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Waste Heat Recovery by Air-to-Water Heat Pump from Exhausted Ventilating Air for Heating of Multi-Family Residential Buildings

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Abstract: The paper presents an analysis of the application of an air-to-water electric compressor heat pump (AWHP) for the recovery of waste heat from the exhaust air in a typical multifamily residential building and the use of this heat for space heating, as well as the impact of this solution on the building energy performance (the *PPR* index). Simulations were performed in TRNSYS for five locations in Poland (Koszalin, Wrocław, Lublin, Białystok, Suwałki), for various heating system parameters (80/60 °C, 75/65 °C, 70/50 °C, 55/45 °C, 35/28 °C), for various temperature limitations of heat pump operation. It was shown that the analyzed system has great potential from an energy and environmental point of view. It can provide significant benefits in terms of the energy performance of the building, depending on the system parameters. The results show that the most energy-efficient system is the one with the lowest heating system temperatures. Moreover, implementing a temperature limitation on the heat pump operation improves its efficiency, but the higher the design parameters of the heating installation and the lower the limitation, the lower the heat pump contribution, and the higher the *SCOP* and the *PPR*. The energy effect is also influenced by location, but its scale depends on the parameters of the heating system and the temperature limitation of the heat pump's operation. It is more significant for lower heating system parameters. This system enables the possibility of further reducing the demand for nonrenewable primary energy by powering the heat pump with photovoltaic cells.

Keywords: TRNSYS; air-to-water electric heat pump; AWHP; hybrid heat source; waste heat; building energy performance; primary energy; exhaust ventilation; energy efficiency

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1. Introduction

Waste heat recovery is essential in many industries. For example, waste heat is recovered in industrial installations [1], in data centers [2], in flue gas installations [3–5] from air conditioning units [6], from expanding natural gas [7,8], from cooking and wastewater [9], and in thermal power plants to recover low-grade heat from turbine exhaust steam [10].

Heat pumps offer a wide range of possibilities for recovering low-temperature heat and transferring it to higher-temperature media. Their various applications are a current topic of discussion in the literature.

1.1. Field of Research of Heat Pumps

In [11], using Energy Plus software, a pump model was developed, taking into account temperature and load levels and was shown to be consistent with the manufacturer's data. In [12], five different constructions of two-stage heat recovery heat pumps were compared on the basis of mathematical models. In [13], an analysis of the operation of an air source heat pump with variable refrigerant flow and vapor injection in cold regions is presented. In [14], the cooperation of heat pumps with PCM-type materials is described. In [15,16], the topic of heat pumps in gray wastewater heat recovery systems was discussed. In [17], the use of heat pumps for heating water in swimming pools was analyzed, and in [18] it was analyzed for fish farming purposes. The studies presented in [19,20] examined the influence of the climate of various cities on the results of calculations of the seasonal energy performance of air heat pumps. In [21], CFD analysis of a field-type ground heat exchanger and its influence on compressor heat pump performance was presented. Another study [22] concerns the technical, legislative, and ecological aspects of the use of compressor heat pumps, particularly considering the possibility of eliminating certain refrigerants. In [23], a review of the solutions for the cooperation of heat pumps with solar collectors, PV photovoltaic cells, and PVT thermal photovoltaic cells is presented. In [24], the cost of cooperation of the air source, heat pump, and photovoltaic was analyzed. In [25], the effect of the partial load of a heat pump on its efficiency was analyzed. A neural network model based on the performance data of the device and coupled with a building model was used as a platform to determine the appropriate parameters of the automatic control system to reduce the number of on/off cycles and increase the COP by 14.6% and reduce energy consumption by 18.9%. In [26], a dynamic simulation of a hybrid heat source for a building, coupling a heat pump and a gas boiler, was carried out. Based on these simulations, primary energy savings were demonstrated concerning the use of the heat pump alone at 6% and the boiler alone at 22%. In [27], six different building heating systems, including electric heat pumps with an electric heater and absorption gas heat pumps combined with a gas boiler, were compared in terms of primary energy consumption and CO_2 emissions. Although the results are sensitive to the primary energy factor, electric heat pumps have been shown to be the most promising technology.

This topic is particularly important due to the constantly tightening regulations concerning the energy performance of buildings [28,29] and the need for renovations of buildings [30,31]. Heating needs mainly consist of space heating (heat losses by transmission and ventilation losses) and the preparation of domestic hot water. Heat losses by transmission are dissipated, but heat for the heating of ventilation air and domestic hot water can be partially recovered. This article deals with heat recovery from ventilation air by means of a heat pump.

1.2. Heat Recovery from Ventilation Air

One of the main heat recovery issues that has been discussed in the literature for many years is heat recovery in ventilation systems that consume significant amounts of energy [32–45]. Currently, the use of heat recovery from exhaust air is mandatory in Poland for ventilation systems with an airflow rate of 500 m^3/h or more [29]. This requirement applies only to supply and exhaust mechanical ventilation systems (MVHR), although, in practice, the growing need to reduce the energy intensity of buildings is forcing it to be applied to systems with smaller flows. In mechanical supply and exhaust installations, it is possible to implement it with different technical solutions. The heat can be transferred from the exhaust air to the outside air via cross-flow heat exchangers, counter-flow heat exchangers, regenerative rotary heat exchangers, and accumulation heat exchangers, as well as heat recovery systems with heat pipes and with an intermediate medium (as well as with the use of heat pumps).

