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Life cycle cost analysis of different residential heat pump systems

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Abstract. Nowadays, the application of air-source heat pumps for heating and cooling in residential buildings has been increased significantly. The main occasion for this is the accessibility of a heat source for these devices - the external air. Nevertheless, the increase of the energy efficiency of the air source heat pump systems is a difficult design problem because their capacity and performance are a function of the dynamically changing parameters of the outdoor air. Because of that, the main aim of this study is to develop an approach for choosing a structural scheme of an air-to-water heat pump system under specific climatic conditions. The considered systems are monovalent, bivalent-parallel and bivalent-alternative heat pump systems. In the current paper is conducted a dynamic energy modeling of heating and cooling demand of a typical residential building situated in Varna, Bulgaria and applying the bin temperature data. It is assessed the effect of the heat pump capacity over the annual and seasonal energy performance of the heat pump systems. It is established the effect of the bivalent temperature, cut-off temperature and on-off cycles duration on rates of the criteria for techno-economic assessment. The seasonal coefficient of performance (SCOP), seasonal energy efficiency rate (SEER) and life cycle costs (LCC) of the analyzed heat pump systems are adopted as assessment parameters.

1 Introduction

The close inspection of the heat pump market in Europe shows that there are about 33 million aerothermal heat pumps in exploitation in 2017 [1]. Almost 1% of all installed heat pumps are operated in Bulgaria. This puts the country in 10th place among EU28 countries in the total number of aerothermal heat pumps in operation. Therefore, besides the focus to grow the energy performance of the building envelope, another central issue is improving the energy performance of the building heating system using an aerothermal heat pump. It is necessary for an in-depth analysis of opportunities for the utilization of the full energy-saving potential of air source heat pump systems. This is a difficult design problem because air-to-water heat pump (AWHP) capacity and performance are a function of the dynamically changing parameters of the external air.

A part of studies in reference literature focuses on a specifically related problem. Dongellini et al. [2] consider the impact of the on-off cycle losses on the seasonal energy

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performances of three reversible AWHP through dynamic simulation of the developed models. In another paper [3], the same author asses the influence of defrosting energy losses on the value of the seasonal coefficient of performance. It is proved that AWHP seasonal efficiency decrease by about 5% taking into account defrost energy losses. Thus, the energy losses due to the defrost cycles do not significantly affect AWHP seasonal efficiency [3]. Huchtemann et al. [4] observe the influence of the mean supply water temperature control on the AWHP efficiency. The results demonstrate that a heating curve decreased by 1 K leads to savings in primary energy of 2.0% consumed by AWHP.

The author's team of Dongellini et al. provides solutions for many problems in the field of determination of the energy performance parameters of AWHP and formulate the rules for the optimal design of air to water heat pump systems [5-10]. For example, in [5, 6, 7] are reported the results obtained from the calculation of seasonal and yearly performance parameters of heat pump systems. It is asses the influence of Italian climate conditions on the annual efficiency of the system. In a study investigating the effect of the external air temperature and relative humidity on the coefficient of performance of the aerothermal heat pumps, Vocale et al. [8] reported that the monthly average of the COP can be reduced to 17% when the relative humidity is higher than 80% and the external air temperature is in the diapason 0-6°C over a long period of time during the heating season. It is indicated that the reason for this is energy losses caused by defrosting cycles of AWHP. In another paper [9] are presented the strategy for the optimal design of the optimal sizing rules for different types of AWHP. These rules are formulated using the optimal value of a technical parameter: an annual performance factor calculated as a function of the optimal value of climate, building thermal loads, and heat pump characteristics. In a follow-up study, Dongellini et al. [10] evaluated the potential energy savings by different types of structural schemes heat pump systems: bivalent-parallel and bivalent-alternative. It was analyzed the impact of the cut-off and bivalent temperatures only on the seasonal coefficient of performance (SCOP) of the AWHP.

In all the studies reviewed here, however, the problem related to the optimal design rules for heat pump systems can be defined as incomplete. Together these studies do not provide important insights into the economic aspects of operating modes of AWHP. Because of that, the goal of this research is to develop an approach for choosing a structural scheme of an air-to-water heat pump system under specific climatic conditions. The considered systems are monovalent, bivalent-parallel and bivalent-alternative heat pump systems. It is assessed the influence of the heat pump capacity over the seasonal performance of the heat pump systems. The analyzed heat pump systems are assessed using preliminary criteria for techno-economic assessment such as seasonal coefficient of performance (SCOP), seasonal energy efficiency rate (SEER) and life cycle costs (LCC) of the considered heat pump systems. In the current paper is conducted a dynamic energy modeling of heating and cooling demand of a typical residential building located in Varna, Bulgaria and using the bin temperature data.

2 The case study

2.1 Climatic conditions

To assess the techno-economic performance of the considered structural schemes of an air-to-water heat pump systems, it is requisite specific climatic conditions to be indicated. It is assumed that the considered building is placed in Varna, Bulgaria (43°13'0.01" N 27°55'0.01" E). The local external bin temperature trend is obtained by collecting and evaluating the data for fifteen years time period (from 2005 until 2019). The external air temperature data is measured by meteorological station WMO ID = 15552, located in Varna, Bulgaria (43°12'45" N, 27°57'9" E, and altitude 43m). For measurement data processing is used as a method detailed described by authors in [13].

The obtained bin temperature distribution of the considered location is presented in Figure 1. Figure 1 highlights that on the annual basis, the bin temperature with the highest frequency is 8°C characterized by 4.1% frequency, 9°C (4.0 %), 10°C (3.9%), 7°C (3.8%), 6°C (3.7%) and 20°C (3.7%).

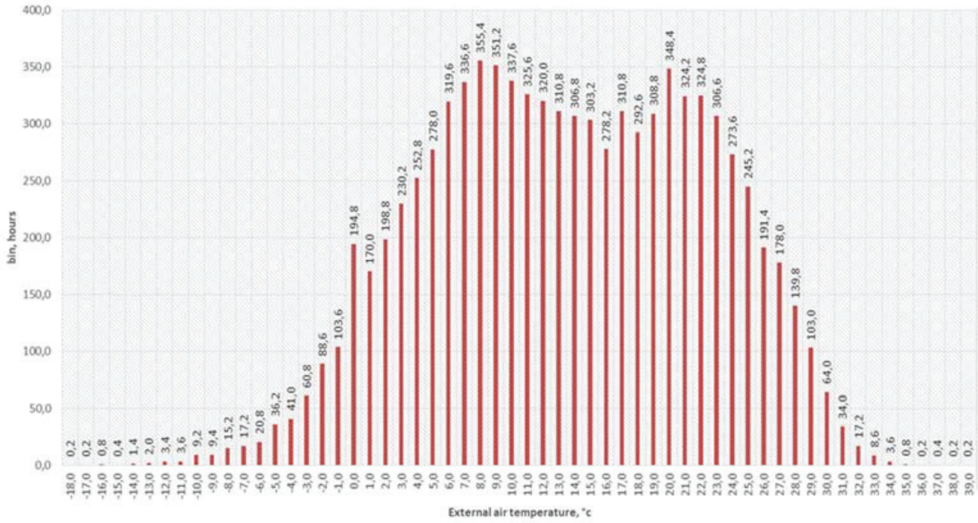

Fig. 1. Bin temperature trend for Varna, Bulgaria

Based on the bin temperature trend analysis, the outdoor temperature frequency curve is defined – Figure 2. It is concluded that a building without significant internal and external heat gains is described by a heating period of 4702 h per year, while the cooling period has a duration of 1260 h per year. It is accepted that the building has a presence of heating and cooling loads, only if the external air temperature is below 15°C. The cooling load is observed when the outside air temperature is above 23°C.

