLAB to analyze the SCOP of an AWHP heating system, located in northern Italy, as a function of the bivalent temperature and of the water storage volume. It was demonstrated that the choice of the right bivalent temperature can significantly increase the SCOP, while an increase of the storage tank volume is usually ineffective and can even reduce the performance. Moreover, inverter-driven AWHPs have a higher SCOP and a lower optimal bivalent temperature, with respect to the on–off ones.

The aim of this paper is to study how the seasonal efficiency for heating of AWHPs is influenced by various parameters. The active mode seasonal coefficient of performance during a heating season (SCOP_{on}), as defined in [9], is estimated in various case studies categorized by a) the heating capacity of the HP in accordance to the heat load of the building, b) the HP’s compressor technology (fixed-capacity or inverter-driven), c) the water output temperature to the heat emitters of the heating system (35°C and 45°C), and d) the control system (operation with or without weather compensation). The calculation of the SCOP_{on} for the various case studies was performed with the method proposed in [9] by using climatic data of four Greek cities (Ierapetra, Athens, Thessaloniki and Florina), representative of the four Greek climatic zones (A, B, C, D) [14]. The results of this work may be helpful in selecting the right equipment in a heating system with AWHP in order to have maximum seasonal performance.

2. Heat pump performance factors

An AWHP heating system consists of a heat source, the HP unit and a distribution system that delivers the heat to a thermal zone or a building. The temperature difference between the outdoor air, from which heat is drawn, and the water to which heat is added, is called temperature lift. Since the efficiency of a HP is increased as the temperature lift decreases, low-temperature heat distribution systems such as radiant-floor or wall heating are the ideal partners for AWHPs [12]. The most common HP type is the electrically driven compression HPs, usually applied in comfort applications of residential and commercial building sector. HPs with a fixed-capacity compressor (mono-compressor) are able to operate only at full power, so they regularly switch on and off in order to operate in part-load conditions and maintain a given internal temperature. With the multi-compressor technology, the compressors are switched on and off according to the heat load, and the thermal power delivered by the HP depends on the number of activated compressors. The fully modulated inverter-driven technology gives the ability to match the power output to the heating demand of the building and thus control precisely the supply water temperatures.

The efficiency of a HP in the heating mode is given by the Coefficient of Performance (COP), defined as the heat output divided by the electrical input at a given set of temperature conditions. The COP is measured under steady-state operating conditions in a lab environment, according to EN14511-3 [15] and represents a momentary performance. The equation which gives the COP is the following:

\[
COP = \frac{\text{Heating Capacity (W)}}{\text{Electrical Input (W)}} \tag{1}
\]
While the COP states a momentary efficiency, the Seasonal Coefficient of Performance (SCOP) represents the average performance of the heating season and is calculated by taking into account the climate data of a given location, the building characteristics and energy needs, the settings and the operation conditions. The SCOP is the ratio of the annual heating demand (expressed in thermal Wh) to the annual electricity consumption (expressed in Wh of electricity) and is an index of the operating cost of the HP. Additionally, the active mode Seasonal Coefficient of Performance (SCOP$_{on}$) is the ratio of the heating demand during the heating season to the respective electricity consumption [9]. It is obvious that both SCOP and SCOP$_{on}$ should be as high as possible so that the operating costs are reduced. The SCOP and SCOP$_{on}$ are determined by the following equations:

$$\text{SCOP} = \frac{\text{Annual Heating Demand (Wh)}}{\text{Annual Electricity Consumption (Wh)}}$$  \hspace{1cm} (2)

$$\text{SCOP}_{on} = \frac{\text{Seasonal Heating Demand (Wh)}}{\text{Seasonal Electricity Consumption (Wh)}}.$$  \hspace{1cm} (3)

In equations (2) and (3), the heating demand is equal to the output heating energy from the HP while the electricity consumption is the input electrical energy (in compressor and auxiliary electric resistance) to the HP.