However, it is much more difficult to realize the heat recovery from exhaust air in exhaust ventilation (EV) systems, which is the subject of this article. This is due to several factors:

- Warm air is removed through diffuse exhaust ducts;
- Exhaust duct outlets are usually located above the roof of the building, often far from the heat source in the building;
- The outside air is supplied through leaks or openings (vents, diffusers) distributed in the external walls of the building.

Consequently, heat transfer from the exhaust air to the external supply air cannot be carried out using cross-flow heat exchangers, counterflow heat exchangers, rotary heat exchangers, etc., which are used in supply and exhaust air handling units.

A lack of heat recovery from exhaust air and problems with its application in Poland occur in:

- Existing and currently built multi-family residential buildings, in which exhaust ventilation (EV) has been and is still installed, and which represent a significant part of residential buildings in Poland, at 42.5% [46], and in Europe, at 42% [47];
- Other types of existing buildings with mechanical supply and extract ventilation (MSEV), which were not covered by successive changes in regulations concerning the requirement of heat recovery [29,32].

The implementation of heat recovery from exhaust air has been considered in the literature, among others, via:

- The installation of a central supply and exhaust mechanical system with heat recovery [39,40];
- Installation in external walls, separate for each room, operating alternately as supply air and exhaust air, pairs of distributed ventilation devices with heat recovery [43,44];
- The installation of an air heat pump on the exhaust ventilation system [39–42] and using it for central heating and/or domestic hot water preparation.

In this paper, the authors consider the latter solution for building heating. The idea is presented in Figure 1.

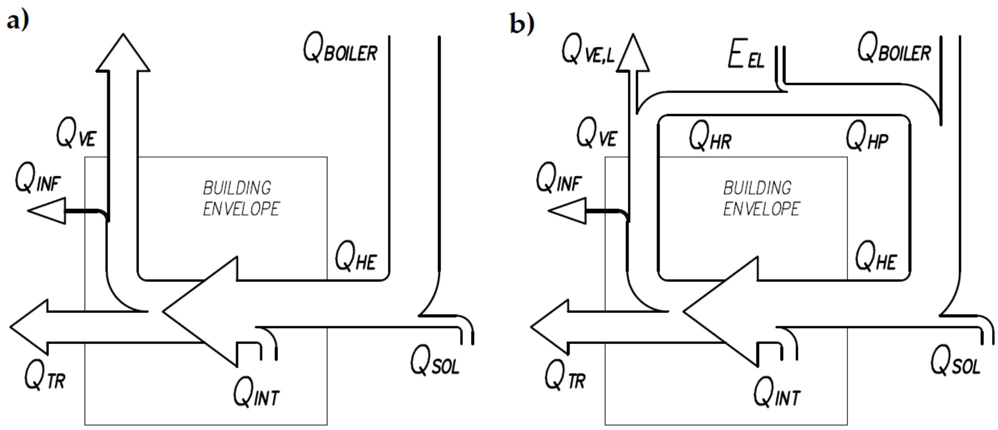

Figure 1. Comparison of energy balance of a building with exhaust ventilation (**a**) without heat recovery and (**b**) with heat recovery using an electric compressor heat pump and its use for space heating. EL—electric; HP—heat pump; HE—heating; SOL—solar gains; INT –internal gains; TR—transmission; INF—infiltration; VE—ventilation; VE,L—ventilation loss; HR—heat recovery.

1.3. Exhaust Air Heat Recovery with Heat Pump

The use of heat pumps to recover heat from exhaust air through ventilation systems is also the subject of many studies and publications. For example, [48] describes the use of a heat pump as an additional heat recovery system in a supply and exhaust ventilation system. However, from the point of view of the topic discussed in this paper, publications [32,39–42] are particularly interesting, as they concern the use of a heat pump for heat recovery from exhaust air in exhaust ventilation systems. Furthermore, publications [32,39,40,42] concern exhaust ventilation systems in residential buildings.

In [32], 60 existing ventilation systems using heat pumps for heat recovery from exhaust air and flue gases in various system configurations for different purposes were analyzed. On average, the studied installations reduced primary energy consumption by 19.4% and CO_2 emissions by 18.4%.

According to [39], two ventilation systems are currently used in Estonia for the renovation of multi-family buildings: HRV supply and exhaust ventilation with heat recovery, and EAHP exhaust ventilation with electric air-source heat pump. Using IDA-ICE 4.7 software, HRV and different schemes for combining EAHP with a dual-function parallel district heating substation were compared. It was found that the use of EAHP practically eliminates the consumption of district heat for domestic hot water in summer and covers about 60% of space heating needs. However, compared to HRV, it negatively affects the district heating network because it raises the return temperature, lowering the efficiency of the heat source, and eliminates the consumption of district heat in summer when it is "free" in CHP systems.

In [40], based on the measurement of energy consumption before and after renovation in 2015 and 2018, three packages of renovation measures for multi-family buildings in Sweden, containing different types of ventilation, were compared from an economic point of view: HRV; EV (exhaust ventilation) without heat recovery, with flow limitation and pressure controllers; and EAHP with flow limitation and pressure controllers. The EAHP system had the lowest energy consumption, especially in terms of heat.

The relatively low temperature of air exhausted from rooms in winter (usually about 20–24 °C), makes the direct use of this heat for building heating or hot water preparation technically impossible. The water temperature required for space heating can range from 30 °C to 80 °C, and for hot water heating this can be higher than 60 °C under Polish conditions.