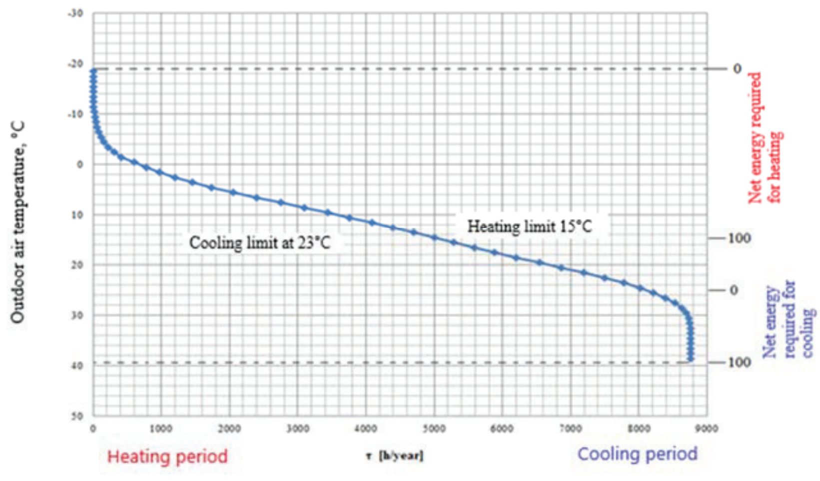

Fig. 2. Temperature frequency curve - Varna, Bulgaria

2.2 Heating and cooling loads determination

The current investigation has been conducted on a two-family residential building located in Varna, Bulgaria. The geometric specification of the building issummarized in Table 1. In the current model of the residential building is accepted that the walls, roof, and floors are insulated and triple glazing for the windows is used. Thus, the elements of the building envelope meet the requirements of the Bulgarian Energy Efficiency Standard [11].

Table 1. Geometric specification of the considered residential building**.**

Parameter	Value	Parameter	Value
The total area of the heatedand/or cooled spaces	179.2 m^2	Area of transparent building elements	36.15 m^2
The volume of the heated and/or cooled spaces	510 m^3	Windows/walls area ratio South North East West	17.54% 57.94% 68.24% 0% 0%
Area of opaque building elements (walls)	206.16 m^2	Total floor area Area of the floor on the ground Area of the floor over the unheated basement	72.35 17.65
Area of the roof	90 m^2		

The value of the thermal transmittance coefficients of the elements of the building envelop are listed in Table 2.

Modeling of the heating and cooling demands of the considered residential building is conducted with the following information:

- The building is located in a residential neighborhood with a predominant building construction with low height (10 m mean height of the surroundings);
- It is assumed that the transmission heat loss coefficient from heated space to the exterior through the unheated space and transmission heat loss coefficient from heated space to a neighboring heated space heated at significantly different temperature is equal to zero, i.e. $H_U = H_A = 0$;
- The modeling of the energy demand of the considered building is conducted assuming that ventilation systems are absent, i.e. the building is naturally ventilated. It is assumed that the supplied air due to infiltration has the thermal characteristics of external air. The average hourly air exchange rate is set as 0.5 h^{-1};

Table 2. The thermal transmittance coefficients and corresponding insulation thickness of considered building elements

Thermal transmittance coeficients, U_i, [W/m^{2}K]				
Walls	Roof	Ground floor	The floor over the unheated basement	Windows
0.2415	0.242	0.2256	0.283	1.406
Insulation thickness, δ_{ins}, [m]				
0.12	0.12	0.09	0.06	-

- Thetotal number of people living in the building is 8;

- The internal heat gains from people, appliances, and lighting were defined to be Φ_{int} = 2.23 kW;
- Internal design temperature is set as $\theta_{int,winter}$ = 20°C in winter and $\theta_{int,summer}$ = 25°C in summer, respectively. Building spaces such as bathrooms are considered as uncooled space.

The calculation of the building energy needs for space cooling and heating are conducted according to [11 and 12]. The basis of this method is the following equation of the required energy for heating for each month of the heating period:

$$Q_{H,nd} = Q_{H,ht} - \eta_{H,gn} \cdot Q_{H,gn} \tag{1}$$

and the required energy for cooling for each month of the cooling period:

$$Q_{C,nd} = Q_{C,gn} - \eta_{C,gn} \cdot Q_{C,ht} \tag{2}$$

where:
$Q_{H,ht}$ and $Q_{C,ht}$ - the total heat loss for the month of the heating and cooling period, respectively;
$Q_{H,gn}$ and $Q_{C,gn}$ - the total heat gains for the month of the heating and cooling period, respectively;
$\eta_{H,gn}$ and $\eta_{C,gn}$ - dimensionless factor for the utilization of the monthly heat losses and the monthly heat gains, respectively.

2.3 Building energy signature

As a result of the used methodology presented in [11 and 12], the monthly net heating and cooling energy required by the building ($Q_{H,nd}$ and $Q_{C,nd}$) has been determined. Moreover, using the calculated results for the $Q_{H,nd}$ and $Q_{C,nd}$, the building energy signature (BES) is defined.

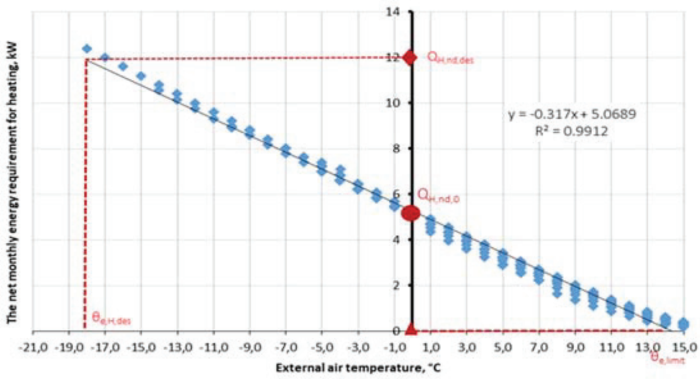

Fig. 3. Building energy signature for heating period

In the current investigation, the BES represents the linear interpolation of the data for the $Q_{H,nd}$ and $Q_{C,nd}$. The values for the monthly net energy required by the building for the heating and cooling are presented depending on the winter and summer outdoor air temperature. The BES obtained for the considered residential building located in Varna is presented in Figures 3 and 4. It is apparent from these figures that there are different values for $Q_{H,nd}$ and $Q_{C,nd}$ at the same external air temperature. The reason for this is the different degree of utilization of heat gains (heat losses) during the months of the heating and cooling period, respectively.