3. Climate data and bin distribution

For the estimation of the SCOP and SCOP$_{on}$ values, the calculation procedure of the standard EN 14825 [9] is applied. According to the standard, the energy calculations are performed by means of the bin method [16], in order the variation of outdoor dry bulb temperature and the part-load performance to be considered. For the application of the method, special weather data are needed based on long-term hourly measurements, which are called “bin data”. A bin is a temperature interval, usually of 1K or 2K size, which contains the number of average hours of occurrence of the outdoor temperature during a month or season. The standard EN 14825 classifies Europe into three different climate zones (Colder, Average and Warmer, with design temperatures for heat load of -22°C, -10°C and 2°C respectively) and provides the bin data for each of the climate zones. Greece is classified in the “Warmer” zone and the bin data given of this zone are supposed to be used for the calculation of SCOP and SCOP$_{on}$ values in all country locations.

![Figure 1. Bin distribution in the four cities for the heating period.](image-url)
In this work SCOP values were calculated by using the local bin data of four representative cities, one in each of the four climatic zones of Greece, in order to take into account the particular climatic conditions of the various climate zones in Greece. These cities are: Ierapetra (35.0 N, 25.44 E, in zone A), Athens (37.98 N, 23.72 E, in zone B), Thessaloniki (40.64 N, 22.94 E, in zone C) and Florina (40.72 N, 21.57 E, in zone D) with outdoor design temperatures for heat load of 7°C, 2°C, -2°C and -7.5°C respectively. In figure 1 the bin distributions for the heating season (October 1st to April 30th) of the four selected locations are presented. It is evident that the heating season in Florina is the coldest among the four cities. Thessaloniki is also characterized by a cold heating season but milder than in Florina. Athens and Ierapetra both feature mild winters but the frequency of occurrence in [h] of temperatures above 16°C, when normally no heating is needed, is higher in Ierapetra than in Athens.

4. Calculation procedure

The first step is the estimation of the building heating load $P_h(i)$ in [W] at each bin, determined as:

$$P_h(i) = P_{des} \cdot \frac{T_{bal} - T_{oa}(i)}{T_{bal} - T_{des}}, \quad (4)$$

where $T_{oa}(i)$ is the corresponding outdoor dry bulb air temperature [°C] at the i-th bin, $T_{des}$ is the design outdoor temperature [°C] of the building location, $T_{bal}$ is the balance-point temperature [°C], and $P_{des}$ is the design heat load of the building in [W], calculated according to the standard EN12831. The balance-point temperature $T_{bal}$ is the outdoor temperature [°C] at which, for a given inside building temperature, the total thermal losses of the building are equal to its internal and solar heat gains, that is the outdoor temperature at which the building heating demand becomes zero. The value of 16°C is suggested as $T_{bal}$ for all climate zones in Europe.

The building thermal energy demand $Q_h(i)$ in [Wh] at each bin, is calculated as a function of $P_h(i)$.

$$Q_h(i) = P_h(i) \cdot t_{bin}(i) \quad (5)$$

where $t_{bin}(i)$, is the occurrence in [h] of the i-th bin.

The next step is the calculation of the thermal power $P_{hp}(i)$ in [W], delivered by the HP to the heating system, in each temperature bin. The $P_{hp}$ is the heating capacity of the HP and is obtained from the manufacturers’ technical data-books as a function of the outdoor temperature $T_{oa}$ and the supply water temperature to the heating system $T_w$.

![Figure 2. Building heating load and HP’s heating capacity curves.](image-url)
Figure 2 gives an example of the building’s heating load curve (BHL) and the heating capacity characteristic curve(s) of an AWHP. The BHL curve corresponds to a heating load equal to 7.0 kW at a $T_{\text{des}}$ of -2°C and a $T_{\text{bal}}$ equal to 16°C. The red line represents the heating capacity characteristic curve of a typical mono-compressor AWHP with on-off operation. The intersection (point B) between the building heat load curve (yellow line) and the HP’s heating capacity curve (red curve), which is the point where the thermal output of the HP matches the building heat load without any supplementary heating, corresponds to an outdoor air temperature which is called bivalent temperature ($T_{\text{biv}}$). When $T_{\text{oa}}$ is below $T_{\text{biv}}$, the building heating demand cannot be covered by the HP’s heating power capacity, and a back-up heater (usually an electric heater) must be activated. On the contrary, when $T_{\text{oa}}$ is higher than $T_{\text{biv}}$, the heating capacity of the HP exceeds the building heating demand and the HP is forced to operate in partial load with on-off cycles.