In addition, when an air heat pump is used with exhaust air as the lower heat source, its efficiency will be much higher than the standard operation with outdoor air as the lower heat source [41]. The smaller the temperature difference between the lower and upper source of the heat pump, the higher the energy efficiency factor of the heat pump [11]. An additional advantage of the system is that the evaporator can be defrosted with warm air, without the need for defrosting procedures in the refrigeration circuit. Therefore, in [42], the authors addressed the possibility of recovering heat from the exhaust air using an electric heat pump and using the recovered heat for hot water preparation. Analyses of such a system were carried out in TRNSYS 17 software for an example of a multi-family building heated by a gas boiler room, located in Poland.

1.4. Research Gap

Based on the literature review, it was found that the topic of heat pumps and their various applications are widely discussed in the literature. The subject of heat recovery from the exhaust air is also frequently addressed, but mainly in mechanical supply and exhaust ventilation systems. However, the subject of heat recovery from exhaust air in exhaust ventilation systems in multi-family buildings with the use of air-source heat pumps is discussed in few publications. This solution is particularly popular in the northern part of Europe. However, the analyses presented in the literature mainly concern the cooperation of such a solution with a district heating substation and network in cold climates (Sweden, Estonia). In Poland, especially in the suburbs of agglomerations, many multifamily buildings heated with local gas boiler plants have been built and are still being built in recent years. Therefore, the combination of a heat pump with exhaust air as the lower source is also particularly relevant and interesting. In the literature, we found publications concerning hybrid sources, combining heat pumps with gas boilers. However, there were no publications concerning hybrid sources combining gas boilers with heat pumps using air from the exhaust ventilation system, except for [42]. The authors of this paper have already presented an analysis of an electric air source heat pump with exhaust air as a lower source in cooperation with a gas boiler, but only for domestic hot water heating [42]. It was shown that in comparison with the use of a gas condensing boiler alone, the primary energy consumption for DHW preparation was reduced by 9.1%. However, there is a literature gap in analyses of such combinations for building heating purposes. This is particularly important because, under Polish conditions, space heating represents about 60% of the heating needs of new residential buildings. We found no analyses of the influence of the type and parameters of the heating system on the effectiveness of such a solution and the energy performance of the building, especially in the climatic conditions of Poland, which represents an area with the central European temperate climate.

1.5. The Aims of the Paper and the Novelty of the Work

The research aims of this study are:

- To determine the energy efficiency of the considered system;
- To analyze the impact of the considered system on the energy performance of the building; and
- To determine the areas of reasonable application of the analyzed solution,

depending on climate conditions in various types of heating systems.

The novelty of the paper is an analysis of the application of an electric compressor air-water heat pump:

- For the recovery of waste heat from the ventilation exhaust air;
- In a typical multi-family residential building;
- Using recovered heat for the heating of the building;
- In cooperation with a condensing gas boiler, in terms of the energy performance of the building.

A diagram of the energy flow of the discussed solution in comparison to the installation without a heat pump is presented in Figure 1.

2. Materials and Methods

2.1. Climatic Data

The analyses presented in this paper were carried out for five cities located in Poland. Poland is divided into five climate zones (I, II, III, IV, and V) [49], according to the value of the outdoor air temperature that was assumed for the heating design of the buildings in each zone, at −16 °C, −18 °C, −20 °C, −22 °C, and −24 °C, respectively. Therefore, one city for each zone was selected for the analysis: I—Koszalin, II—Wrocław, III—Lublin, IV—Białystok, V—Suwałki. The climatic data published in [50], used in Poland to determine the energy performance of buildings, were used. Table 1 lists the minimum, average, and maximum values and Figure 2 compares the frequency of outdoor air temperatures in each location, excluding the months from June to September.

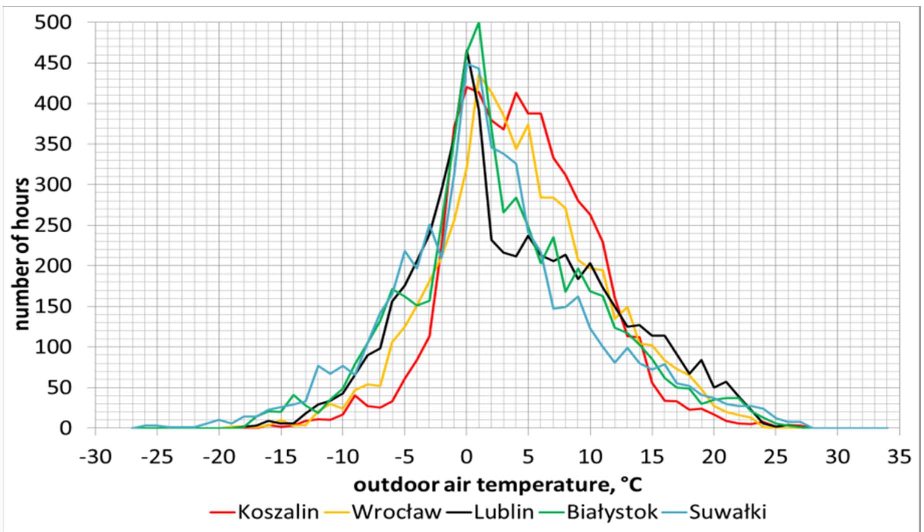

Figure 2. Comparison of the frequency of outdoor air temperatures at the analyzed locations, excluding the months of June to September, based on [50].

Table 1. Outdoor air temperatures depending on location, based on [50].