The remarkable points shown in Figure 3 and Figure 4 are:

- The building design peak loads, $Q_{H,nd,des}$ or $Q_{C,nd,des}$, which is the net energy requirement for heating or cooling in accordance of the external minimal design temperature $\theta_{des,h}$ = -18°C ($\theta_{des,c}$ = 39°C);

- As can be seen from Fig. 3, when the external air temperature is equal to the $\theta_{e,limit}$, the building net energy required for the heating is zero. At the same time, the building net energy required for the cooling is minimal;
- The net energy for the heating required by the building at 0°C, $Q_{H,nd,0}$ and the cooling load required by the building at 26°C is $Q_{C,nd,26}$. As can be seen from Fig. 3 and 4, $Q_{H,nd,0}$ = 5.0689 kW and $Q_{C,nd,26}$ = 4.5815 kW, respectively.

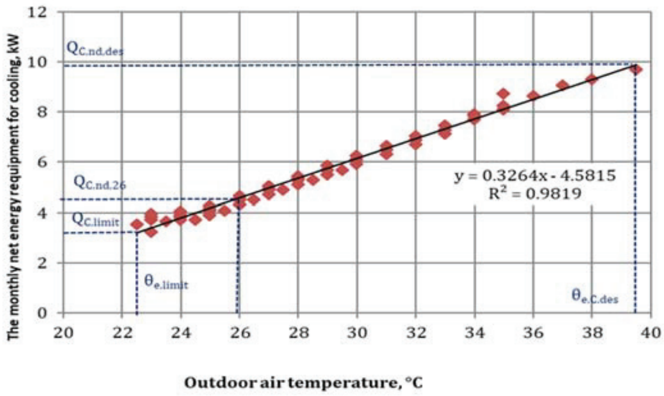

Fig. 4. Building energy signature for cooling season

As a result of the construction of BES, the function between the required building thermal load, $Q_{H,nd}$, and the external air temperature, θ_e, can be described by the following expression

$$Q_{H,nd} = -S_H \cdot \theta_e + Q_{H,nd,0}, \tag{3}$$

where the slope of the BES for heating period, S_H, is as follow:

$$S_H = \frac{Q_{H,nd,0} - Q_{H,nd,\lim it}}{\theta_{e,\lim it}} \tag{4}$$

During the cooling season, the relationship between the required building thermal load, $Q_{C,nd}$, and the outdoor air temperature, θ_e, is:

$$Q_{C,nd} = S_C \cdot \theta_e + Q_{H,nd,26}, \tag{5}$$

where the slope of the BES for cooling period is:

$$S_C = \frac{Q_{H,nd,26} - Q_{C,nd,\lim it}}{\theta_{e,\lim it}} \tag{6}$$

From eq. (4) and (6) is clear that the slope of the BES expresses the sensitivity of the net thermal load required by the building with respect to the outdoor air temperature.

As can be seen from Fig. 3, the slope of the BES for the heating period is S_H = -0.317 and the base thermal load is $Q_{H,nd,0}$ = 5.0689 kW. Fig. 4 demonstrates that S_C = 0.3264 and $Q_{C,nd,26}$ = 4.5815 kW.

Another result of the definition of the BES for the heating and cooling period is the possibility to define an indicator of the presence of seasonal unbalanced loads. This parameter can be calculated as following [9]:

$$\mathrm{UI} = \frac{Q_{C,\mathrm{nd,des}}}{Q_{H,\mathrm{nd,des}}} \tag{7}$$

In the current paper UI = 0.76 and, therefore, there are no significant seasonal unbalanced loads of the considered building.

2.4 Descriptionof the considered structural schemeof air-to-water heat pump systems

The considered heat pumps coupled to the residential building are reversible inverter-driven units. The back-up system has been perceived as an electric boiler operating during the winter season. Air-to-water heat pump and back-up heater form the structural scheme of the system, and in the current investigation, the following structural schemes have been analyzed: monovalent, bivalent-parallel and bivalent-alternative heat pump systems. For each structural scheme have been selected different units with different nominal thermal capacity.

2.4.1 Monovalent heat pump system

In the case of the monovalent heat pump system, the required energy for heating of the building is met entirely by the AWHP unit. Therefore, the criteria for the choice of the AWHP size are made on the basis of the peak load of the building in the winter and summer season. According to eq. (3) and (5) the peak loads of the building at $\theta_{e,H,min} =$ -18˚C and $\theta_{e,C,max} = 39$˚C are as follow:

$$Q_{H,nd} = -0.317 \cdot \theta_{e,H,\min} + 5.0689 = 10.775 kW \quad (8)$$

$$Q_{C,nd} = 0.3264 \cdot \theta_{e,C,\max} - 4.5815 = 8.148 kW \quad (9)$$

A graphical representation of the heating and cooling capacity of the considered size AWHP unit as a function of the outdoor temperature is presented in Fig. 5*a* and 5*b*, respectively.

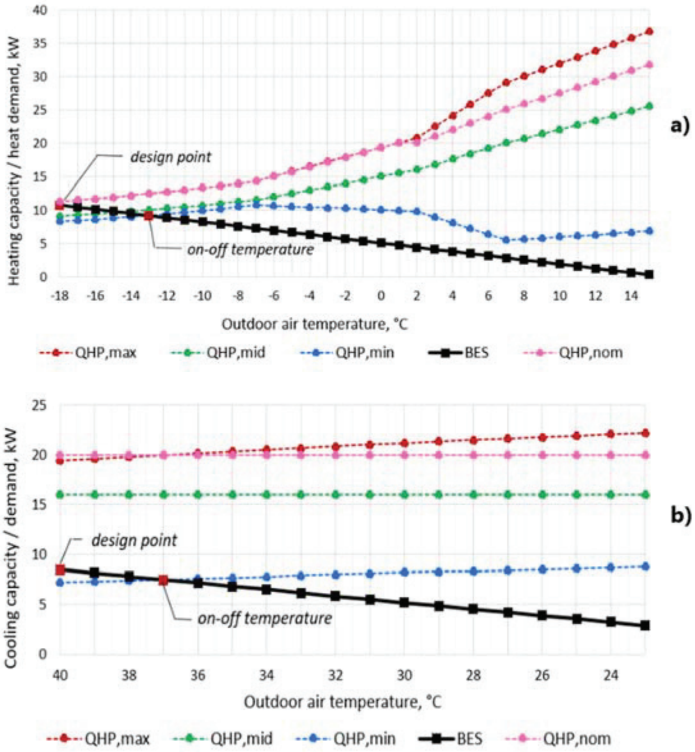

Fig. 5. Heating (*a)* and cooling capacity (*b*) of the selected size AWHP unit as a function of the external air temperature (case HP_0)

Closer inspection of Fig. 5*a* shows that the selected unit will be operated at close to nominal capacity only at certain hours of the heating season. This is the consequence of the enforced design criteria of the monovalent heat pump system: the AWHP unit size is determined in order to cover the peak heating and cooling load. Moreover, the heat pump will operate at partial load only 5 hours, i.e. when the external temperature is in the range from -18.5°C to -13.5°C. In the remaining hours of the heating season (4697 hours), the heat pump will perform on-off cycles.

