ISEC 2024 – 3rd International Sustainable Energy Conference

Energy Flexibility through Sector Coupling

https://doi.org/10.52825/isec.v1i.1141

Published: 18 Apr. 2024

Dimensioning Method for PVT Collectors as Heat Source of Heat Pumps for Residential Buildings

Krishna Timilsina[1,*], Bharat Chhugani[1], Harshvadan Modi[1], and Peter Pärisch[1]

[1] Institute for Solar Energy Research Hamelin, Germany

*Correspondence: Krishna Timilsina, timilsina@isfh.de

Abstract. Photovoltaic-thermal (PVT) collectors are an emerging technology that is increasingly being used as a heat source for heat pump systems in residential buildings. However, a suitable standard methodology for sizing the PVT collectors for these systems is still not available. This paper initially provides a framework for the sizing of a PVT-heat pump system for small houses according to the German guideline VDI 4645. The dimensioning method of VDI 4645 and the sizing method for PVT collectors are incorporated in a web-based tool that is aimed to assist planners (or homeowners) during the preliminary planning of a heat pump system in single and multi-family houses. The methodology also covers the planning of systems with backup/additional heaters (e.g. gas boiler and heating rod), e.g. for buildings with limited roof areas for PVT installations. Different heat pump operation modes, i.e. monovalent, bivalent-alternative, or bivalent-parallel are also considered. A model of the IEA SHC Task44 SFH100, an existing single-family house with radiators, has been chosen as an example case to demonstrate the methodology. For the evaluation of the performance of the designed system, yearly simulations of the system are done in TRNSYS. The results show that the developed methodology provides plausible sizes for the example case. However, further development and validation are necessary to provide flexibility in system dimensioning.

Keywords: PVT Collectors, Heat Pumps, VDI 4645, Heat Load, Design Point, Operation Modes

1 Introduction

Heat pumps are considered to be one of the key technologies for achieving climate targets in the building sector by reducing primary energy consumption and CO_2 emissions. The installation of heat pumps for heat supply in Germany grew by 53 % in 2022 compared to 2021 [1]. Although the air-source heat pumps cover the largest market share, it is forecasted that the market share of brine-water heat pumps in Europe will go up by 110 % in 2024 compared to 2020 [2]. Due to the noise limitations and relatively lower efficiencies of air-source heat pumps, brine-water heat pumps with various sources are increasingly used for larger buildings. Brine-water heat pumps are mainly coupled with geothermal sources such as borehole heat exchangers (BHE) and horizontal ground heat exchangers (HGHE). In recent years, PVT collectors have also been increasingly used as a heat source for brine-water heat pumps due to their potential to co-generate both heat and electricity from a single module. Increasing research and development of the PVT collectors have resulted in improved designs and increased thermal outputs in recent years [3], [4], [5]. Uncovered PVT collectors without insulation on the back side and with increased heat exchange surface through fins on the rear side are found to operate at very low outdoor temperatures as low as -15 °C as a heat source for a low-temperature heat pump [6]. Monitoring and simulation results show that the PVT collectors could be a key technology as a heat source for the heat pump, providing better system efficiency than air-source heat pumps and can cut off CO_2 emissions even more than ground-source heat pumps [7]. Although the quality of PVT collectors is assured by ISO 9806 standard, PVT-heat pump systems are not covered by standards and guidelines.

The ongoing German project "integraTE" focuses on heat pump systems with PVT collectors as a heat source and aims to increase the market penetration of technically and economically attractive energy supply concepts with PVT collectors in the building sector. As part of the project, it has been aimed to develop a methodology to facilitate PVT-heat pump system design. In this paper, a method for sizing the components of a PVT-heat pump system in residential buildings as shown in Figure 1 is presented.

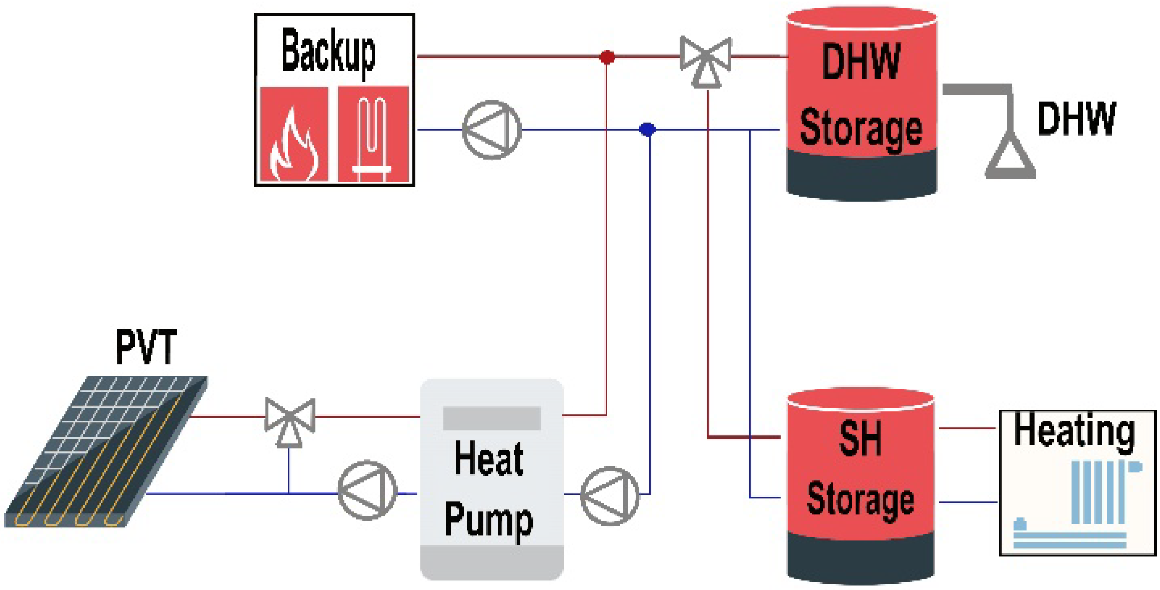

***Figure 1**. Schematic representation of the PVT-heat pump system covered by the dimensioning method.*

The existing German guideline VDI 4645 [8] mainly provides a framework for the planning and dimensioning of air-source heat pumps based on the demands for space heating and domestic hot water at the design point. Various heat pump manufacturers also provide the dimensioning procedures for the heat pumps. However, most of them are focused on the dimensioning of the air-source heat pumps and ground-source heat pumps [9], [10]. This work presents the adapted methodology for brine-water heat pumps with PVT collectors as a sole heat source. The methodology described in this paper could be used to dimension brine-water heat pump systems that are capable of operating at lower temperature levels. This paper also presents an example case study for dimensioning the components of the IEA SHC Task 44 SFH100, an existing single-family house. The performance of the dimensioned heat pump system for a year has also been analyzed through simulations in TRNSYS.

2 Methodology

Figure 2 illustrates the outline of the methodology for the dimensioning of heat pump systems with PVT collectors as the sole heat source. It will be described in detail in Chapter 4. For the dimensioning, information about the building such as heat load, number of residents, and the number of households (for multi-family houses) is required. Further information about the location of the building and design weather conditions i.e. the nominal outdoor temperature, irradiation, wind speed, and balance point temperatures are also required. The desired operation modes (monovalent or bivalent), method of heating (radiator or floor heating), the blocking time specified by the utility, and the temperature levels of the domestic hot water (DHW) and space heating storage tanks are also required parameters for the calculation. The above-specified parameters are processed using the mathematical models and procedures specified in VDI 4645 to determine the daily heat demands for space heating and DHW preparation. The size of the storage tanks for space heating and domestic hot water preparation for the undistorted operation of the heating system is also estimated. Furthermore, the methodology estimates the dimension of the heat pump, which is sufficiently large to meet the heat requirements based on the operation modes and blocking time. At this point, the PVT collectors as heat source have to be designed. Once the heat pump is dimensioned, the information about the heat pump and its characteristics under the design conditions has to be known or calculated. The weather conditions at the design point are necessary to calculate the thermal output (in kW) per m^2 of PVT based on a mathematical model specified by ISO 9806 standard and the thermal performance parameters of the chosen PVT collectors. In the context of bivalent systems, the size of a backup heater is calculated based on the heat demand that the heat pump does not meet at the design point.