Location	Outdoor Temperatures from October to May			Whole Year Average Outdoor Temperature
	Minimum	Maximum	Average	
-	°C	°C	°C	°C
Koszalin	−16.5	27.1	4.5	8.0
Wrocław	−18.8	24.0	4.2	8.2
Lublin	−17.7	26.5	3.8	7.8
Białystok	−17.6	26.9	2.9	6.9
Suwałki	−26.2	27.3	2.1	6.4

2.2. Description of the Analyzed Heating System

A diagram of the analyzed system is shown in Figure 3. Air is exhausted from the apartments by exhaust fans installed on the collector air ducts. It is cooled by a heat pump evaporator, before being exhausted to the outside. The heat captured in the evaporator from the exhaust air is transported in the heat pump circuit to the condenser and is used to heat the heating medium of the space heating system. The heating medium heated by the heat pump is transported by the heat pump from the lower part of the buffer storage tank, and after being heated, is transferred to the upper part of the buffer. The heating medium from the buffer is transported to the radiators using a circulation pump. If the water temperature in the storage tank is too low for the heating system, it is heated by an additional heat source (in this case a gas boiler). If the water temperature is too high, it is lowered using a three-way valve with an actuator through cold mixing. The way the heat pump, gas boiler, and heating system are connected makes it possible to apply the heat pump in an existing boiler room and to analyze its cooperation with a heating system with different supply and return temperatures of the medium.

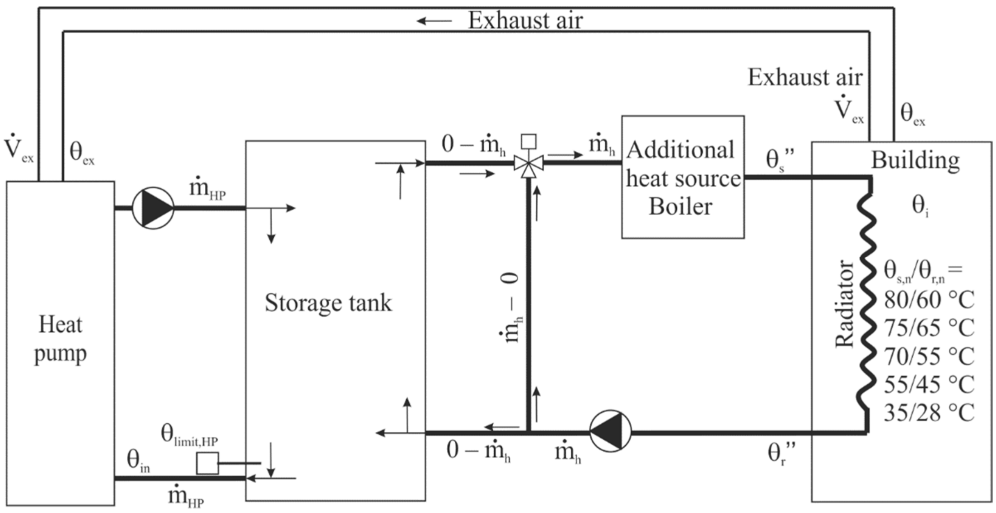

Figure 3. Diagram of the analyzed heating system.

2.3. Building and System Model

The model of the building and the analyzed heating system were developed using TRNSYS 17 software [51] (Figure 4). The assumptions for the simulations performed using this system are presented below.

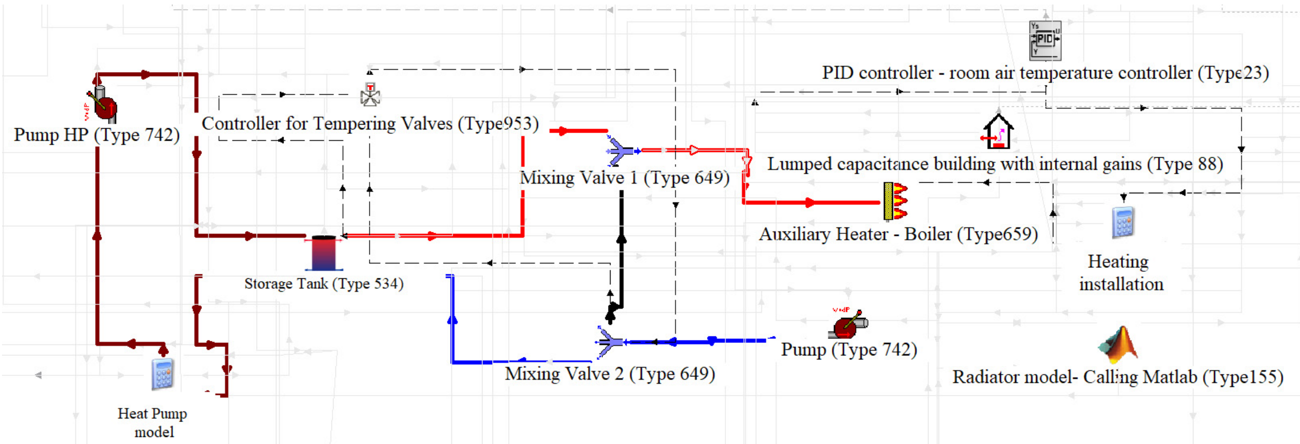

Figure 4. Main components of the heating system and building model in TRNSYS 17 [51] (part of the program code).

2.3.1. Building

The assumptions for the analyzed building were as follows:

- A 5-storey building, 3 apartments per storey. One air zone with a volume of 2220 m^3, an area of 888 m^2, heat capacity of 240,882 kJ/K, heat transfer coefficient by conduction H_{tr} equal to 441 W/K, and heat load of the building at the design outdoor air temperature as in Table 2.
- Controlled temperature in the building equals 20.3 °C (setpoint).
- Collective mechanical exhaust ventilation with fans running continuously. Extract air ducts in the insulation shield of the building. The ventilation airflow supplied to the rooms directly from the outside equals $\dot{V}_{ex} = 1500$ m^3/h; the airflow infiltrating through leaks at 80 m^3/h. Exhaust airflow from each apartment 100 m^3/h, with an average temperature of the exhaust air of $\theta_{ex} = 22$ °C (20 °C—kitchen, 50 m^3/h; 24 °C—bathroom, 50 m^3/h).
- Heat gains from solar radiation and internal gains from occupants according to the schedule (Table 3) adopted from PN-EN ISO 13790:2009 [52].