Fig. 5*b* presents that during the cooling season, the AWHP will not operate at nominal inverter frequency. The peak cooling load of the building is covered when the AWHP work at about 42% of the maximum capacity of the heat pump. For a long time period during the cooling season (1527.6 h), the AWHP will perform on-off cycles.

2.4.2 Bivalent-parallel heat pump system

In order to prevent the described above disadvantages in the operation of the monovalent heat pump system, a bivalent - parallel structural scheme has been adopted. The size of the heat pump is chosen so as to match only part of the net heating load required by the building, but at the same time entirely to satisfy the peak cooling load.

It is considered two variants of the bivalent-parallel heat pump system:

• Case study HP_1: the heat pump match the peak cooling load of the building during the entire cooling period (except for 0.8 hours, in which the outside air temperature is in the range θ_e = 37.5 ÷ 39.5°C) and 50% of the net heating load required by the building at minimal outside air temperature ($\theta_{e,min}$ = -19°C). The heat pump produces water with a temperature of 45°C / 40°C in heating mode and 7 °C / 12°C in cooling mode, respectively;

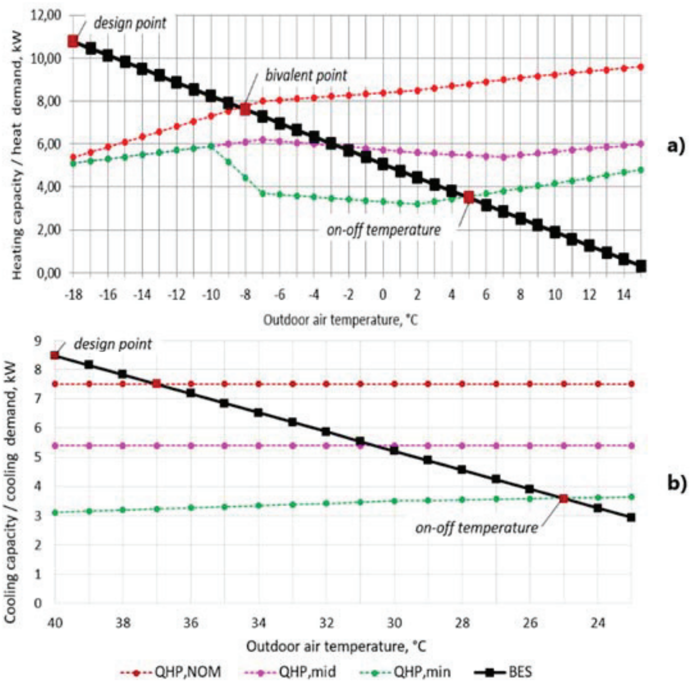

Fig. 6. Heating (*a*) and cooling capacity (*b*) of the selected size AWHP unit as a function of the external air temperature (case study HP_1)

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• Case study HP_2: the heat pump provides 117% of the peak cooling load of the building (i.e. the net cooling load required by the building at $\theta_{e,max}$ = 39°C) at the maximum inverter frequency and 62% of the heating load under similar operating conditions.

In case of bivalent - parallel structural schemes, the dependence of the heating and cooling capacity of the selected AWHP units of the external air temperature are show in Fig. 6 and 7, respectively.

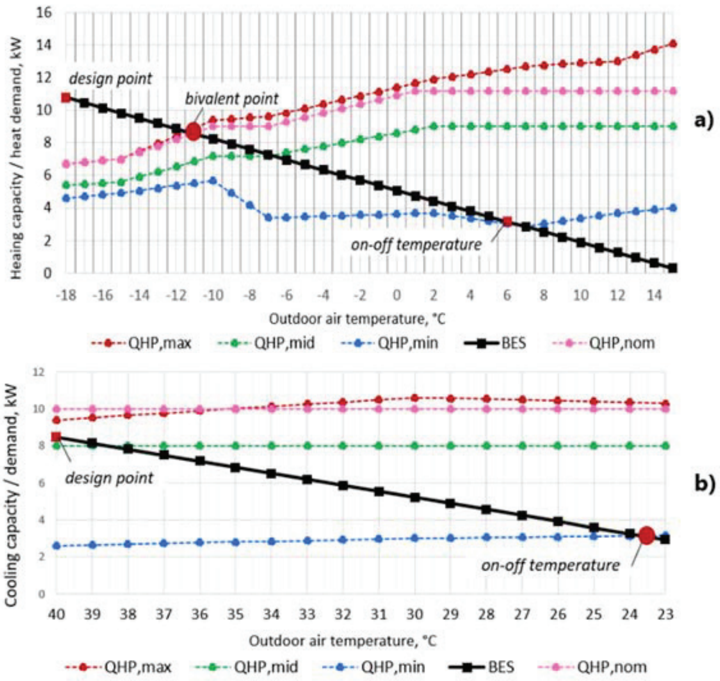

Fig. 7. Heating (*a*) and cooling capacity (*b*) of the selected size AWHP unit as a function of the external air temperature (case study HP_2)

Taken together, the results present in Fig, 6 and 7 suggest that the bivalent temperature decline with increasing the AWHP heating capacity. Furthermore, the lower bivalent temperature also leads to a lower temperature at which the heat pump realizes the on-off cycles. For example, the heat pump unit from case study HP_1 will perform on-off cycles during 3505.8 hours (or about 71%) of the heating season. This is significantly longer than the on-off cycles of the heat pump in case study HP_2. It will be operated in on-off mode 59 %of the duration of the heating season.

Moreover, the AWHP unit in case study HP_2 will operate about 80% of the cooling season at partial load and it will realize the on-off cycles only 293 hours. On the other hand, the heat pump unit in case study HP_1 is characterized by a longer period of operation in on-off mode - 799 hours, and during 0.4 hours of the cooling season, there will be a shortage of cooling capacity. However, hours with a lack of cooling capacity can be defined as insignificant.

2.4.3 Bivalent-alternative heat pump system

The considered standard size heat pump is suitable for the realization of a bivalent-alternative structural scheme because it is characterized by the temperature operating limit, $TOL = -15^{\circ}C$. Therefore, at external air temperatures below $\theta_e = -15^{\circ}C$, the required energy for heating can only be matched by a back-up heater. As can be seen from Fig. 8*b*, at ambient air temperatures in the range $\theta_e = 35.5^{\circ}C \div 39.5^{\circ}C$, there is a lack of cooling capacity. However, the period with a shortage of cooling capacity is negligibly small – roundabout 0.8 hours from the cooling period.