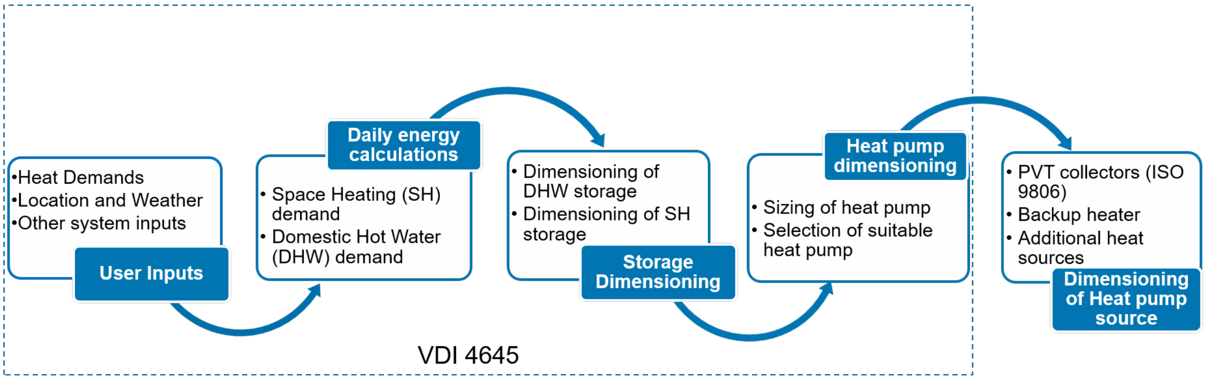

Figure 2. *Methodology for dimensioning of heat pump system.*

3 Description of inputs for example building

For a clear illustration of the dimensioning steps, the methodology described in this paper is implemented for designing a PVT-heat pump system for a building model, the IEA SHC Task44 SFH100 [11], for the location of Würzburg, Germany. The daily average heat load of the house for various ambient temperatures is shown in Figure 3. The diagram also illustrates the trendline that is considered as a heating curve for the building. The nominal ambient temperature for Würzburg is -10.1 °C [12]. Based on the heating curve, the heat load at nominal ambient temperature was found to be 7.5 kW. The design supply temperature to the space heating for the radiators is 55 °C and the return temperature is 45 °C. The hot water tapping temperature for domestic hot water is 45 °C. The detailed description of the building and system is derived from Chhugani et. al. [13].

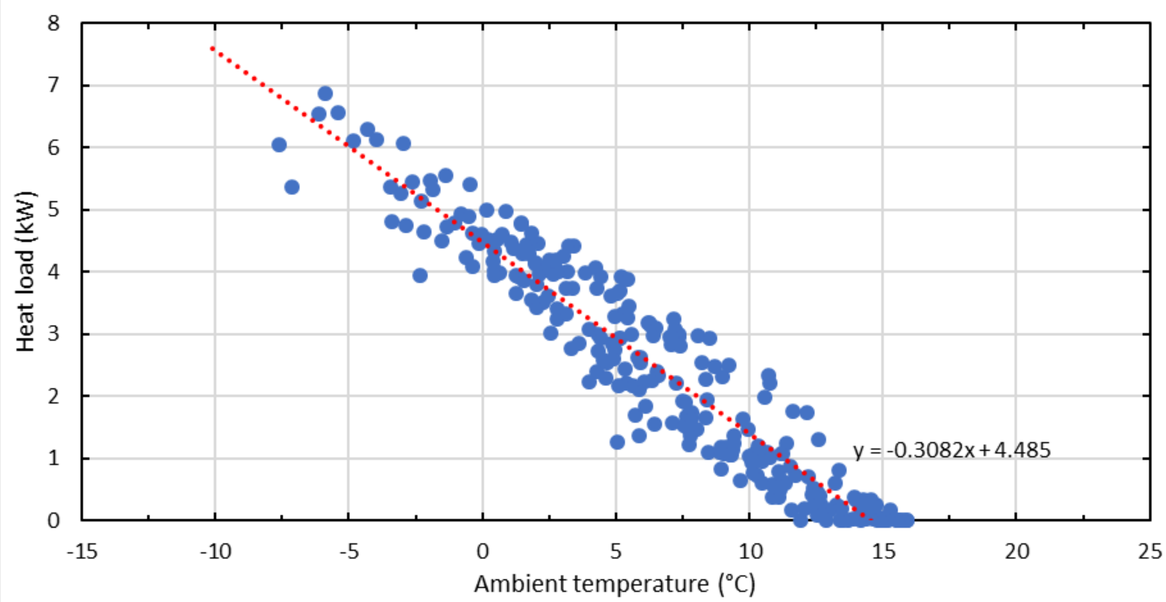

Figure 3. *Average daily heat load for space heating as a function of ambient temperature for SFH 100 at Würzburg.*

4 Dimensioning steps in detail

4.1 Space heating demands

The heat load refers to the amount of heat required to maintain the specified nominal indoor temperature in the building. The heat load of the building at nominal outdoor temperature is to be considered for the calculations as it refers to the maximum heat requirement. This heat load should be either already available from measurements for existing buildings or can also be calculated with the information about the ambient conditions and the information about the house using DIN EN 12831-1 [14]. The latter tends to overestimate the heat load as it neglects capacitance effects.

If the heat load of a building is unknown, it can be estimated based on the nominal outdoor temperature and nominal indoor temperature for various buildings based on the construction year and the type and state of retrofitting i.e. existing condition, conventional or future-oriented retrofitting [15]. Eq. (1) is the simplified model for the stationary calculation of the heat load ($\dot{Q}_{H,N}$) at the nominal outdoor (ϑ_{ext}) and indoor (ϑ_{int}) temperatures based on the standard DIN EN 12831-1. On the basis of the information about the building, a value for the transmission heat loss (H_T) in Eq. (1) can be estimated. Additionally, the coefficient of heat transfer for ventilation (H_V) is calculated based on Eq. (2) with the height of the heated living space (h), the airflow rate (n), and the total living area (A). The nominal indoor temperature for residential buildings can be fixed to 20 °C and the product of specific heat of air ($c_{p,a}$) and the density of air (ρ_a) also could be fixed to 0.34 Wh/(m^3·K). In Eq. (1), the term $\dot{Q}_g$ refers to the sum of heat gains by the building including the solar gains and internal heat gains whereas $\eta_{h,gn}$ refers to the gain utilization factor for heating. The heat up losses of the building heating system, like after a night setback, are neglected as it would increase the heat pump's power and investment costs and lower the efficiency in part load.

$$\dot{Q}_{H,N} = \sum H_T \cdot A \cdot (\vartheta_{int} - \vartheta_{ext}) + \sum H_V \cdot (\vartheta_{int} - \vartheta_{ext}) - \eta_{h,gn} \cdot \dot{Q}_g \tag{1}$$

$$H_V = A * h * n * c_{p,a} * \rho_a \tag{2}$$

Eqs. (1) and (2) can also be used to calculate the heating load of a building at ambient temperatures marked as intermediate point (ϑ_{int}) in Figure 4.

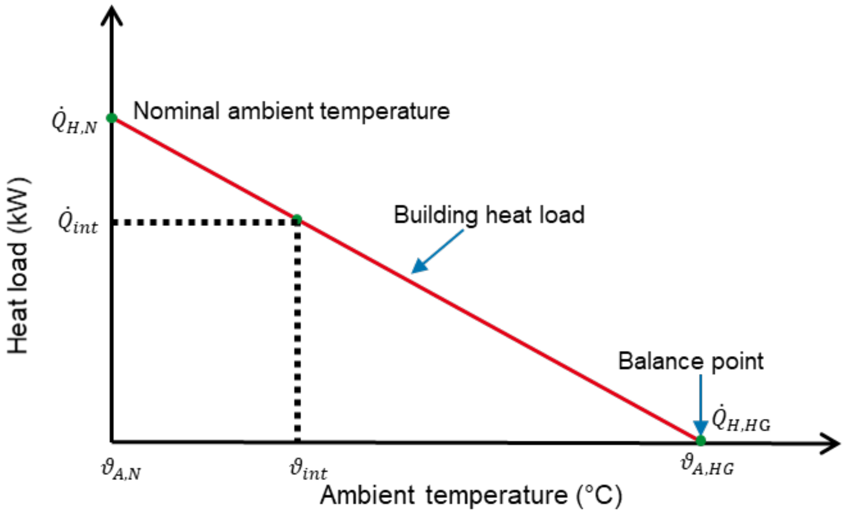

Figure 4. Heating curve of a building for space heating.