Table 2. Designated outdoor temperatures and heat loads of the building.

Location	Designated Outdoor Air Temperature [49]	Heat Load of the Building
-	°C	kW
Koszalin	−16	35.5
Wrocław	−18	37.5
Lublin	−20	39.4
Białystok	−22	41.4
Suwałki	−24	43.3

Table 3. Internal gains, determined based on [52].

Days	Hours	Living Room Plus Kitchen	Other Conditioned Areas	Average
		q_{int}, W/m^2	q_{int}, W/m^2	q_{int}, W/m^2
Monday to Friday	07.00 to 17.00	8	1	3.8
	17.00 to 23.00	20	1	8.6
	23.00 to 07.00	2	6	4.4
Saturday to Sunday	07.00 to 17.00	8	2	4.4
	17.00 to 23.00	20	4	10.4
	23.00 to 07.00	2	6	4.4

2.3.2. Central Heating System

The heat is delivered to the building by the heating system with a capacity ranging from 0 to the value of the design heat load, depending on the location of the building (Table 2), required to maintain the assumed air temperature in the building. The current supply temperature setpoint depends on the current outdoor temperature and is derived from heating curves which are different according to different system design parameters. Heat losses from the pipes of the heating system have been neglected.

The simulations were carried out variably for different design parameters of the heating installation, characteristic for different types of installation and/or heat source:

- 80/60 °C—typical for the heat substation and heating system with radiators,
- 75/65 °C—standard according to PN-EN 442 [53],
- 70/50 °C—typical for a gas boiler and heating system with radiators,
- 55/45 °C—typical for a condensing gas boiler and heating system with radiators,
- 35/28 °C—typical for floor heating.

The theoretical current supply and return temperature θ_s' and θ_r' of the heating system were determined as a function of the current outdoor temperatures θ_e' based on the heating curve equations:

$$\theta_s' = \theta_i + \left(\frac{\theta_s - \theta_r}{2} - \theta_i\right) \cdot \varphi^{1/n} + \frac{\theta_s - \theta_r}{2} \cdot \varphi \tag{1}$$

$$\theta_r' = \theta_i + \left(\frac{\theta_s - \theta_r}{2} - \theta_i\right) \cdot \varphi^{1/n} - \frac{\theta_s - \theta_r}{2} \cdot \varphi \tag{2}$$

where:

θ_s design supply temperature of the heating system, °C
θ_r design return temperature of the heating system, °C
θ_i design room air temperature, which equals 20 °C

where the relative heat load:

$$\varphi = \frac{\theta_i - \theta_e'}{\theta_i - \theta_e} \tag{3}$$

where:

θ_e design outdoor air temperature (Table 2), °C
θ_e' current outdoor air temperature, °C.

The current heat transferred to the room by the radiator was determined thus:

$$\dot{Q}'' = \dot{Q}_n \cdot \left(\frac{\Delta\theta''}{\Delta\theta_n}\right)^n \tag{4}$$

where:

n the exponent of the thermal characteristics of the radiator, for the parameters 35/28 °C, assumed as for the floor heating, 1.10; for the other parameters, assumed as for the panel radiators, 1.3,
$\dot{Q}_n$ standard radiator output at standard heating medium temperatures $\theta_{s,n}$ and $\theta_{r,n}$ and standard room air temperature $\theta_{i,n}$, kW

where the nominal logarithmic mean temperature difference:

$$\Delta\theta_n = \frac{\theta_{s,n} - \theta_{r,n}}{ln\left(\frac{\theta_{s,n} - \theta_{i,n}}{\theta_{r,n} - \theta_{i,n}}\right)} \tag{5}$$

where:

$\theta_{s,n}$ catalog supply temperature of the heating system, np.: 75 °C
$\theta_{r,n}$ catalog return temperature of the heating system, np.: 65 °C
$\theta_{i,n}$ designated room air temperature, assumed as 20 °C.

The current logarithmic mean temperature difference can be expressed as

$$\Delta\theta'' = \frac{\theta_s'' - \theta_r''}{ln\left(\frac{\theta_s'' - \theta_i''}{\theta_r'' - \theta_i''}\right)} \tag{6}$$

The current heating system supply temperature θ_s'' results from the building's current heating needs, associated with maintaining a controlled temperature in the building of 20.3 °C. The current supply temperature is determined based on the PID controller indications of the room air temperature and the inequality is satisfied thus:

$$\theta_i'' \leq \theta_s'' \leq \theta_s' \tag{7}$$

where:

θ_s' theoretical current supply temperature of the heating system, °C
θ_s'' current supply temperature of the heating system (from the boiler or the mixing valve outlet), °C
θ_i'' current resulting indoor air temperature (from building model), °C.

The resulting current return temperature is calculated by iterating on the basis of the instantaneous heat balance:

$$\theta_r'' = \theta_s'' - \frac{\dot{Q}''}{c_w \cdot \dot{m}_h} \tag{8}$$

where:

θ_r'' current return temperature of the heating system (from the boiler or the mixing valve outlet), °C
c_w specific heat of water, kJ/(kg·K).