For a multi-compressor or inverter driven AWHP, the heating capacity is represented by a number of curves, corresponding to the activated compressors or to the inverter frequencies respectively. In Figure 2, the red and the blue curves ($P_{\text{hp max}}, P_{\text{hp min}}$) represent the heating capacities of a two-compressor AWHP or the maximum and minimum capacity for an inverter-driven AWHP at maximum and minimum frequency respectively. When $T_{\text{oa}}$ is below $T_{\text{biv}}$, the operation is similar to the mono-compressor AWHP, as described above. When $T_{\text{oa}}$ is higher than $T_{\text{biv}}$ and lower than $T_{\text{biv max}}$ (point C in figure), then the AWHP operates in partial load by reducing the number of the activated compressors or the working frequency respectively. Multi-compressor HPs start the on-off cycles and operate with only one compressor while inverter-driven HPs (IDHP) at the inverter frequency corresponding to the heating demand. When $T_{\text{oa}}$ is higher than $T_{\text{biv max}}$, namely when the heating demand is lower than the minimum HP’s heating capacity, then both types are in on-off operation. Additionally, all HP’s heating capacity curves end at a certain minimum temperature TOL (-6°C in figure), which is the operational limit of the HP below which the HP declared capacity is equal to zero.

When the HP is operating at full load, that is when $T_{\text{oa}}<T_{\text{biv}}$, the COP values are calculated from the $P_{\text{hp}}(i)$ delivered by the HP and the electricity input power $P_{\text{el}}(i)$ to the HP, corresponding to the operation conditions ($T_{\text{oa}}$ and $T_{\text{w}}$) in each bin. These data are given in the manufacturers’ data-books. When the HP is operating at part load, namely when $T_{\text{oa}}>T_{\text{biv}}$, then the COP at part load operation ($\text{COP}_{\text{bin}}$) must be determined, by considering the necessary correction due to the on-off cycling of the HP. The $\text{COP}_{\text{bin}}$ of the fixed-capacity AWHPs is calculated as [9]:

$$\text{COP}_{\text{bin}} = \frac{\text{COP}_d \cdot \frac{CR}{C_d \cdot CR + (1 - C_d)}}{\text{P}_{\text{hp}}(i)} \cdot \left(1 - C_d\right), \quad (6)$$

where COP$_d$ is the COP corresponding to the declared capacity $P_{\text{hp}}(i)$ at each bin at part load operation conditions, C$_d$ is the degradation coefficient, and CR is the capacity ratio. If the C$_d$ is not determined by the manufacturer, then the default value of C$_d$ is 0.9 [9]. The procedure for the COP$_{\text{bin}}$ calculation of the multi-compressor and inverter-driven AWHPs is similar and is described in [9]. The CR can be estimated as follows:

$$\text{CR} = \frac{P_{\text{el}}(i)}{P_{\text{hp}}(i)} \cdot \left(1 - C_d\right). \quad (7)$$

At part load operation, the CR<1 while at full load operation CR=1.

The electricity input power of the unit at each bin $P_{\text{hp}}(i)$ is calculated as:

$$P_{\text{el}}(i) = \frac{P_{\text{hp}}(i)}{\text{COP}_{\text{bin}}(i)} \cdot \left(1 - C_d\right). \quad (8)$$

The electricity consumption $E_{\text{hp}}(i)$ of the HP and the electricity consumption of the back-up heater $E_{\text{bu}}(i)$ in each temperature bin are determined by the following equations:

$$E_{\text{hp}}(i) = P_{\text{el}}(i) \cdot t_{\text{bin}}(i) \quad (9)$$
\[ E_{bu}(i) = P_{bu}(i) \cdot t_{bu}(i), \]  
\[ \text{(10)} \]

where \( P_{bu}(i) \) is the electrical power of the back-up heater.

When performing the calculations at all bins and summing up all energy quantities, the active mode seasonal coefficient of performance (SCOP\(_{on}\)) is calculated according to the equation:

\[ SCOP_{on} = \frac{Q_{on}}{E_{HP} + E_{bu}} \]  
\[ \text{(11)} \]

where \( Q_{on}, E_{HP} \) and \( E_{bu} \) are the sums of \( Q_{on}(i), E_{HP}(i) \) and \( E_{bu}(i) \) at all bins.