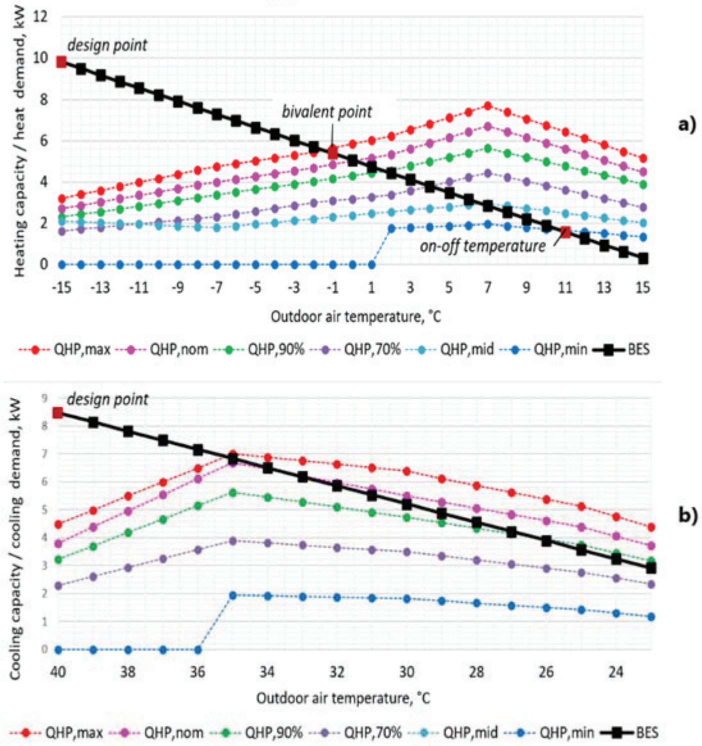

Fig. 8. Heating (*a*) and cooling capacity (*b*) of the selected size AWHP unit as a function of the external air temperature (case study HP_3)

The analysis of the bivalent - alternative heat pump system design parameters in the heating mode shows that the AWHP is characterized by the highest bivalent temperature and, accordingly, the shortest on-off cycles time period - only 1528.7 hours or about 31% of the heating season.

The design parameters of the bivalent - alternative heat pump system in the cooling mode are also more advantageous compared to the previously considered AWHP units form case studies HP_0, HP_1 and HP_2. From Fig.8*b* it is clear that during the cooling period the heat pump will be operated at partial load and no on-off cycles will be observed.

2.5 Criteria for technical assessment of the considered structural scheme of air-to-water heat pump systems

The parameters used for technical assessment the seasonal and the annual energy performance of the considered structural scheme of the AWHP systems are the Seasonal

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Coefficient of Performance (*SCOP*), the Seasonal Energy Efficiency Ratio (*SEER*), the Annual Performance Factor (*APF*) and Oversizing Parameter (*OP*).

The seasonal coefficient of performance, *SCOP* and *SEER*, are calculated using the methodical framework described in [14]. EN 14825:2019 [14] introduces a seasonal coefficient of performance, $SCOP_{net}$, expressing as follow:

$$\mathrm{SCOP_{net}} = \frac{\sum_i \dot{Q}_{\mathrm{HP},H}(i)}{\sum_i \dot{Q}_{\mathrm{HP},H,\mathrm{el}}(i)} \tag{10}$$

To determine the seasonal performance of the whole system (AWHP and back-up heater) is $SCOP_{on}$.It is defined as follows[14]:

$$\mathrm{SCOP_{on}} = \frac{\sum_i \dot{Q}_{\mathrm{HP},H}(i)+\sum_i \dot{Q}_{\mathrm{BUH},H}(i)}{\sum_i \dot{Q}_{\mathrm{HP},H,\mathrm{el}}(i)+\sum_i \dot{Q}_{\mathrm{BUH},H,\mathrm{el}}(i)}, \tag{11}$$

where: $\dot{Q}_{\mathrm{BUH},H}(i)$ is thermal energy delivered by the back-up heater and $\dot{Q}_{\mathrm{BUH},H,\mathrm{el}}(i)$ is electrical energy consumed by the back-up heater.

Since there is not a back-up system during the cooling season, in the current paper is assumed that $SEER_{on} = SEER_{net}$. The seasonal energy efficiency ratio, $SEER_{net}$, is calculated as [14]:

$$\mathrm{SEER_{net}} = \frac{\sum_i \dot{Q}_{\mathrm{HP},C}(i)}{\sum_i \dot{Q}_{\mathrm{HP},C,\mathrm{el}}(i)} \tag{12}$$

The electrical energy absorbed by the AWHP in the heating season is defined as follow [16]:

$$Q_{\mathrm{HP},H,\mathrm{el}} = \begin{cases} \dfrac{\dot{Q}_{\mathrm{T\Pi,H}}(i)[\mathrm{kWh}]}{\mathrm{COP_{eff}}(i)} & \text{if } Q_{\mathrm{nd},H}(i) < Q_{\mathrm{HP,H},\Phi_{\min}} \\ \dfrac{\dot{Q}_{\mathrm{T\Pi,H}}(i)[\mathrm{kWh}]}{\mathrm{COP}_{\Phi\mathrm{eff}}(i)} & \text{if } Q_{\mathrm{T\Pi,H},\Phi_{\min}} Q_{\mathrm{nd},H}(i) < Q_{\mathrm{HP,H},\Phi_{\max}} \\ \dfrac{\dot{Q}_{\mathrm{T\Pi,H}}(i)[\mathrm{kWh}]}{\mathrm{COP}_{\Phi\max}(i)} & \text{if } Q_{\mathrm{nd},H}(i) \geq Q_{\mathrm{HP,H},\Phi_{\max}} \end{cases}$$

(13)

In the cooling season the electrical energy consumed by the AWHP is [16]:

$$Q_{\mathrm{HP,C,el}} = \begin{cases} \dfrac{\dot{Q}_{\mathrm{T\Pi,C}}(i)[\mathrm{kWh}]}{\mathrm{EER_{eff}}(i)} & \text{if } Q_{\mathrm{nd},C}(i) < Q_{\mathrm{HP},C,\Phi_{\min}} \\ \dfrac{\dot{Q}_{\mathrm{T\Pi},C}(i)[\mathrm{kWh}]}{\mathrm{EER}_{\Phi\mathrm{eff}}(i)} & \text{if } Q_{\mathrm{HP},C,\Phi_{\min}} Q_{\mathrm{nd},C}(i) < Q_{\mathrm{HP},C,\Phi_{\max}} \\ \dfrac{\dot{Q}_{\mathrm{T\Pi},C}(i)[\mathrm{kWh}]}{\mathrm{EER}_{\Phi\max}(i)} & \text{if } Q_{\mathrm{nd,}C}(i) \geq Q_{\mathrm{HP},C,\Phi_{\max}} \end{cases}$$

(14)

The electrical energy consumed by the back-up heater is calculated as:

$$\dot{Q}_{\mathrm{BUH},H,\mathrm{el}}(i) = \frac{\dot{Q}_{\mathrm{BUH,el}}(i)}{\eta_{\mathrm{BUH}}} \tag{15}$$

where $\dot{Q}_{\mathrm{BUH,el}}(i)$ is the electrical energy consumed by the back-up system. The therm η_{BUH} is the back-up heater efficiency. In current paper η_{BUH} = 98% because it is electrical boiler.