Alternatively, if the heat load is already available for the nominal ambient temperature, Eq. (3) is used to calculate the building heat load at intermediate temperatures (ϑ_{int}) using linear interpolation based on Figure 4.

$$\dot{Q}_{int} = \frac{\dot{Q}_{H,N}.(\vartheta_{A,HG} - \vartheta_{int})}{(\vartheta_{A,HG} - \vartheta_{A,N})} \tag{3}$$

The balance point referred by the temperature ($\vartheta_{A,HG}$) is the temperature at which the heat loss of the building is zero i.e. the heating system turns on below this temperature.

The daily heat demand can be calculated from the heat load $\dot{Q}_{H,AP}$ as depicted in Eq. (4) where n is the total number of operating hours for the space heating in a day. The heat load is considered to be constant for 24 hours during the calculation of daily heat demand.

$$Q_{H,AP} = \int_0^n \dot{Q}_{H,N} \cdot dt \tag{4}$$

As already discussed in Section 3, the heat demand for the example building taken in this paper at nominal ambient temperature results in 7.6 kW. And based on Eq. (4), the total daily heat demand is 182.4 kWh.

4.2 Domestic hot water demands

As the methodology is intended for the dimensioning of a system with both space heating and domestic hot water requirements, the heat demand for DHW preparation has to be estimated. In the literature, several simplified procedures as well as detailed procedures with a complete daily profile of the DHW demands are available [8], [16]. The total energy demand for the DHW preparation from the daily energy demand for n_{NE} number of residential units ($Q_{DP,NE}$) is given by Eq. (5).

$$Q_{DP} = n_{NE} \cdot Q_{DP,NE} \tag{5}$$

Additionally, to calculate the storage volume of the DHW storage, the reference period with the greatest energy demand in the DHW profile ($Q_{DPB,NE}$) is to be considered (e.g. duration 1 h). The total energy demand during the reference period with the highest energy demand for the specified number of residential units (n_{NE}) is determined using Eq. (6).

$$Q_{DPB} = n_{NE} \cdot Q_{DPB,NE} \tag{6}$$

If the number of persons is specified, the daily energy demand and the highest energy demand for the reference period can be selected from standardized profiles more accurately. For larger buildings, the simultaneity factor should be taken into account.

The standby losses for the selected storage (Q_{SP}) are taken from the manufacturer's specifications to calculate the total daily energy demand for the DHW using Eq. (7). If existent, the circulation losses (Q_{Zirk}) have to be taken into account additionally, so that the heat loss due to circulation is compensated. For line losses without circulation, it is recommended to apply losses in the amount of 1 kWh per day and residential units of use [8].

$$Q_{DP,ges} = Q_{DP} + Q_{SP} + Q_{Zirk} \tag{7}$$

4.3 Storage dimensioning

Heat storage in heat pump systems is primarily intended to compensate for the differences in output and time between heat generation and demand. Additionally, heat storage units have a useful function in the load management of power grids [17]. Also, storage tanks play an important role in bridging the blocking periods and optimization of self-consumption in the systems with electricity production. To simplify the calculations, it has been considered that there are two different heat storages for domestic hot water and space heating in this methodology.

4.3.1 Domestic Hot Water

In order to determine the volume of the DHW storage tank, the set point temperature of the storage and the temperature of cold water has to be defined. Depending on the DHW profile and the amount of energy calculated in Eq. (7), the required storage volume for the direct storage of the DHW is determined by Eq. (8). For instantaneous water heaters, flow and return temperatures have to be taken instead.

$$V_{DPB} = \frac{Q_{DPB}}{c_W \cdot (\vartheta_{soll} - \vartheta_{KW})} \tag{8}$$

If the circulation line exists, the DHW volume that compensates for the circulation losses is given by Eq. (9). The circulation heat ($Q_{Zirk,ref}$) is taken for the same reference period as the demand (Q_{DPB}) (e.g. 1 h).

$$V_{Zirk} = \frac{Q_{Zirk,ref}}{c_W \cdot (\vartheta_{soll} - \vartheta_{Zirk,RL})} \tag{9}$$

Additionally, the stratification behavior during tapping or recharging of the DHW storage should also be considered. Furthermore, an allowance for unusable storage volume due to mixing (storage efficiency) has to be considered. This is considered a surcharge in the equation as f_{TWE}. Therefore, the required storage volume (V_{DHW}) is calculated using Eq. (10).

$$V_{DHW} = (V_{DPB} + V_{Zirk}) \cdot f_{TWE} \tag{10}$$

In the IEA SHC Task 44 example, the reference period (1 hour) with the maximum demand, which is necessary to dimension the DHW storage, has the maximum demand of 2.2 kWh [18]. Using Eq. (8) with the cold-water temperature (ϑ_{KW}) of 10 °C and the tapping temperature (ϑ_{soll}) of 45 °C, the volume of the storage domestic hot water for the peak demand is 54 liters. Considering the mixing losses of 15 % and no circulation, the volume of DHW storage is 63 liters. The next available size of the chosen manufacturer could be 80 liters for instance.

For the chosen building model, the daily DHW demands are 5.8 kWh [18], and as the DHW storage is selected without circulation, the recommended minimum daily pipe loss of 1 kWh is considered. Considering the daily standby loss of the selected storage of 0.9 kWh, the total heat requirement for DHW preparation given by Eq. (7) is 7.7 kWh.

4.3.2 Space Heating Storage

The volume for space heating storage is estimated based on the recommended values for the sizing of the storage considering the blocking hours. Heat pump manufacturer Stiebel Eltron recommends different storage sizes for radiator and floor heating systems [9]. The coefficients of the regression derived from storage sizes suggested in the manufacturer guideline depend on the maximum heat load and the thermal inertia of the system. Eqs. (11) and (12) can be used to calculate the recommended storage size in liters for floor heating (high inertia) and radiator heating (low inertia) systems respectively [17]. The linear fitting relationship is shown in Eqs. (11) and (12) are valid for space heating loads ($\dot{Q}_{H,N}$) up to 108 kW.

$$V_{RH,floor} = 19.4\ [\mathrm{L}] + 28.1\ \left[\frac{\mathrm{L}}{\mathrm{kW}}\right] \cdot \dot{Q}_{H,N}\ [\mathrm{kW}] \tag{11}$$

$$V_{RH,radiator} = 81.54\ [\mathrm{L}] + 53.8\ \left[\frac{\mathrm{L}}{\mathrm{kW}}\right] \cdot \dot{Q}_{H,N}\ [\mathrm{kW}] \tag{12}$$

The building model has space heating with radiators. Therefore Eq. (12) results in a storage size of 485 liters.