5. Air-to-water heat pumps characteristics and calculation case studies

Table 1 shows the technical data of the fixed-capacity AWHP (FCHP) and table 2 the technical data of the inverter-driven AWHP (IDHP) examined in this work, as given by the manufacturer. The two AWHPs have almost similar heating capacity at rated conditions.

| Table 1. Heating capacity in kW and COP of a FCHP. |
|-----------------------------------------------|
| Heating Capacity in [kW] and COP values (in brackets) |
| \( T_{oa} \) [°C] | \( T_w = 35°C \) | \( T_w = 45°C \) |
|-----------------|-----------------|-----------------|
| TOL             | 4.32 (2.60)     | 4.05 (1.89)     |
| -7              | 4.56 (2.91)     | 4.48 (2.26)     |
| 2               | 5.47 (3.65)     | 5.33 (2.84)     |
| 7               | 6.30 (4.23)     | 6.13 (3.30)     |
| 12              | 7.13 (4.81)     | 6.93 (3.76)     |

| Table 2. Heating capacity in kW and COP of an IDHP. |
|-----------------------------------------------|
| Heating Capacity in [kW] and COP values (in brackets) |
| \( T_{oa} \) [°C] | \( T_w = 35°C \) | \( T_w = 45°C \) |
|-----------------|-----------------|-----------------|
| TOL             | 0.61 (2.40)     | 0.53 (2.00)     |
| -7              | 0.85 (2.70)     | 0.74 (2.29)     |
| 2               | 1.15 (3.27)     | 0.96 (2.66)     |
| 7               | 1.46 (4.18)     | 1.28 (3.60)     |
| 12              | 1.67 (4.82)     | 1.46 (4.11)     |

The energy analysis for evaluating the SCOP\(_{on}\) of the AWHPs was conducted according to [9], by applying the calculation procedure described in the previous section, with the following assumptions: the building’s maximum heat load (design load \( P_{des} \)) was matched exactly with the maximum heating capacity of the AWHP at the \( T_{des} \) (red line in figure 2).

The variables of this analysis are: a) the climate at the building location (Ierapetra, Athens, Thessaloniki and Florina), b) the water temperature produced by the HP (35°C and 45°C) and c) the water outlet temperature variance, namely either fixed-water temperature (FWT) or variable water temperature (VWT) (with or without weather compensation respectively).
Additionally, SCOP\(_{\text{on}}\) values were calculated for each AWHP by increasing and decreasing the building thermal load, namely by oversizing or selecting a HP with a declared capacity at \(T_{\text{des}}\) lower than the \(P_{\text{des}}\) (down-sized) respectively. Table 3 presents the nine cases of the SCOP\(_{\text{on}}\) calculations in this work, for each of the four selected locations. In one case the building heating load (\(P_h\)) equals to the HP heating capacity at the outdoor design temperature (\(T_{\text{des}}\)), while in the other eight cases the HP is either oversized or has a heating capacity lower than the design load.

**Table 3. SCOP\(_{\text{on}}\) calculation cases in accordance with building design load.**

| HP heating capacity | HP heating capacity | HP heating capacity |
|---------------------|---------------------|---------------------|
| larger than \(P_{\text{des}}\) | equal to \(P_{\text{des}}\) | lower than \(P_{\text{des}}\) |
| \(P_{\text{HP}}\) | \(P_h\) \([\text{kW}]\) | \(P_{\text{HP}}\) | \(P_h\) \([\text{kW}]\) | \(P_{\text{HP}}\) | \(P_h\) \([\text{kW}]\) |
| 140% \(P_{\text{des}}\) | 3.91 | 100% \(P_{\text{des}}\) | 5.47 | 90% \(P_{\text{des}}\) | 6.08 |
| 130% \(P_{\text{des}}\) | 4.21 | 80% \(P_{\text{des}}\) | 6.84 |
| 120% \(P_{\text{des}}\) | 4.56 | 70% \(P_{\text{des}}\) | 7.81 |
| 110% \(P_{\text{des}}\) | 4.97 | 60% \(P_{\text{des}}\) | 9.12 |

In figure 3 the building thermal load of three case studies (green, red and yellow lines) and the HP’s heating capacity characteristic curve (blue line) are shown. The red line represents a building heating load \(P_{\text{des}}\), which at the \(T_{\text{des}}=2^\circ\text{C}\) is equal to the HP’s declared heating capacity \((P_{\text{des}} = P_{\text{hp}} = 5.5\ \text{kW})\). The green and yellow line represent a building heating load, which at the same \(T_{\text{des}}=2^\circ\text{C}\) corresponds to a \(P_{\text{des}}\) of 6.8 kW and 4.5 kW respectively. In other words, these two lines represent the cases where the HP is over-sized or down-sized respectively.