The terms $COP_{\Phi eff}$ and $COP_{\Phi max}$ (or $EER_{\Phi eff}$ and $EER_{\Phi max}$) from equations (13) and (14) are coefficient of performance and energy efficiency rate at the partial load of the AWHP and at maximum inverter frequency, respectively. They are defined as per the information published by manufacture [15].

The evaluation of the energy losses due to heat pump on-off cycles is performed using the following correction factor [9, 14]:

$$f_{\mathrm{corr}} = \frac{\mathrm{CR}(i)}{C_c \cdot \mathrm{CR}(i) + (1 - C_c)} \tag{16}$$

where *CR(i)* is a capacity ratio. It is defined as the ratio of the building's thermal load and the AWHP thermal capacity at the same temperature conditions [16]. The term C_c in eq. (16) is the degradation coefficient in heating or cooling mode. It is assumed as $C_c = 0.9$ according to [14].

The correction factor from eq. (16) is used to obtain the effective coefficient of performance, COP_{eff}, and effective EER value, respectively. The data for COP_{eff} and EER_{eff} give information about the reduction of the energy performance of the AWHP due to the realizing of on-off cycles. They are determined as follow [14, 16]:

$$COP_{eff}(i) = COP(i) \cdot f_{corr}(i) \text{ and } EER_{eff}(i) = EER(i) \cdot f_{corr}(i) \tag{17}$$

In current investigation, the Annual Performance Factor (*APF*) and Oversizing Parameter (OP) has been used in order to assess the annual performance of the considered heat pump systems. The *APF* takes into account the energy performance of the AWHP system both the heating and cooling operating mode. It is determined as follow [16]:

$$\mathrm{APF} = \frac{\sum_i \dot{Q}_{\mathrm{HP},H}(i) + \sum_i \dot{Q}_{\mathrm{BUH},H}(i) + \sum_i \dot{Q}_{\mathrm{HP},C}(i)}{\sum_i \dot{Q}_{\mathrm{HP},H,\mathrm{el}}(i) + \sum_i \dot{Q}_{\mathrm{BUH},H,\mathrm{el}}(i) + \sum_i \dot{Q}_{\mathrm{HP},C,\mathrm{el}}(i)} \tag{18}$$

The heat pump oversizing with respect to the building design load is evaluate by using the following oversizing parameters [16]:

- In heating mode:

$$\mathrm{OP}_H = \frac{Q_{\mathrm{HP},H,\Phi\mathrm{max}} - Q_{\mathrm{nd},H,\mathrm{peak}}}{Q_{\mathrm{nd},H,\mathrm{peak}}} \tag{19}$$

- In cooling mode:

$$\mathrm{OP}_H = \frac{Q_{\mathrm{HP},C,\Phi\mathrm{max}} - Q_{\mathrm{nd},C,\mathrm{peak}}}{Q_{\mathrm{nd},C,\mathrm{peak}}} \tag{20}$$

where $Q_{\mathrm{HP},H,\Phi\mathrm{max}}$ and $Q_{\mathrm{HP},C,\Phi\mathrm{max}}$ are the heating and, respectively, the cooling capacity of the AWHP unit at maximum inverter frequency.

2.6 Life cycle costs over the calculation periodof the considered structural schemeof air-to-water heat pump systems

The life cycle costs (LCC) over the calculation period of the considered structural scheme of heat pump systems are determined according to the the guidance presented in the EuropeanRegulation 244/2012/EU. As per [17], the LCC over the calculation period is determined as follows:

$$C_g(\tau) = C_I + \sum_j \left[\sum_{i=i}^{\tau} \left(C_{a,i}(j) \cdot R_d(i) \right) - V_{f,\tau}(j) \right] \tag{21}$$

where: τ- the economic life of the heat pump system. It is assumed to be $\tau = 30$ year [17];
$V_{f,\tau}(j)$ - residual value of measure or set of measures *j* at the end of the calculation period (discounted to the starting year τ_0), in EUR. In this study is assumed that $V_{f,\tau}(j) = 0$ at the end of the economic life of the system;
$R_d(i)$ - discount factor for year *i* based on discount rate, *r*.It can be written as[17]:

$$R_d(i) = \left(\frac{1}{1 + r/100} \right)^i \tag{22}$$

where: *r* is the real discount rate.

In the current investigation, it is assumed that the real discount rate is equal to $r = 6$ % [17].
C_I - initial investment costs for measure or set of measures *j*, EUR;
$C_{a,I}$ *(j)* - annual cost during year *i* for measure or set of measures *j*, EUR.

In this study, the initial investment costs, C_I, are determined as a sum of the cost of heat pump and back-up heater. The initial investment costs for the heat pump is defined according to official price list of the manufacture [18], whereas the initial investment costs of the back-up heater, C_{BUH}, is determined as follow:

$$C_{I,\mathrm{BUH}} = 0.02586 \cdot Q_{BUH}^2 + 5.8941 \cdot Q_{\mathrm{BUH}} + 373.045 \quad (23)$$

The term Q_{BUH} in equation (22) is the thermal capacity of back-up heater (in kW).

In the mathematical model of the considered systems, maintenance costs during the *i-th* year are considered as the sum of the repair, recommissioning, replacement, and asset preservation costs of the heating and cooling system. These costs can be defined as:

$$C_{I,P} = 1\%.C_I\ , [EUR] \quad (24)$$

The operating costs of the considered structural scheme of heat pump systems, $C_{H/C}$, involve the fuel costs realized during the *i-th* year of the system's economic life. They are defined using the annual energy demand of the building and the electricity price (c_{el} = *0.11984 EUR/kWh)* as follow[19]:

$$C_{H/C} = c_{el}\ .\ Q_{HP,H/C,el}(i),\ [\mathrm{EUR}] \quad (25)$$

3 Results and discussion

To understand how the thermal capacity of the AWHP affects the seasonal energy efficiency and *LCC*, various simulations have been conducted taking into consideration the studied structural schemes of the heat pump system.

The bivalent temperature, θ_{biv}, corresponding to the analyzed structural schemes of the heat pump system is summarized in Fig. 9. It is clear that the heating capacity of the AWHP at maximum inverter frequency decreased with increasing of the bivalent temperature. On the other hand, the lowest bivalent temperature also leads to a decrease in the cooling capacity of the heat pump unit.