4.4 Heat pump dimensioning

After the total daily heat demand of the building including the space heating and domestic hot water is calculated, the size of the heat pump required to fulfill the heat demand at the design point (index: AP) can be calculated. Eq. (13) is used to determine the required heat pump output at the design point considering the blocking time specified by the utility (t_{SD}). The quantity Q_{sonst} refers to the heat quantity in kWh for any other existing consumers at the design point and is neglected in our example case.

$$\dot{Q}_{HP,req} = \frac{Q_{H,AP} + Q_{DP,ges} + Q_{sonst}}{d - \sum t_{SD}} \tag{13}$$

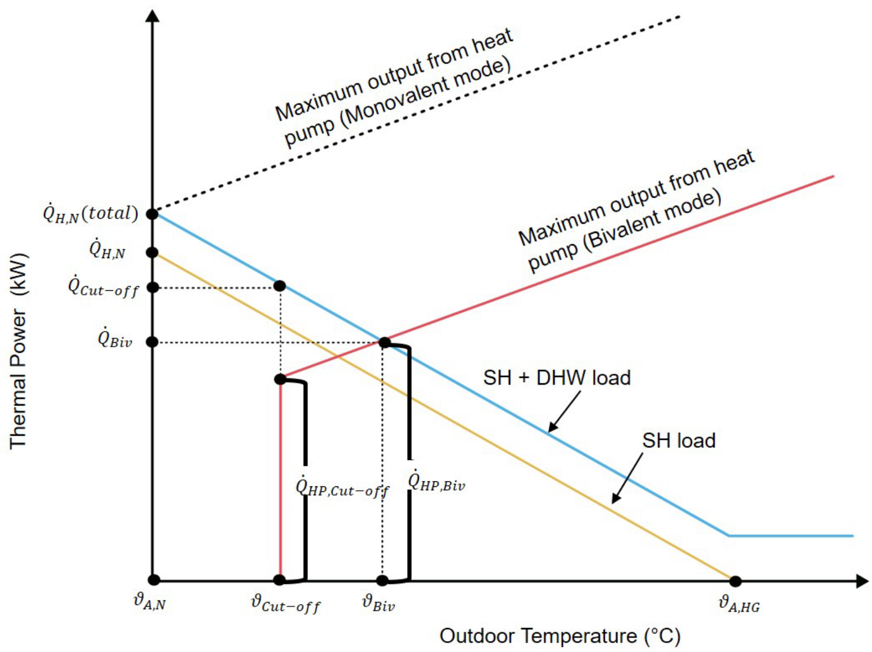

Figure 5. *Variation of heat load and maximum heat pump output at various outdoor temperature.*

The operation modes of the heat pump play a vital role in the selection of the heat pump. Figure 5 shows the heating curve and maximum heat output for two exemplary heat pumps at various operating points. It is also to be noted that in contrast to Figure 4, the heat demands at various temperatures include both space heating and DHW demands.

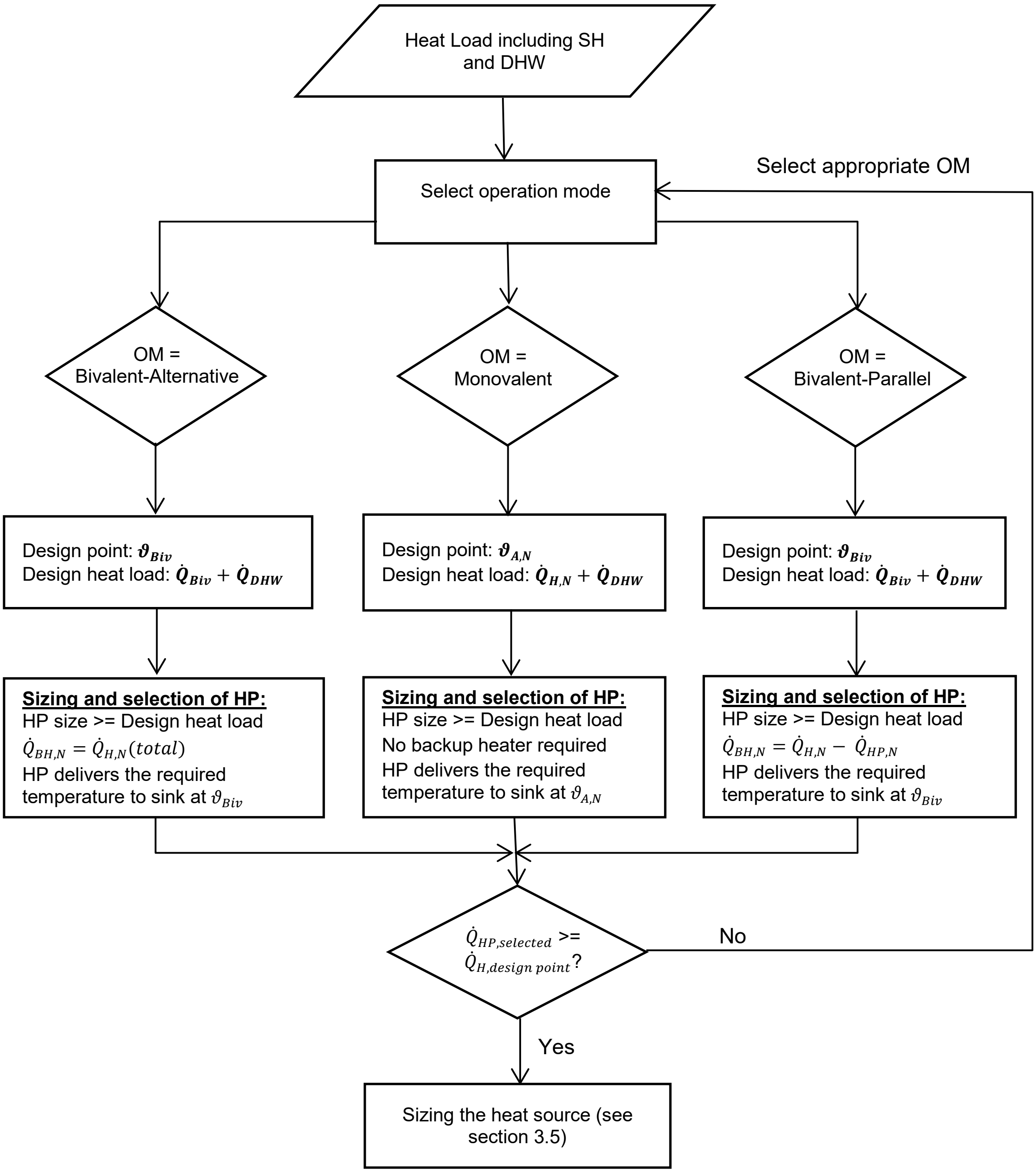

Figure 6. *Design and selection of heat pump based on operation modes.*

In general, the bivalence point refers to the operating point at which the heat pump output is sufficient to meet the total heat demand of the building and below which the backup heater or a second heat generator is needed to support the heat pump either partially or fully. Below the cut-off point, the heat pump reaches its operating limit and the second heat generator takes over the full thermal load. If the cut-off point is equal to or below the nominal ambient temperature, the heat pump operates in bivalent-parallel mode along with the backup heater. If the cut-off point is between the bivalence point and the nominal ambient temperature, the operation mode of the heat pump is called bivalent-partially parallel. If the cut-off point is equal to the bivalence point, the operation mode is called bivalent-alternative, as there is no parallel operation. If the bivalence point is equal to the nominal ambient temperature, the heat pump is designed for monovalent operation. In that case, the heat pump should be able to deliver the total heat requirement at the design point, i.e. the nominal outdoor temperature, and all year round.

Running a heat pump at lower temperature levels is less efficient and oversizing of heat pump in part load will increase costs and start-stop losses. So, a partial fulfillment of the heat load at the design point (bivalence mode) is a common solution, especially for air source heat pumps, and can be economically attractive.

Figure 6 shows a flow chart for the design and selection of the heat pump based on the operation mode (OM). The sum of the space heating load ($\dot{Q}_{H,AP}$) and the heat load for DHW preparation ($\dot{Q}_{DHW}$) at design point is denoted as $\dot{Q}_{H,design\ point}$. For each operation mode, the respective heat load varies with the selection of the design point. The selected heat pump therefore has to meet the heat load at the respective operation mode. Additionally, the temperature at the condenser has to be considered during the selection of the heat pump.

For the example case, it is considered that there is no shut-off time for the heat pump system. The required heat pump size at the nominal ambient temperature calculated from the daily heat demands in sections 4.1 to 4.3, is 7.9 kW. However, as the heat load varies with the variation in ambient temperature, the heat pump size required is to be calculated for the design point based on the selection of operation modes.