**Figure 3. Building heating load for 3 case studies and HP’s heating capacity curve.**

6. Results and discussion

In order to examine how the various parameters influence the performance of the AWHPs, the results are analyzed and plotted in diagrams. In figure 4, the SCOP\(_{\text{on}}\) values of the two examined AWHPs are illustrated for the four cities (Ierapetra, Athens, Thessaloniki and Florina). These values were estimated by matching the heating capacity of the AWHPs in each location equal to the building’s maximum heat load (\(P_{\text{des}}\)), either with 35°C or 45°C water outlet temperature to the heating system. In each case study, the operation with weather compensation (VWT) or without weather compensation (FWT) was also considered. Additional SCOP\(_{\text{on}}\) calculations were conducted in Ierapetra for a \(T_{\text{des}}=4^\circ\text{C}\) (lower than the value 7°C given in the Greek regulations [20]), because in this city it is observed an unusually high occurrence of the outdoor temperature in [h] below \(T_{\text{des}}=7^\circ\text{C}\), in relation with the other three cities.
Figure 4. SCOP\textsubscript{on} values a) Fixed-capacity AWHP (FCHP) b) Inverter-driven AWHP (IDHP).

From the results shown in figure 4, it is obvious that the climate affects notably the seasonal performance. The SCOP\textsubscript{on} values are increasing progressively from the northern locations (city of Florina, at climate Zone D) to the southern ones (city of Ierapetra with T\textsubscript{des}=4°C, at climate Zone A). The only exception appears in the case of the city of Ierapetra with T\textsubscript{des}=7°C, where the occurrence of the outdoor temperature in [h] below 7°C is high, as it was mentioned previously. Since the auxiliary electrical resistances are turned on under the T\textsubscript{des}, their operation time is extended, more electrical energy is consumed, and this fact leads to lower SCOP\textsubscript{on} values. In this case the SCOP\textsubscript{on} value in Ierapetra is lower than this of Athens, even though the heating season of Ierapetra is warmer and a higher SCOP\textsubscript{on} would be expected. In the following, the T\textsubscript{des}=4°C will be considered for Ierapetra.

The results also show that: a) higher SCOP\textsubscript{on} values are observed with the IDHP than the FCHP, which is in a good agreement with the conclusions of Naldi et al. [7], [23], b) the operation with a water output temperature of 35°C instead of 45°C increases the SCOP\textsubscript{on} values significantly, and c) the seasonal performance is increased with the use of weather compensation, namely with VWT instead of FWT.

Table 4 presents the increase of SCOP\textsubscript{on} values in the case of FCHP and IDHP respectively, resulting by shifting from FWT to VWT and from 45°C to 35°C water output temperature. More specifically, with weather compensation the FCHP’s performance is increased by 6÷11% (0.20 to 0.35 in value) and by 12÷24% (0.34 to 0.59 in value) for 35°C and 45°C water outlet temperature respectively, while the IDHP’s performance is increased by 3÷7% (0.12 to 0.25 value) and 7÷15% (0.22 to 0.46 in value) respectively.

| Location | Shift from FWT to VWT | Shift from 45°C to 35°C |
|----------|----------------------|------------------------|
|          | 35°C | 45°C | FWT | VWT | 35°C | 45°C | FWT | VWT |
| Ierapetra (7°C) | 6% (3%) | 12% (7%) | 23% (21%) | 16% (17%) | 12% (7%) | 23% (21%) | 16% (17%) |
| Ierapetra (4°C) | 10% (6%) | 22% (13%) | 28% (20%) | 15% (13%) | 22% (14%) | 28% (20%) | 15% (12%) |
| Athens | 11% (6%) | 22% (14%) | 28% (20%) | 15% (12%) | 22% (14%) | 28% (20%) | 15% (12%) |
| Thessaloniki | 11% (7%) | 24% (15%) | 28% (20%) | 15% (12%) | 24% (15%) | 28% (20%) | 15% (12%) |
| Florina | 10% (7%) | 22% (14%) | 28% (20%) | 16% (13%) | 22% (14%) | 28% (20%) | 16% (13%) |