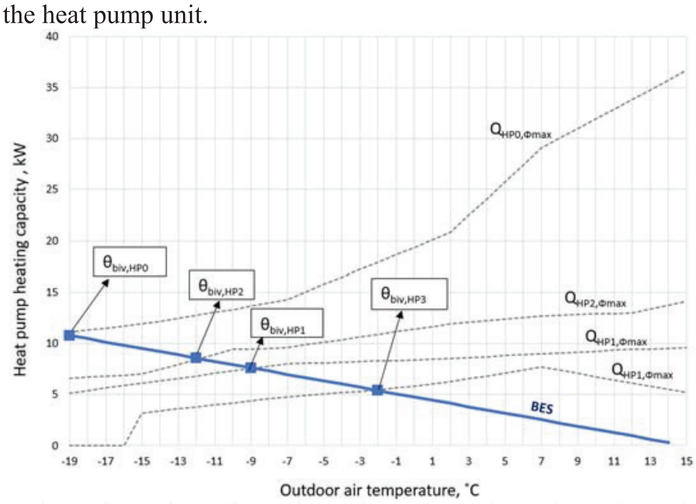

Fig. 9. BESand heating capacityat maximum inverter frequency of simulated heat pump systems

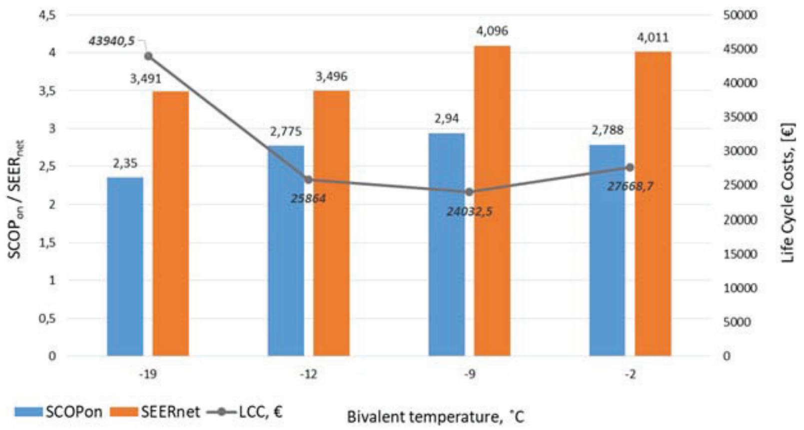

Fig. 10. SCOP$_{on}$, SEER$_{net}$ and LCC of simulated heat pumps as a function of the bivalent temperature

The effect of the bivalent temperature on the seasonal energy performance of the AWHP unit and *LCC* of the system can be easily understood from Fig. 10 and Table 3. It is clear that with increasing of the θ_{biv}, $SCOP_{on}$ and $SERR_{net}$ also rise. Therefore, if the installed AWHP unit is described by a thermal capacity much larger than (or equal to) the building design load, it will always operate at partial load or it will realize on-off cycles. Thus, it is introduced the energy losses into the heat pump cycle. An endorsement for this statement is the fact that $SCOP_{on}$ and $SEER_{net}$ values grow by 20.6% and around 13%, respectively when the bivalent temperature rises with 17K.

The next section of the survey has been concerned with the seasonal and annual energy performance of the as well as the life cycle cost overall heat pump system (AWHP unit and back-up system). The results from Table 3 show that the maximum of the seasonal and annual energy performance of the system is observed at θ_{biv} = -9°C. The possible reason for this unexpected outcome is that the time of operation at a partial load of the AWHP from case study HP_1 is around 43% shorter than the partial load operation time of the AWHP in case study HP_2. It is important to note that the statements described above are valid when the heat pump system operates in parallel mode.

As a personification of the results presented so far is the *LCC* data listed in Fig. 10. The highest value of the life cycle cost is observed for the monovalent heat pump system. Furthermore, the case study HP_0 is characterized by the highest value of the bivalent temperature. According to Fig. 10, it can be argued that the optimal value of the bivalent temperature in the case of the considered bivalent-parallel heat pump system is θ_{biv} =-9°C because the *LCC* is minimal.

Table 3. Annual performance coefficients of simulated heat pump systems as a function of the bivalent temperature

Case study	Capacity at $\theta_{e,peak}$		θ_{biv}, °C	SCOP$_{net}$	SEER$_{on}$	Oversizing		APF
	Heating	Cooling				Heating	Cooling	
HP_0	11.3	19.58	-19	2.35	3.491	+5 %	+140 %	2.57
HP_2	6.7	9.52	-12	2.778	3.496	-38.8 %	+17 %	2.92
HP_1	5.38	7.5	-9	2.952	4.096	-52.3 %	-8 %	3.14
HP_3	0	4.99	$\theta_{cut-off}$ = -2	3.079	4.011	-100 %	-39 %	2.70
HP_4	0	4.99	-2	2.959	4.011	-100 %	-39 %	3.06

Note: In case study HP_3 the heat pump unit operates in alternative mode,while in caseHP_4 it operates in parallel mode.

On the other hand, the comparison between parallel and alternative operation mode of the heat pump system (i.e. the case studies from Table 3 denoted as HP_4 and HP_3, respectively) shows that there is a significant difference between the seasonal and annual energy performance values. When a structural scheme of the heat pump system is based on the same AWHP unit, the $SCOP_{on}$ of the system in parallel mode is 17 % higher than the $SCOP_{on}$ of the system in the alternative mode. This is valid when the cut-off temperature is setting equal to the bivalent temperature. On the other hand, the values of the seasonal performance coefficient of the unit, $SCOP_{net}$, indicate that more efficiency for the AWHP is an operation in the alternative mode. An analysis of the effect of the heat pump cut-off temperature on seasonal and annual performance coefficients, however, is necessary.

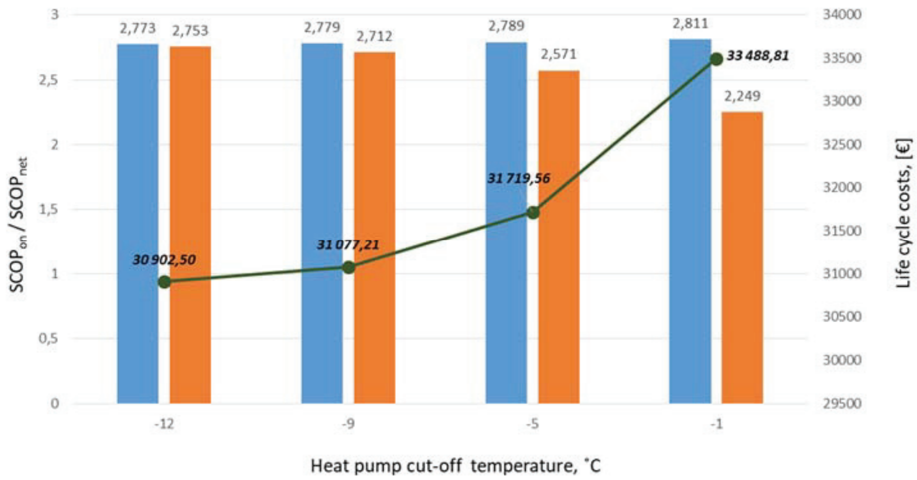

Fig. 11. $SCOP_{on}$ and $SCOP_{net}$ of simulated heat pump systems as a function of the heat pump cut-off temperature

It can be seen from the data in Table 4 and Fig. 11 that if the cut-off temperature increase, the seasonal energy performance of the AWHP unit ($SCOP_{net}$ and $SEER_{net}$) also raise. The reason for this is the fact that at high cut-off temperatures, the heat pump consumed a lower amount of electrical energy. Contrarily, the seasonal and annual performance coefficients of the overall heat pump system decrease with the cut-off temperature increasing. The reason is higher electrical power consumed by the back-up heater. Moreover, this leads to gradually raising the LCC. The results highlight that the best values of the LCC and the annual performance factor of the considered bivalent – alternative systems are observed at the value of the cut-off temperature equal to -12°C.