4.5 Design of heat source

The design of the heat source is influenced by the heat demand, heat pump efficiency, and the temperature levels of the heat source. Additionally, the existence of storage tanks and other heat sources also impacts the dimensioning of the heat sources. A simple block diagram of the factors influencing the design of heat sources and the calculation pathway is shown in Figure 7 [19].

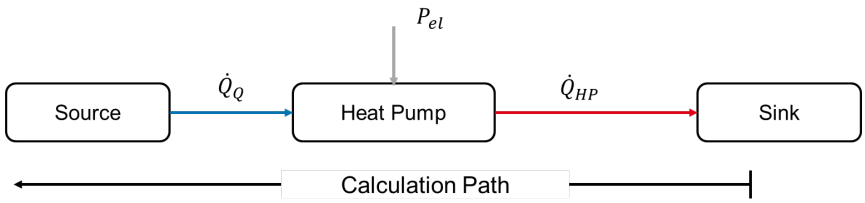

Figure 7. *Factors influencing the design of heat sources acc. to [19].*

$$\dot{Q}_{HP,selected} = \dot{Q}_Q + P_{el} = \dot{Q}_Q + \frac{\dot{Q}_{HP,selected}}{\varepsilon} \tag{14}$$

$$\dot{Q}_Q = \dot{Q}_{HP,selected} - \frac{\dot{Q}_{HP,selected}}{\varepsilon} \tag{15}$$

The dimensioning method of a heat pump with the consideration of the factors from sink sides has already been discussed in previous sections. The capacity of the heat source needed to fulfill the heat demand on the evaporator (source) side of the heat pump can be calculated by using Eqs. (14) and (15). This heat input ($\dot{Q}_Q$) depends on the selected heat pump, the coefficient of performance (COP) ε, operation modes, and design point.

However, the COP of the heat pump varies with the design point and the required supply temperature. During heat pump operation, this temperature requirement for DHW and SH can be on different levels. Therefore, we extend the methodology of VDI 4645 to include these variations, ensuring accurate integration of the heat pump's operation at different temperature levels for different purposes. The fact that different losses (e.g. circulation losses, standby losses, etc.) occur at different temperature levels is also taken into account. In Eq. (16), the quantity $\dot{Q}$ refers to the heat demand and the quantity ε refers to the temperature-dependent COP. The suffixes SH and DHW refer to heat pump outputs and COPs for space heating and domestic hot water respectively. The suffix Losses,i refers to the various losses occurring in the heat supply system at various temperatures. Hence, the design of the heat source is also dependent of the hydraulic layout of the system, which is not covered by this paper.

$$\dot{Q}_Q = \dot{Q}_{SH}\left(1-\frac{1}{\varepsilon_{SH}}\right)+\dot{Q}_{DHW}\left(1-\frac{1}{\varepsilon_{DHW}}\right)+\sum_i \dot{Q}_{Losses,i}\left(1-\frac{1}{\varepsilon_{Losses,i}}\right) \tag{16}$$

Based on the design ambient temperature, the inlet temperature of the heat pump is calculated considering that the inlet temperature of the heat pump should not be more than 5 K lower than the ambient temperature for a brine-water heat pump with PVT as the heat source [6]. The datasheet from the manufacturer at various reference inlet temperatures gives the capacity and the coefficient of performance of the heat pump.

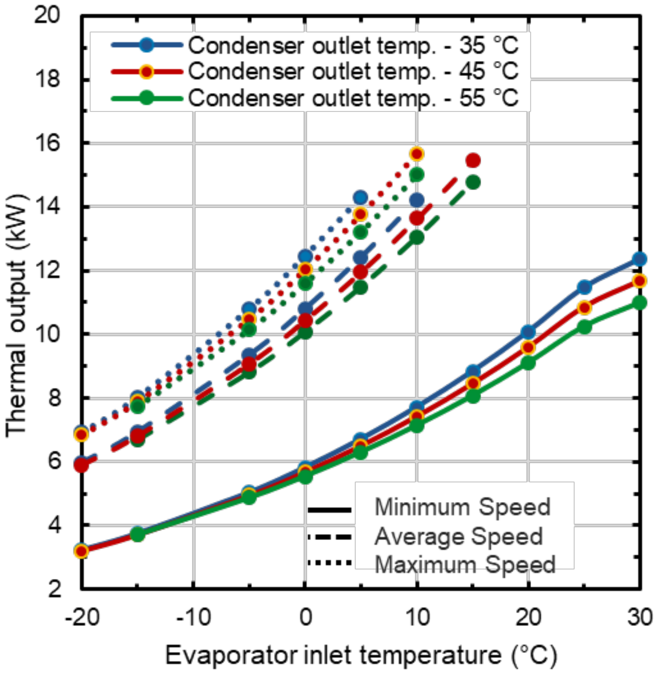

Figure 8. *Thermal output of heat pump (y-axis) for different evaporator inlet temperatures (x-axis) and condenser outlet temperatures (blue 35 °C, red 45 °C, green 55 °C) for three different compressor speeds.*

A modulating brine-to-water heat pump with a thermal capacity of 12.46 kW and a COP of 4.29 at B0/W35 is selected according to the total heat demand. For the required supply temperature of 55 °C, this heat pump has a thermal power output ranging from 7.73 kW at -15 °C to 15.01 kW at 10 °C. This accounts for 98 % of the heat demand at the nominal outdoor temperature. The reason for selecting this heat pump model is because heat pumps larger than this would deliver more heat than it is required at -15 °C and this would oversize the PVT collector field as well and cause intermittent operation. On the other hand, heat pumps smaller than the selected heat pump would be insufficient to meet the heat demand.

Figure 8 illustrates the characteristics curves at various compressor speeds of the selected heat pump. The lowest temperature at evaporator inlet temperature is -20 °C. The heat pump can deliver 55 °C from the condenser for evaporator inlet temperatures above -15 °C. For the system, the supply temperature of the heat pump has to be 55 °C. Therefore, the cut-off temperature of the selected heat pump is -15 °C.

4.5.1 Dimensioning of PVT-Collectors

The performance of the PVT collector depends on various meteorological factors like irradiance, wind speed, ambient temperature, and longwave radiation. Figure 9 shows the illustration of heat transfer and the energy balance in a PVT collector. In order to incorporate all the factors affecting the PVT performance in the design process, the ISO 9806:2013 standard [20] is used to calculate the thermal output of the PVT collectors at design conditions. Eq. (17) shows the steady-state model for the calculation of the specific thermal power of a PVT collector at user-defined operating conditions. For the calculation, the thermal performance parameters available from the tests from the manufacturers based on the ISO 9806 standard are necessary and these parameters can be obtained from the manufacturer's datasheets. The datasheets for the PVT collectors contain the results from the tests that are carried out under Maximum Power Point (MPP) modes which consider the performance of both the electrical and thermal parts of the PVT collectors. Therefore, the effect of electrical output is already compensated by the parameters used.

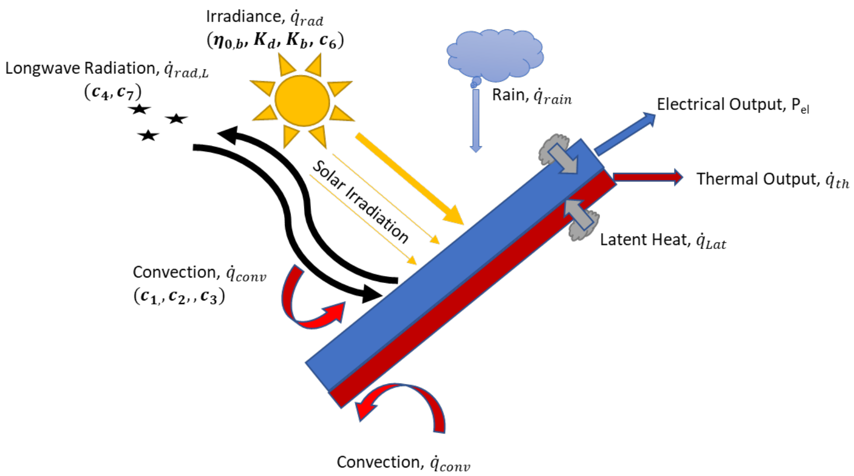

Figure 9. *Illustration of energy transfers in a PVT collector.*

$$\dot{q}_{th} = \eta_{0,b}K_b(\theta_L,\theta_T)G_b + \eta_{0,d}K_dG_d - c_1(\vartheta_m - \vartheta_a) - c_2(\vartheta_m - \vartheta_a)^2 - c_3u(\vartheta_m - \vartheta_a) + c_4(E_L - \sigma T_a^4) - c_6uG_{hem} - c_7u(E_L - \sigma T_a^4) - c_8(\vartheta_m - \vartheta_a)^4 \tag{17}$$

The more recent version of ISO 9806 standard i.e. ISO 9806:2017 suggests the use of u' = u-3 m/s instead of u in Eq. (17). However, the reduction in wind speed has not been considered in the model used in this methodology, as we corrected the coefficients. Therefore, it should be noted that the use of thermal parameters obtained from such models could lead to deviation in results. Additional effects of rain, condensation, and rear-side radiative heat exchange are neglected for now as the collector test parameters associated with these factors are not yet available and the ISO 9806 standard has not yet included these parameters in the mathematical model.