Constant (for a given location and system parameters) heating medium flow rates were determined from the relationship (with results shown in Table 4):

$$\dot{m}_h = \frac{\dot{Q}_n}{c_w \cdot (\theta_{s,n} - \theta_{r,n})} \tag{9}$$

Table 4. The heating medium flow for particular building locations and heating system parameters.

Design Parameters of Central Heating Installation, θ_s/θ_r	**Flow through Central Heating Installation Adjusted to Design Heat Load and Parameters of Central Heating Installation, $\dot{m}_h$**				
	Koszalin	**Wrocław**	**Lublin**	**Białystok**	**Suwałki**
°C	**kg/h**	**kg/h**	**kg/h**	**kg/h**	**kg/h**
80/60	1560	1640	1730	1810	1900
75/65	3110	3290	3450	3630	3800
70/55	2070	2190	2300	2410	2520
55/45	3090	3270	3430	3600	3770
35/28	4390	4630	4870	5120	5350

2.3.3. Additional Heat Source (Boiler)

The role of the boiler in the analyzed system is to heat the heating water to the current setpoint supply temperature θ_s''. To ensure the highest possible contribution of the heat pump, a limit was introduced for the operation of the additional heat source (the boiler) depending on the outdoor air temperature. Values of the outdoor air temperature limit $\theta_{lim,boil}$ for particular parameters of the heating system and building location are given in Table 5. Hysteresis of ±1 °C has been taken into account, which means that in the example of Koszalin and parameters of 80/60 °C, above 7.5 °C no boiler (additional heat source) operation is allowed, whereas below 5.5 °C it is.

Table 5. Limits of the outdoor air temperature for the operation of an additional heat source.

Design Parameters of the Central Heating System, θ_s/θ_r, °C	Limits of the Outdoor Air Temperature for the Operation of an Additional Heat Source, °C
80/60	6.5
75/65	6.5
70/55	6.0
55/45	6.0
35/28	5.0

The typical bivalent point value for an air source heat pump under Polish climate conditions is between −10 °C and −5 °C. This value is typical when the heat pump operates on outdoor air. In this case, the lower source of the heat pump is the indoor air extracted from the building; thus, it has a much higher temperature than the outdoor air temperature. In addition, with an air pump running on the outdoor air, the bivalent point indicates the temperature of the lower source (outdoor air) below which the additional heat source must be turned on because the heating capacity of the heat pump itself is insufficient. In contrast, in the case studied, the values shown in Table 5 are for the outdoor air temperature above which the additional heat source should not operate, and the heat pump covers the entire heat demands of the building to ensure the highest possible heat pump contribution. The values in Table 5 were determined taking into account the heat gains in the building.

Additionally, a limitation of the boiler operation time outside the heating season was applied: hysteresis depending on the outside air temperature. The end of the heating season is assumed at 16 °C—activation of the limitation. The beginning of the heating season is assumed at 14 °C—limitation deactivation.

The flow of the heating medium through the additional heat source is the same as through the heating system $\dot{m}_h$.

2.3.4. Heat Pump

The selection of the heat pump was based on the highest heating capacity achievable from a given airflow through the evaporator. The air capacity of the heat pump was also adjusted to the airflow exhausted from the building. However, the flow of the heating medium through the heat pump $\dot{m}_{HP}$ during its operation is constant and equals 1750 kg/h, and during its standstill equals 0 kg/h.

The equations describing the characteristics of the heat pump were determined based on the manufacturer's data for a typical chosen device. However, the heat pump's heating power was corrected for heat transfer losses. The heat transport efficiency equal to 0.95 was taken from [54] as for the VRV system (the structure of piping and system elements is similar to the analyzed system). To obtain the below equations, depending on the temperature of the heating medium at the inlet of the heat pump θ_{in}, the polynomial regression method was used.

- Heat transfer from the heat pump to the water in kW:

$$\dot{Q}_{HP} = 0.0003 \cdot \theta_{in}^2 - 0.1149 \cdot \theta_{in} + 16.765 \quad (10)$$

- Electric power of the heat pump in kW:

$$\dot{Q}_{HPel} = -0.0005 \cdot \theta_{in}^2 + 0.0771 \cdot \theta_{in} + 1.1877 \quad (11)$$

where:

θ_{in} temperature of the heating medium at the inlet to the heat pump, °C.

The analyses were performed for different assumed values of the heating medium temperature limit $\dot{Q}_{lim,HP}$ in the bottom part of the buffer. For the variants, these values were: 55 °C, 45 °C, 35 °C, θ_s', and θ_r'. In addition to the variant limit, a fixed limit of 55 °C was applied regardless of the variant. an upper dead band of 1 K was taken into account, which in the variant with a limit of 45 °C means that the limit is tripped at 46 °C and turned off at 44 °C.

Additionally, a limitation of the heat pump operation time beyond the heating season was applied: hysteresis depending on the outside air temperature. The end of the heating season is assumed at 16 °C (limitation activation). The beginning of the heating season is assumed at 14 °C (at which the limitation is switched off). The system does not work in the summer months (June, July, August).

The heat pump may be restarted a minimum of 10 minutes after being turned off.

2.3.5. Buffer Tank

Simulations were performed for a buffer tank of volume 300 dm^3. Its volume was based on an index of 25 dm^3/kW [55] of the heating power of the heat pump (25 dm^3/kW × 12 kW = 300 dm^3). The water space of the buffer was modeled by dividing it into 8 layers, evenly distributed over the height of the tank.

Heat losses from the tank surface were considered. The heat transfer coefficient of the tank wall U = 0.5 W/(m^{2}K).