Additionally, the lowering of the supply water temperature of a HP without compensation (FWT) from 45°C to 35°C results an increase in the estimated SCOP\textsubscript{on} values of 23÷28% (0.64 to 0.73 in value) and 20÷21% (0.55 to 0.69 in value) for the FCHP and IDHP respectively, while the
corresponding SCOP\textscript{o}n increase for HP with compensation (VWT) is 15\%-16\% (0.45 to 0.50 in value) and 12\%-17\% (0.39 to 0.54 in value) for the FCHP and IDHP respectively. It is important to note that by shifting from FWT to VWT combined with a decrease of the water output temperature from 45°C to 35°C, results to a SCOP\textscript{o}n increase of 40\%-43\% (0.95 to 1.07 in value) and 27\%-28\% (0.76 to 0.93 in value) for FCHP and IDHP respectively.

In order to highlight the influence of the AWHPs sizing on the seasonal performance, additional calculations were performed by varying the bivalent temperature $T_{biv}$, the water output temperature $T_w$, and the type of the AWHP considered (FCHP and IDHP), and by considering weather compensation control or not (VWT or FWT). The resulted SCOP\textscript{o}n values are plotted on diagrams in figure 5. A part of the results, which show more clearly how the correct sizing of an AWHP is related to the $T_{biv}$, are presented in table 5.

The diagrams in figure 5 and the results given in table, show that in the case of inverted driven heat pump (IDHP), in all locations, the higher SCOP\textscript{o}n values are obtained by adopting a $T_{biv}$ equal to the design temperature $T_{des}$, which was also concluded in [7]. This is easily observed on figure 5 where the red and yellow SCOP\textscript{o}n curves have a peak at $T_{des}$. On the contrary, in the case of fixed-capacity heat pump (FCHP) the higher SCOP\textscript{o}n values appear when sizing at a $T_{biv}$ higher by 1\%-2\% than the $T_{des}$, which was also observed by Claudia Naldi et al. [7] and Matteo Dongellini et al. [10]. The corresponding green and blue SCOP\textscript{o}n curves are increasing as going from temperatures lower than $T_{biv}$ to higher temperatures, and show a peak at a temperature 1\%-2\% higher than $T_{biv}$.

More specifically, the results of table 5 highlight in the case of FCHP, a $T_{biv}$ increase by 2°C results a SCOP\textscript{o}n increase of 1\%-2\%, when $T_w=35°C$, and an increase of 2\%-4\% when $T_w=45°C$. Oppositely, the sizing of IDHP with a $T_{biv}$ higher than $T_{des}$ by 2°C causes a reduction of the SCOP\textscript{o}n by 1\%-6\% for all supply water temperatures.

|          | $T_{biv}$ (°C) | $T_w = 35°C$ | $T_w = 45°C$ |
|----------|---------------|--------------|--------------|
|          |               | FCHP         | IDHP         | FCHP         | IDHP         |
|          |               | FWT VWT      | FWT VWT      | FWT VWT      | FWT VWT      |
| Ierapetra (T\text{\scriptscriptstyle des}=4°C) | 4              | 3.37 3.70    | 4.10 4.34    | 2.70 3.27    | 3.35 3.77    |
|          | 5              | 3.42 3.76    | 4.02 4.25    | 2.71 3.26    | 3.21 3.64    |
|          | 6              | 3.40 3.72    | 3.87 4.11    | 2.71 3.26    | 3.21 3.64    |
| Athens   (T\text{\scriptscriptstyle des}=2°C)  | 2              | 3.31 3.66    | 4.03 4.28    | 2.59 3.17    | 3.35 3.81    |
|          | 3              | 3.37 3.72    | 4.00 4.25    | 2.65 3.23    | 3.32 3.77    |
|          | 4              | 3.39 3.73    | 3.89 4.13    | 2.68 3.26    | 3.22 3.68    |
| Thessaloniki (T\text{\scriptscriptstyle des}=-2°C) | -2             | 3.17 3.52    | 3.77 4.02    | 2.47 3.06    | 3.13 3.59    |
|          | -1             | 3.22 3.57    | 3.76 4.01    | 2.52 3.11    | 3.12 3.59    |
|          | 0              | 3.27 3.63    | 3.73 3.98    | 2.57 3.16    | 3.10 3.56    |
| Florina  (T\text{\scriptscriptstyle des}=-7°C) | -7             | 2.96 3.27    | 3.24 3.46    | 2.32 2.82    | 2.70 3.07    |
|          | -6             | 2.99 3.30    | 3.23 3.44    | 2.34 2.85    | 2.68 3.06    |
|          | -5             | 3.01 3.32    | 3.20 3.41    | 2.37 2.87    | 2.65 3.03    |