A specific day is selected from the historical weather data from *Deutscher Wetterdienst* based on the nominal ambient temperature [21]. This means that the daily average weather conditions, i.e. wind, irradiance, and longwave radiation, are selected from one of the days with nominal ambient temperature for the given location. The specific thermal output ($\dot{q}_{th}$) of PVT collectors in kW/m^2 on that day will be calculated based on the collector parameters and the average daily weather conditions using Eq. (17). Then, the total area of the PVT collectors will be estimated based on the calculated specific energy output and the required output from the heat pump source. Researches show that the PVT collectors have reduced external heat transfer coefficients due to the effects of frost or ice formation and lower internal heat transfer coefficients due to low viscosity while operating with heat pumps at very low inlet temperatures [3]. Therefore, a safety margin is used to dimension the PVT fields considering the periods below 0 °C. The parameter f_{safety}, which is the percentage by which the heat output from the heat source has to be increased to compensate for the energy lost during deicing as shown in Eq. (18). For single-family houses we recommend using 0.15 [3]. For large systems, we see interdependencies that cause reductions compared to parameters derived from lab testing [22], but further study is still necessary in this topic.

$$A_{PVT} = \frac{(1 + f_{safety}).\dot{Q}_Q}{\dot{q}_{th}} \tag{18}$$

Table 1. *Parameters for the calculation of PVT area.*

System characteristics	
Operation mode	Bivalent-Parallel
Temperature at bivalence point (only for bivalent systems)	-10 °C
Characteristics of selected heat pump	
Minimum temperature on evaporator	-15 °C
Performance @B-15W55	
COP	2.19
Capacity	7.73 kW
Performance @B-5W55	
COP	2.64
Capacity	8.81 kW
Weather conditions at design point	
Total irradiance	0 W/m^2
Wind Speed	1.3 m/s
Sky temperature	T_a - 20 °C (Clear sky conditions)
PVT Module	
Manufacturer	PVT_4 (See Figure 10 & 11)

The inputs for sizing the PVT collector and the backup heater for the selected building model are given in Table 1. The weather condition with an average wind speed of 1.3 m/s and clear sky without solar gains is taken into account considering the design points during night or instances when the collectors are covered with ice or snow. The PVT collectors are designed for various inlet temperatures at the evaporator of the heat pump or the outlet temperature of the PVT collectors. The heat pump characteristic curves are taken from the manufacturer to determine the heat pump performance at various temperature levels. The total heat output needed from the PVT collectors are determined based on Eq. (17), which resulted in 4.78 kW at -15 °C evaporator inlet temperature.

Based on the performance of the PVT collectors at lower operating temperatures and very extreme weather conditions without solar irradiance, four PVT collectors are selected for comparison. The collectors presented in Figure 10 are unglazed collectors without insulation on the rear sides, which are specially designed to operate under low ambient temperatures. The performance characteristics of the PVT collectors in weather conditions without solar irradiance, wind speed of 1.3 m/s, and sky temperatures 20 K lower than the ambient temperatures are shown in Figure 10. The figure shows the variation in the specific thermal power of PVT collectors based on the difference between the mean collector outlet temperature and the ambient temperature. The parameters from Solar Keymark Tests are considered to calculate the specific thermal output of the PVT collector based on Eq. (17).

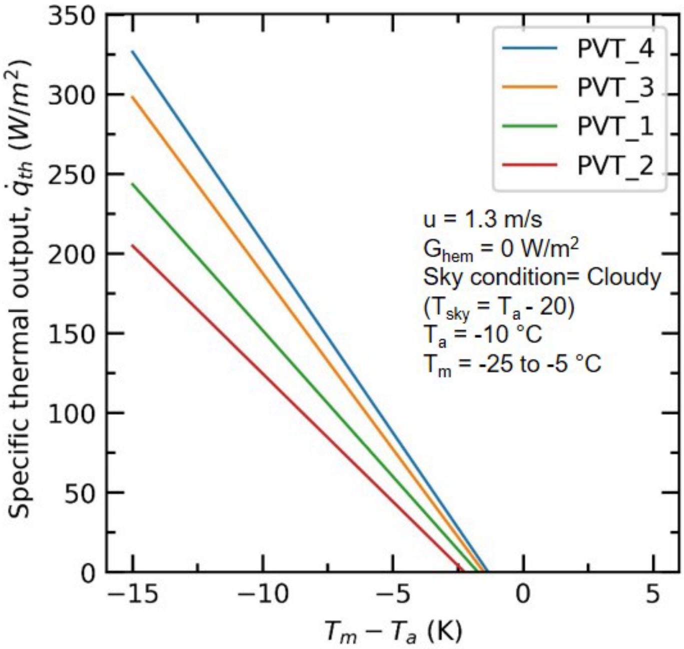

Figure 10. *Specific thermal power curves for various PVT.*

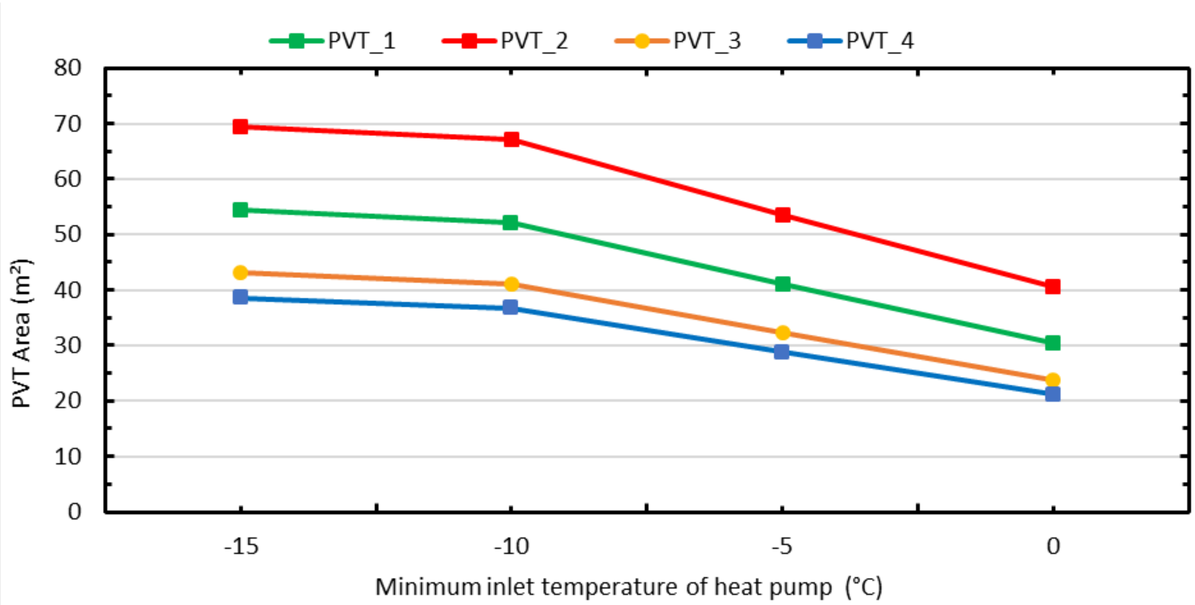

Figure 11. *Variation of area requirement for various PVT collectors with the inlet temperature.*