The connections of the buffer are arranged as follows:

- Supply from the heat pump (tank loading)—one layer below the highest layer of the tank;
- Return to the heat pump—the lowest layer of the tank,
- System supply (in the direction of additional heat source)—the highest layer of the tank;
- Return from the system—one layer above the lowest layer of the tank.

2.4. Performance Indexes

Based on the results of the simulations, a number of indicators were determined in order to characterize the operation of the heat pump and the entire system. The most important ones are described below, and their values are provided in the next section of the article.

Seasonal energy efficiency of a heat pump:

$$SCOP_{PC} = \frac{E_{HP}}{E_{HPel}} \tag{12}$$

Heat pump contribution to building heating energy demand:

$$n_{PC} = \frac{E_{HP}}{E_{heating}} \tag{13}$$

Indicator characterizing the effect of the heating system on the energy performance of the building (ratio of the nonrenewable primary energy factor to the efficiency of the heat source including their share) proposed by the authors (*PPR*—primary energy/performance ratio) in [19] for two heat sources (gas boiler and heat pump) taking the form:

$$PPR = \frac{PEF_g}{\eta_g} \cdot (1 - n_{PC}) + \frac{PEF_{el}}{SCOP_{HP}} \cdot n_{PC} \tag{14}$$

where:

E_{HP} heat transferred by the heat pump to the heating medium during the heating period, kWh;
E_{HPel} electric energy consumption by the heat pump during the heating period, kWh;
PEF_g the non-renewable primary energy factor for natural gas, which equals 1.1 under Polish conditions [54];
PEF_{el} the non-renewable primary energy factor for grid electricity, which equals 3.0 under Polish conditions [54];
η_g the efficiency of heat generation in a condensing gas boiler is assumed as 0.95 [54].

Assuming the heat pump contribution of $n_{PC} = 0$, i.e., for the operation of the boiler alone, *PPR* = 1.16. This reference value was used to evaluate the simulation results for the different variants.

In addition, a number of other values were determined to characterize the operation of the system, including the average temperature of the water in the buffer during the heating season, the operating time of the heat pump, and the operating time of the boiler during the heating season, the values of which are included in the Supplementary Materials to the article.

3. Results and Discussion

On the basis of the obtained results (Figure 5) of the simulation of the heat pump operation, the influence of different heating installation temperature parameters, the temperature limitations of the heat pump operation, and the building location on the operation parameters and the heat pump's share—as well as on the energy characteristics of the analyzed building—are presented.

Figure 5. Results of the simulations.

The highest *SCOP* value in each location was obtained for the lowest temperature of installation 35 °C/28 °C, with a variable limitation of heat pump operation to a theoretical supply temperature θ'_s. These values were 5.10 for Koszalin, Wrocław, and Lublin, and 5.12 for Białystok and Suwałki, whereas the lowest *SCOP* values were found in each location for installations with the highest return temperature of 75/65 °C, with a constant limitation of heat pump operation to 55 °C. The lowest value of 3.17 was observed for Koszalin.

The highest heat pump contribution in covering the building's heat demand n_{PC} in each location was obtained for the installation with the lowest parameters of 35/28 °C, with the heat pump operation constantly limited to 55 °C. The highest value of 0.869 was obtained for Koszalin, whereas, the lowest values of this contribution were in every location for installations with the highest return temperature at 75/65 °C, with a constant limitation of heat pump operation to 35 °C. The lowest value of 0.110 was found in Suwałki.

The highest *SCOP* and heat pump n_{PC} contribution were not observed for the same variants. Therefore, the highest value of *PPR,* which is a function of both of these parameters, occurring for all locations, was obtained for the lowest parameter of installation 35/28 °C, with the variable limitation of heat pump operation to a theoretical return temperature θ'_r. This result was different from the variant with the highest *SCOP* and different from the variant with the highest heat pump contribution n_{PC}. In contrast, the lowest *PPR* values occurred in the same variants as the lowest heat pump n_{PC} contribution.

3.1. Influence of Heating Installation Parameters

The influence of the heating system parameters on the analysis results was different for fixed and variable temperature limitations during heat pump operation.

In the case of limiting the buffer temperature to fixed values of 35 °C, 45 °C, and 55 °C, the results depended on the system's return temperature. As it decreased with respect to the results for the parameters 75/65 °C:

- The *SCOP* of the heat pump increased by 15–30%;
- Heat pump operation time increases by 242–2955 h (see Supplementary Materials);
- The share of the heat pump in heat production was increased by 14–70%; and
- The *PPR* indicator for the building was lowered by 20–34%,

depending on the variant of the heat pump operation limit and building location.

Buffer water temperature, averaged over time, with the falling supply temperature of the central heating system, decreased by 2.9–9.7 °C depending on the variant of heat pump limitation and building location, in relation to results for the highest parameters. In the case of the limits to constant values of 35 °C and 45 °C for all locations and for the limitation to a constant 55 °C in the case of Koszalin, Wrocław, and Lublin it decreased with the lowering of the designated supply temperature. In other cases, the value decreased with the decreasing of the designated return temperature of the system.

In the case of variable limitation applied to both the theoretical instantaneous supply temperature θ'_s and the theoretical instantaneous return temperature θ'_r, the heat pump *SCOP* increased by 48–54% with decreasing design supply temperature, depending on the building location.

In the case of variable limitation to the theoretical instantaneous supply temperature θ'_s, with the decreasing of the design return temperature the *PPR* increased by 26–30% and the heat pump's share increased by 12%, depending on the building's location.