Table 4. Data for seasonal and annual performance coefficients for a heat pump system operating in alternative mode

Case study	Capacity at θ_{des}		$\theta_{HP\ cut\text{-}off}$, ˚C	$SCOP_{net}$	$SCOP_{on}$	$SEER_{on}$	APF
	Heating	Cooling					
HP_0	6.7	9.52	-11	2.773	2.753	3.496	2.898
HP_1	6.7	9.52	-9	2.779	2.712	3.496	2.863
HP_2	6.7	9.52	-5	2.789	2.571	3.496	2.742
HP_3	6.7	9.52	-1	2.811	2.249	3.496	2.455

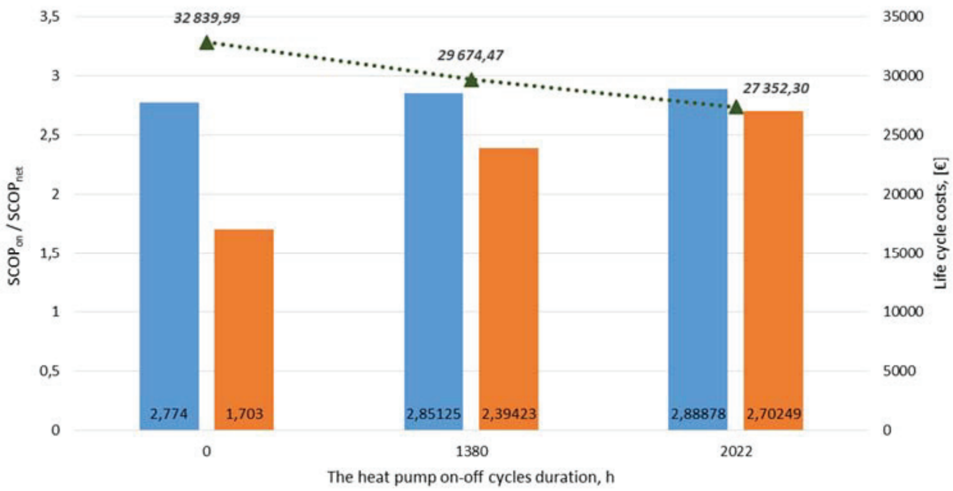

Fig. 12. $SCOP_{on}$ and $SCOP_{net}$ of simulated heat pumps as a function of the on-off cycles duration

In the final part of the survey has been evaluated the influence of on-off cycles duration on the considered criteria for techno-economic assessment. In the conducted simulations of the heat pump system, instead of realizing the on-off cycles by the heat pump, it is assumed that the back-up system will operate. In this way aims to demonstrate how the energy performance and *LCC* of the system based on the same size heat pump unit vary when changing the duration of this operation mode. The obtained results are summarized in Table 5 and Fig. 12. A significant decrease in on-off cycles duration does not lead to an increase in the seasonal and annual energy performance values of either the AWHP unit and the overall system. Moreover, the findings confirm the notion that if the objective is to minimize on-off cycles duration to optimize the *LCC* and *SCOP* values, then the system must be designed with a smaller size heat pump unit. The verification of this statement is the results listed in Table 3: when the oversizing parameters are reduced, the on-off cycle duration is shorter around 51% in the case study HP_2 than HP_0.

Table 5. Data for seasonal and annual performance coefficients as a function of the on-off cycles duration

Case study	Capacity at θ_{des}		$\tau_{HP\ on\text{-}off}$, h	$SCOP_{net}$	$SCOP_{on}$	$SEER_{on}$	APF
	Heating	Cooling					
HP0	6.7	9.52	0	2.774	1.703	3.496	2.023
HP1	6.7	9.52	1380	2.851	2.394	3.496	2.511
HP2	6.7	9.52	2022	2.889	2.702	3.496	2.574

4 Conclusion

The purpose of the current study was to assess the influence of the heat pump thermal capacity over the seasonal and annual performance of the heat pump systems. It was modeled several configurations of the structural schemes of the heat pump systems: monovalent, bivalent-parallel and bivalent-alternative heat pump systems. The analyzed heat pump systems were evaluated through preliminary criteria for techno-economic assessment such as seasonal coefficient of performance (SCOP), seasonal energy efficiency rate (SEER) and life cycle costs (LCC).

The current investigation demonstrated that there are two options for planning air – to – water heat pump systems: the AWHP system to be modeled based on the lowest external air temperature (θ_e = -19°C) or to be planned the operation of the system in bivalent mode and to set the bivalence point. In general, therefore, it seems that the monovalent heat pump system, for which the AWHP unit thermal capacity is determined based on the peak thermal load required by the building is not an appropriate decision for the Black Sea climate of Bulgaria. On the other hand, the obtained results demonstrate that if the design objective is an AWHP system with minimal LCC, then the bivalent point must never deviate too far from the design external air temperature.

When the heat pump unit operates in parallel mode, the energy performance parameters and LCC are better than the alternative operation mode. For example, realizing the bivalent-parallel structure scheme of the heat pump system is achieved a 17% rise of the seasonal coefficient of performance of the overall system. Furthermore, this study demonstrated that the design parameter with the greatest influence on the energy performance and life cycle costs of the system is the bivalent temperature. Considerably more work will need to be done to determine the techno-economic optimal value of the bivalent temperature.

In conclusion, the best value of the seasonal coefficient of performance (SCOP), seasonal energy efficiency rate (SEER) and life cycle costs (LCC) is obtained when the AWHP is designed to match around 50% of the peak heating load required by the building. Therefore, the increase in the energy performance of the overall heat pump system (the AWHP unit and back-up electrical heater) can be achieved by reducing the difference between the nominal thermal capacity of the heat pump at the design outdoor temperature and the net energy demand for heating required by the building.

This study was considered actual AWHP units. The cooling and heating capacities, as well as COP and EER of the units, are determined according to the data published by the manufacture. Therefore, the current investigation can be defined as useful because it outlines the independent variables of the further optimization problem. Previously, however, it is necessary to be developed a generalized mathematical model taking into account the specific climatic conditions, inverter frequency, and coefficient of performance (i.e., $COP_{\Phi eff}$, $COP_{\Phi max}$, $EER_{\Phi eff}$ and $EER_{\Phi max}$) of the AWHP regardless of the data declared by a particular manufacturer.

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