Figure 11 shows the sizes of PVT collectors necessary to meet the heat demand at the heat pump evaporator at various minimum inlet temperatures. The required PVT area reduces with the increase in minimum inlet temperature of the heat pump as the heat load also reduces. For the undistorted operation of heat pump, the maximum area of PVT required to meet the heat demand of the heat pump within its operating range has to be selected. PVT_4 is the most efficient among all for the operation under lower inlet temperatures of the heat pump and thus results in the lowest area requirements at the design point. The reason for that is due to the presence of fins as extended heat exchanger profiles and open rear side, which allows heat gains even during very low ambient temperature conditions. For the selected heat pump, PVT_4 modules with a total area of 38.63 m^2 are required for heat pump operation at -15 °C. At a heat pump inlet temperature of -15 °C, the selected heat pump cannot provide sufficient heat to the buildings, so a secondary heat source has to be designed in parallel.

4.5.2 Backup heater

In order to supplement the heat pump at times of peak heating demand or during times when the heat pump is not able to deliver required temperature levels due to very low inlet temperatures from the source, a backup heater is to be designed. In this methodology, the backup heater refers to the electric heating rod which is commonly used as a backup in heat pump systems. This is called mono-energetic operation. For bivalent-parallel operation, the size of the backup heater is determined based on Figure 5 as the difference in heat demand and the heat output of the heat pump at the design point. However, for bivalent-alternative operation, the backup heater is designed to fulfill the entire heat demand at the nominal ambient temperature i.e. $\dot{Q}_{H,N}(total)$ as shown in Figure 5.

$$\dot{Q}_{BH,AP} = \dot{Q}_{H,AP} - \dot{Q}_{HP,AP} \tag{19}$$

For the example building, the design point is the nominal ambient temperature. Therefore, the minimum size of the backup heater calculated as per Eq. (19) for the system is 0.2 kW. However, for safety, the selection of a backup heater could be made according to the maximum heat demand to assure heat supply even in case of breakdown.

5 Yearly simulations in TRNSYS

In the previous sections, a PVT-heat pump system is dimensioned alongside the dimensioning methodology. In this section, a yearly simulation is carried out in TRNSYS to analyze the performance of the selected PVT module in a heat pump system with similar dimensions obtained from the methodology. The area of the PVT collector used in the simulation is 38.63 m^2 and is similar to the PVT collector with fins on the backside explained in Chhugani et. al. [7]. Simulations were performed for the standard model for existing single-family house SFH100 with the TRNSYS setup described by Chhugani et. al. [13] for the location of Würzburg, Germany. The simulation results show that, the seasonal performance factor SPF_{bSt} (bSt=before storage) of the heat pump system given by Eq. (20) during the operation of the heat pump with the cut-off point of -15 °C is 3.7. The heat pump system works in bivalent-alternative mode with the backup heater operating at evaporator inlet temperatures below -15 °C. Throughout the year, the backup heater covers around 0.2 % of the heat demand, and the remaining demand is fulfilled by the heat pump. It is to be noted that the model used in the simulation does not allow the simulation of frost formation on the PVT, and defrosting is not necessary for most of the locations in Germany. However, a direct defrosting loop is added with the heat of the storage to avoid overestimation at very low temperatures [23]. Therefore, the negative energy called "defrosting" is considered.

$$SPF_{bSt} = \frac{\int (\dot{Q}_{HP} + \dot{Q}_{backup} + \dot{Q}_{PVT,defrosting}) \cdot dt}{\int (\dot{E}_{HP} + \dot{E}_{backup} + \dot{E}_{Pump,defrosting}) \cdot dt} \tag{20}$$

Through yearly simulations, it is found that the designed system meets the heat demand of the building, and the system efficiency is also plausible. Although it has been found that the selected PVT collector area is sufficient for the operation of the selected heat pump, it is important to note that the selected PVT dimensions obtained from the steady-state methodology may not be optimal.

Therefore, TRNSYS simulations with different PVT areas are carried out to investigate whether the dimensioned area is optimal in terms of system efficiency. The minimum inlet temperature at the heat pump evaporator is taken as -15 °C. Figure 12 shows the variation of the seasonal performance factor (SPF_{bSt}) and electrical consumption of backup heater with the variation of the area of PVT collectors. From technical point of view, there is not an optimum, as the efficiency is growing monotone with PVT area. Above the area of 30 m^2, the SPF does not increase significantly. The figure also illustrates that the electrical consumption of the backup heater decreases significantly with the increase in area up to 22 m^2. An increase in area of more than 22 m^2 causes a relatively lower reduction of the electrical consumption and the impact is very low for the increase in area above 30 m^2. These results show that the PVT collector area obtained using the methodology gives reasonable results.

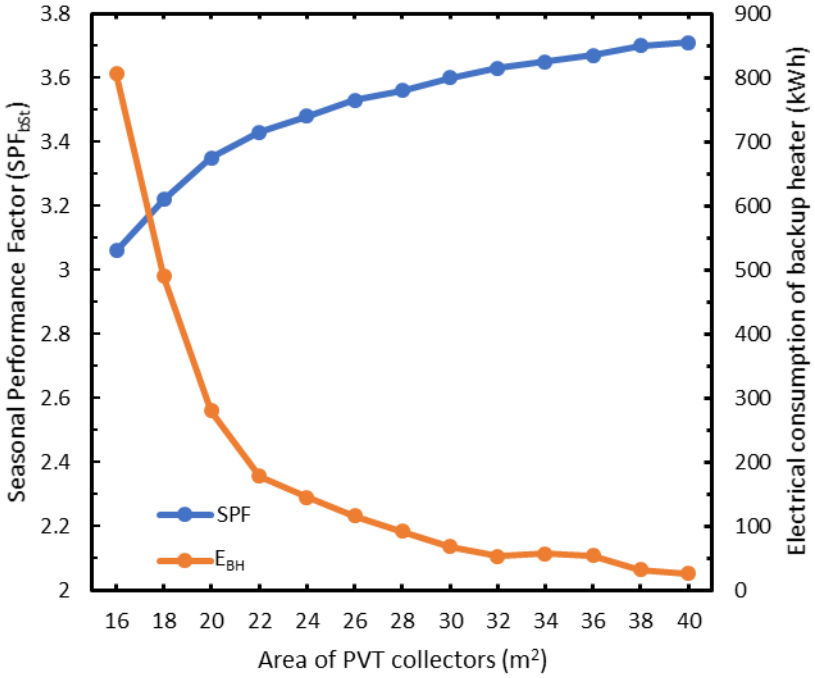

Figure 12. *Variation of SPF and electrical consumption of backup heater with area of the selected PVT.*

6 Conclusion

This paper describes a steady-state methodology for dimensioning a heat pump system with PVT collectors as a single heat source. The methodology is based on the German guideline VDI 4645 to size the heat pump and the dimensioning of the PVT area is an extension of the guideline which implements the calculation procedure of the ISO 9806 standard. The dimensioning methodology has been used to dimension a heat pump system for an existing single-family house model SFH 100 for the location of Würzburg, Germany. Based on the heat demands for space heating and domestic hot water preparation, a heat pump is selected from the manufacturer's specification, and the PVT collectors are sized according to the heat pump characteristic curve. For comparison, four different PVT collector models have been considered. The dimension of PVT collectors at various heat pump inlet temperatures is determined based on the thermal performance of the respective PVT collectors at design conditions. The PVT collector resulting in the lowest area requirement is selected. The resulting dimensions are investigated through an annual simulation in TRNSYS for the same building model, system parameters, and boundary conditions for heating and DHW preparation. The results show that the selected PVT collectors' area and heat pump size for the house sufficiently meet the heat demand for the building. The seasonal performance factor SPF_{bSt} obtained from the simulation is 3.7, which is considered good for the PVT-heat pump systems. The transient simulations of a similar system with different PVT areas show that the steady-state design method explained in this paper gives reasonable results. The optimum sizing of the PVT area would require economic considerations.