In the case of the variable limitation applied to the theoretical instantaneous return temperature θ'_r, for Koszalin, Wrocław, Lublin, and Białystok, with the decreasing design supply temperature the *PPR* increased by 26–31%, and the heat pump share increased by 18–36%. On the other hand, as the designated return temperature decreased, the *PPR* increased by 23%, and the share of heat pumps increased by 7%.

3.2. Effect of Temperature Limitation on Heat Pump Operation

Limiting the heat pump operation based on the temperature at the bottom of the buffer tank significantly affected its efficiency and contribution. When limiting it to a fixed temperature value, the limit value increased from 35 °C to 55 °C:

- The time-averaged temperature in the buffer increased by 9.9–17.0 °C (see Supplementary Materials);
- The time of heat pump operation increased by 199–2856 h and the heat pump share increased by 2–58% (see Supplementary Materials);
- The heat pump SCOP decreased by 10–21%,

depending on location and installation parameters.

Due to these changes, the effect of the limit value on the *PPR* was not always the same. For heating installation parameters 80/60 °C and 75/65 °C, the higher the constant limiting temperature, the higher the *PPR* value by 8–10% depending on the location and installation parameters. In the case of the parameters at 35/28 °C, this influence was the opposite and equaled 6–10%. For parameters 70/55 °C and 55/45 °C, there was a minimum. When limited to 45 °C, the *PPR* value was lower than when limited to 35 °C and 55 °C.

Regarding the change from a fixed to a variable constraint, the change to the instantaneous theoretical supply temperature or to the instantaneous theoretical return temperature (depending on the current value of the outdoor temperature) clearly influenced the changes in the average temperature and the buffer efficiency, operation time, and heat pump share, and as a consequence *PPR* was affected. Compared with the results for constant temperature limits, the analyzed variable limits caused a beneficial reduction of *PPR* of 1–4%. Lower values were obtained when limiting the instantaneous heating system supply temperature at system-designated parameters of 35/28 °C and 80/60 °C, whereas at other parameters of 55/45 °C, 70/55 °C, and 75/65 °C the lowest values were obtained when limiting the instantaneous theoretical heating system return temperature.

3.3. Climate Impact

The relationships described above are valid for all analyzed locations and their climate data. However, for individual locations, the lower the average outdoor air temperature during the heating season, the:

- Longer the operation time of the heat pump up to 1837 h, for parameters at 80/60 °C, whereas for other cases this time will be shortened by 133–792 h, depending on the heating installation temperature parameters and the type and value of limitation;
- Higher the heat pump operation share up to 25% for parameters at 80/60 °C, whereas for other cases this share will be 4–10% smaller, depending on the heating installation temperature parameters and the type and value of limitation.

Furthermore, the:

- Lower the average buffer water temperature by 0.1–4.7 °C,
- Lower the *SCOP* up to 7%,
- Higher the *PPR* up to 7%,

depending on heating installation temperature parameters and the type and value of the limitation.

For particular locations, despite the change in the heating installation parameters, the most favorable *PPR* was obtained at similar values of heat pump operation time. For Koszalin and Lublin it was about 2800–2900 h, for Wrocław about 2600–2700 h, for Białystok 3000–3200 h, and for Suwałki about 3300–3400 h.

4. Conclusions

The study provided the quantitative and qualitative evaluation of the analyzed system.

The most favorable variant, *PPR* = 0.668 (*SCOP* = 4.98, heat pump contribution 0.845) was achieved for the building in Koszalin, with heating installation parameters of 35/28 °C and the limiting of the heat pump operation temperature to the theoretical supply temperature of the heating installation. The least favorable variant was achieved for the heating installation with parameters of 75/65 °C, limiting the operation temperature of the heat pump to 35 °C in buildings located in Wrocław (*PPR* = 1.11, *SCOP* = 4.05, heat pump contribution 0.116), Lublin (*PPR* = 1.11, *SCOP* = 4.05, heat pump contribution 0.116), Białystok (*PPR* = 1.11, *SCOP* = 4.05, heat pump contribution 0.116), and Suwałki (*PPR* = 1.11, *SCOP* = 4.05, heat pump contribution 0.110).

The analyzed system has great potential from an energy and environmental point of view. It can provide significant benefits in terms of the energy performance of the building when considering the following points.

- The operation of two heat sources for a single purpose requires the definition of the rules of their cooperation.
- The choice of the design parameters of a central heating system combined with a heat pump and boiler has a key impact on the energy efficiency of the heating system.
- Lowering these parameters increases the efficiency of the heat pump and its contribution to the energy demand for central heating.
- The most energy efficient system, with the lowest *PPR*, is achieved with the lowest heating system temperatures.
- Implementing a temperature limitation for heat pump operation improves its efficiency, but the higher the design parameters of the heating installation and the lower the limitation, the lower the heat pump's contribution, and the higher the *SCOP* and *PPR*.
- The energy effect is also influenced by the location, but its scale depends on the parameters of the heating system and the temperature limitation of the heat pump operation. This is more significant for lower heating system parameters.

Therefore, the use of the considered solution is particularly beneficial in:

- New buildings with low energy demands for heating, enabling the use of low-temperature heating systems;
- Existing buildings that have been thermally renovated with reduced energy demands for heating, in which old radiators have been preserved and the heating system power has been reduced by lowering central heating parameters.

This system provides the possibility of further reducing the demand for nonrenewable primary energy by powering the heat pump with photovoltaic cells. On the other hand, operating the heat pump in more favorable conditions (using indoor air instead of outdoor air) and with higher efficiency, allows the limiting of the size of the PV installation and the nominal capacity of the heat pump itself.

Supplementary Materials: The following are available online at https://www.mdpi.com/article/10.3390/en14237985/s1, Table S1: Simulation results.

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