From the results given in figure 5 and table 5 it can be deduced that the maximum SCOP\textscript{o}n values for a FCHP appear when its declared heating capacity is lower than the design load of the building, namely when the HP is down-sized. On the other side, the best performance of an IDHP is obtained when its declared capacity matches exactly the design heat load of the building, therefore when the $T_{biv}$ is equal to the $T_{des}$. Furthermore, in hot climates as in the case of Ierapetra, sometimes it is better to size the HP with a $T_{biv}$ lower that the $T_{des}$ (see figure 4), when a high occurrence of the outdoor temperature in [h] is observed.
Figure 5. SCOP$_{on}$ values and bin distribution for: 1) Ierapetra ($T_{des}=4^\circ C$) (a) $T_w = 35^\circ C$ (b) $T_w = 45^\circ C$, 2) Athens (c) $T_w = 35^\circ C$ (d) $T_w = 45^\circ C$, 3) Thessaloniki (e) $T_w = 35^\circ C$ (f) $T_w = 45^\circ C$ and 4) Florina (g) $T_w = 35^\circ C$ (h) $T_w = 45^\circ C$. 
7. Conclusions
In this work the behaviour of the active mode seasonal coefficient of performance, SCOP\textsubscript{on}, of two electric-driven air-to-water heat pumps (AWHPs) with electric resistances as back-up heater, has been analysed in order to examine the impact of various parameters on the efficiency during the heating season. The AWHPs are considered to be installed in representative cities of the four Greek climate zones and the calculations were conducted by means of the bin method, according to European standard EN14825, using the local climatic data of the four cities. The variables of the analysis were the compressor technology (fixed-capacity and inverter-driven), the supply water temperature to the heat emitters (35°C and 45°C) and the control system (operation with or without weather compensation).

The obtained numerical results show that the climate affects the seasonal performance which is increasing progressively from the northern locations to the southern ones. The results also confirm that the use of inverter-driven heat pump (IDHP), in all case studies, achieves higher SCOP\textsubscript{on} values with respect to fixed-capacity heat pump (FCHP). The increase in performance by adopting the inverter technology can reach up to 30%. It is also highlighted that the seasonal performance rises by lowering the supply water temperature to the heat emitters and by adopting a weather compensation control system. The reduction of the output water temperature from 45°C to 35°C combined with a shift from fixed-water temperature (FWT) to variable water temperature (VWT), results a SCOP\textsubscript{on} increase of 40-43% and 27-28%, for FCHP and IDHP respectively. In addition, it is demonstrated that the correct sizing of an AWHP can increase the seasonal performance. More specifically, the IDHP shows optimal performance when its declared heating capacity (P\textsubscript{des}) at the design outdoor air temperature (T\textsubscript{des}) is equal to the building design heat load, namely when the bivalent temperature T\textsubscript{biv} is equal to T\textsubscript{des}. On the other hand, the maximum SCOP\textsubscript{on} values of the FCHP appear when its declared heating capacity is lower than the building design load, that is to say when the T\textsubscript{biv} is larger than the T\textsubscript{des} and the heat pump is down-sized.

Summing, it can be concluded that HP’s performance is mostly driven by many factors such as: the climate of the location, the HP’s compressor technology, the supply water temperature, the control system and the sizing of the HP. The results of this work confirm that when sizing an AWHP in a heating system and designing its operating parameters, a thorough consideration is needed in order to have maximum seasonal performance.

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