7 Discussion

PVT collector is a very sensitive source to design. Its performance varies significantly with a small variation in design conditions and the choice of thermal performance parameters. The plausibility of using Solar Keymark Test parameters for the calculation of heat flow of the PVT collectors is still questionable as the parameters are calculated based on the data from the weather conditions that vary drastically with the design conditions explained in this paper. Also, the parameters do not include the effects of condensation and freezing.

In future work it is planned to improve the design method for PVT-heat pump systems for large multi-family houses, considering safety margins dependent on field size and installation, and simultaneity factors for many apartments. Furthermore, the inclusion of solar irradiation in dimensioning PVT collectors will be investigated to obtain the optimal design conditions. This would also take into account the dimensioning of various types of PVT collectors in the market whose thermal gains are mainly from solar irradiation. This would also cover the systems consisting of heat storages to store the heat gains during the day for use at night. The methodology described in this paper has already been implemented in a web-based tool and is under the stage of testing and further development.

Acknowledgment

The presented work and the results are part of the research project “integraTE - 03EGB0023”, funded by the Federal Ministry of Economics and Climate Protection (BMWK) as well as the State of Lower Saxony and carried out in cooperation with Fraunhofer-ISE and IGTE of Stuttgart University. Furthermore, we would like to thank our industry partners nD-System GmbH, Architektur- und TGA-Planungsbüro Carsten Grobe, PA-ID Prozess GmbH, TripleSolar BV, SunMaxx PVT GmbH, Consolar Solare Energiesysteme GmbH and Splus2 GmbH for their financial support. The responsibility for the content of this publication lies solely with the authors.

Competing Interests

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Author Contributions

The contributions of each author to this paper have been assessed according to the CreDIT guidelines. The specific contributions of each author are as follows:

Krishna Timilsina: Conception of the methodology, literature review, analysis of results and drafting of the manuscript.

Bharat Chhugani: Simulation and review of manuscript.

Harshvadan Modi: Implementation of the methodology and review.

Peter Pärisch: Conception of the methodology, critical revision of manuscript, and final approval of the version to be published.

All authors read and approved the final manuscript.

References

[1] Bundesverband Wärmepumpe (BWP), 'Branchenstudie 2023: Marktentwicklung – Prognose – Handlungsempfehlungen'. Bundesverband Wärmepumpe (BWP) e. V., 2023.

[2] Statista, 'Annual sales of brine-to-water heat pumps in Europe from 1990 to 2020 with forecasts until 2030'. Accessed: Jan. 29, 2024. [Online]. Available: https://www.statista.com/statistics/1314212/brine-to-water-heat-pump-sales-in-europe/

[3] M. Littwin, C. Lampe, M. Kirchner, B. Chhugani, P. Pärisch, and F. Giovannetti, 'TwinPower: Integrierte Gesamtenergieversorgung von Wohngebäuden mit PV-thermischen Kollektoren als bisolare Wärmepumpenquelle', Institut für Solarenergieforschung Hameln GmbH (ISFH), Abschlussbericht, 2020.

[4] D. Zenhäusern, E. Bamberger, and A. Baggenstos, 'Energiesysteme mit Photovoltaisch-Thermischen Solarkollektoren', Institut für Solartechnik SPF, HSR Hochschule für Technik Rapperswil, Switzerland, Schlussbericht PVT Wrap-Up, Mar. 2017.

[5] U. Leibfried, S. Asenbeck, S. Fischer, C. Twerdy, and U. Ruoff, 'Abschlussbericht "SOLINK": Hocheffiziente, auf intelligenter Verknüpfung von PVT- und Wärmepumpentechnik basierende Wärmeversorgung für Gebäudebestand und Neubau', 2020.

[6] Consolar, 'SOLINK: Die Energiequelle für Wärmepumpen, Technische Dokumentation'. Consolar Energiesysteme, 2022.

[7] B. Chhugani, P. Pärisch, S. Helmling, and F. Giovannetti, 'Comparison of PVT - heat pump systems with reference systems for the energy supply of a single-family house', *Solar Energy Advances*, vol. 3, p. 100031, 2023, doi: 10.1016/j.seja.2022.100031.

[8] VDI 4645, 'Planung und Dimensionierung von Heizungsanlagen mit Wärmepumpen in Ein- und Mehrfamilienhäusern'. VDI-Gesellschaft Energie und Umwelt (GEU), 2023.

[9] Stiebel Eltron, 'Engineering and Installation of Heat Pumps'. 2013.

[10] Viessmann, 'Heat pump principles'. 2020.

[11] R. Dott, M. Y. Haller, J. Ruschenburg, F. Ochs, and J. Bony, 'The Reference Framework for System Simulations of the IEA SHC Task 44 / HPP Annex 38, Part B: Buildings and Space Heat Load', 2013.

[12] Bundesverband Wärmepumpe e.V., 'Klimakarte', Bundesverband Wärmepumpe e.V. Accessed: Jan. 20, 2024. [Online]. Available: https://www.waermepumpe.de/normen-technik/klimakarte/

[13] B. Chhugani, P. Paerisch, and F. Giovannetti, 'Decarbonizing Heating Supply Systems in Existing Single-family Houses Through PVT - Heat Pump Systems', presented at the EuroSun, Kassel, Germany, 2022.

[14] DIN EN 12831-1, 'Energy performance of buildings - Method for calculation of the design heat load Part 1: Space heating load, Module M3-3'. DIN Deutsches Institut für Normung e V, 2017.

[15] T. Loga, B. Stein, N. Diefenbach, and R. Born, Eds., *Deutsche Wohngebäudetypologie: beispielhafte Maßnahmen zur Verbesserung der Energieeffizienz von typischen Wohngebäuden*, 2., erw. Aufl. Darmstadt: IWU, 2015.

[16] DIN EN 12831-3, 'Energy performance of buildings – Method for calculation of the design heat load – Part 3: Domestic hot water systems, heat load and characterization of needs, Module M8-2, M8-3'; DIN Deutsches Institut für Normung e. V., 2021.

[17] D. Fischer, K. B. Lindberg, H. Madani, and C. Wittwer, 'Impact of PV and variable prices on optimal system sizing for heat pumps and thermal storage', *Energy and Buildings*, vol. 128, pp. 723–733, Sep. 2016, doi: 10.1016/j.enbuild.2016.07.008.

[18] M. Y. Haller, R. Dott, J. Ruschenburg, F. Ochs, and J. Bony, 'The Reference Framework for System Simulations of the IEA SHC Task 44 / HPP Annex 38, Part A: General Simulation Boundary Conditions Report C1 Part A', 2013.

[19] R. Vollmer, J. Wapler, and S. Hess, 'Quellen-Verfügbarkeit für Wärmepumpen-Systeme in Mehrfamilienhäusern', Bericht zu AP 2.1 (LowEx-Bestand Analyse), 2019.

[20] ISO 9806, 'Solar energy — Solar thermal collectors — Test methods'. ISO, 2013.

[21] Deutsche Wetterdienst, 'Open Data'. Accessed: Jan. 20, 2024. [Online]. Available: https://opendata.dwd.de/climate_environment/CDC/observations_germany/climate/daily

[22] K. Timilsina, P. Paerisch, and B. Chhugani, 'In-situ monitoring of a PVT-heat pump system with ground sources used for heating and cooling applications', presented at the Solar World Congress, New Delhi, 2023.

[23] B. Chhugani, P. Paerisch, M. Kirchner, M. Littwin, C. Lampe, and F. Giovannetti, 'Model Validation and Performance Assessment of Unglazed Photovoltaic-Thermal Collectors with Heat Pump Systems', in *Proceedings of the ISES EuroSun 2020 Conference – 13th International Conference on Solar Energy for Buildings and Industry*, Online: International Solar Energy Society, 2020, pp. 1–12. doi: 10.18086/eurosun.2020.05.13.