Exploring the Efficiency of a Residential Air-to-Water Heat Pump in Summer Changeover Regimes Based on Exergy Analysis: A Case Study

***Goran Vučković*[a], *Mirko Stojiljković*[b], *Marko Ignjatović*[c] *and Branka Radovanović*[d]**

[a] *University of Niš, Faculty of Mechanical Engineering in Niš, Serbia, goran.vuckovic@masfak.ni.ac.rs, CA*

[b] *University of Niš, Faculty of Mechanical Engineering in Niš, Serbia, mirko.stojiljkovic@masfak.ni.ac.rs*

[c] *University of Niš, Faculty of Mechanical Engineering in Niš, Serbia, marko.ignjatovic@masfak.ni.ac.rs*

[d] *University of Niš, Faculty of Mechanical Engineering in Niš, Serbia, branka.radovanovic@masfak.ni.ac.rs*

Abstract. Heat pumps meet the needs of the residential sector through the following three purposes: domestic water heating, space heating and cooling. This paper presents experimental research into the energy performance of an air-to-water heat pump in real summer changeover regimes, given that these are periods that significantly affect the reduction of efficiency. The transition regimes between the two regimes of operation with the most pronounced changes were especially considered. These were the transition regimes characteristic of operation in the summer period, namely: the transition from the space cooling regime to the domestic water reheating regime, and vice versa. In the observed period, the focus has been on 60 readings, with a resolution of 1 minute. In that period two changes in the operating mode were made, from the stationary space cooling regime (when the heat pump prepared water for the underfloor cooling at 17ºC) to the domestic water reheating mode (when the heat pump prepared hot water in the tank at 43ºC during the day), as well as switching to the previous, cooling mode. The research results show that in the observed period the heat pump worked in stationary regimes of space cooling and domestic water reheating, respectively, with average performance values of 3.52 and 4.75, and average exergetic efficiency of 6.93% and 33.38%.

Keywords: Heat pump, Exergy analysis, Underfloor cooling, Domestic water reheating.

1. Introduction and background

Over the past couple of decades, the research community has witnessed a paradigm shift in the development of technical systems. In addition to efficiency, which has always been an important requirement in technical systems, the concepts of sustainability and green energy [1, 2] have started to take precedence in recent years. Even if the focus is limited to the field of civil engineering, this trend is easily recognizable in various areas, such as heating and cooling of buildings [3], wastewater treatment [4], spatial planning for energy efficient buildings [5].

Energy demand in the building sector in the past ten years has seen an average annual increasing over more than 1%. In 2021 and 2022 the space cooling saw the largest increase in energy demand, having in mind all needs in buildings sector. From the other hand, in the same period the space heating energy consumption decreased by 4%. In the past years, the heat pump technology is recognised as one of main decarbonisation technology [6]. Also, heat pumps are crucial to enable balancing of the power grid as a flexibly tool for balancing energy supply and energy needs, by acting as a virtual electricity battery via thermal energy storage.

In the buildings of modern history, the energy for domestic water heating accounts for approximately 40–50% of total needs [7]. The main conclusion in paper [7] is that for performance optimizing of heat pump the significant role is refrigerant charge. The heat pump coefficient of performance increased by 16.27% with optimal refrigerant charge. The maximum value of exergetic efficiency was 34.17%, and in the evaporator was observed the largest destruction of exergy (44.04–49.22% of the total exergy destruction).

The author in paper [8] identifies three main reasons that affect the efficiency of vapour-compression heat pumps. These are: irreversibility in the heat transfer between the working fluid and the heated space, and the presence of two intermediaries in the heat transfer. The first intermediary is the refrigerant through which energy is extracted from the low-temperature source and the second intermediary is water (most often) through which energy is transferred to the space heating. The article [9] presents experimental investigation of a new air source heat pump with integrated functions for total residential comfort. The heat pump reached a Seasonal EER of 17.0 (average COP of 4.98) and a Seasonal heating performance factor of 11.0 (average SHP of 3.22) in laboratory conditions. In the integrate regime of domestic water heating and space cooling the heat pump attained maximal performance factor of 7.0.

In paper [10] presented the investigation based on the second law of thermodynamics about the impacts of the condensing and evaporating temperatures on the pressure losses, the exergy destruction, and energy and exergy-based efficiency of a refrigeration cycle. Results show that the condensing and evaporating temperatures have strong impacts on the exergy destruction in the condenser and evaporator, and on the exergetic efficiency and energy-based performance of the cycle but small impacts on the exergy destruction in the expansion valve and compressor. The exergetic efficiency and energy-based performance increases, but the total exergy destruction decreases with decreasing temperature difference between the evaporator and cooling space and between the condenser and environment. The analysis of heat pump system based on the second law of thermodynamics have been presented in paper [11]. Authors considered irreversibilities due to friction and heat transfer. They have been analysed the COP as a function of different parameters, and efficiency of the second law. Results shown that Coefficient of performance varies from 7.40 to 3.85 and the exergy efficiency varies from 37% to 25%. The entropy-cycle method used in paper [12] to considers for thermodynamic analysis of heat pump and cooling devices. The authors conclude that all irreversibilities can be described quantitatively and qualitatively in a real cycle to carry out engineering design and optimization.

The paper [13] reviews second law-based research for cooling systems with vapour-compression. Authors conclude that value of exergy depends on condensing and evaporating temperatures, sub-cooling temperature, and compressor pressure. The suction and discharge temperature should between 14°C and 65°C, respectively, for better performance of the vapour-compression cooling system. The article [14] analysed air-source heat pump for simultaneous produce heating and cooling energies for hotels, residential buildings, and office buildings. From the second law of thermodynamic point of view the unit with propane (R290) has a higher performance than the unit with refrigerant R407C.

In study [15] authors conclude that reduction 25% of heat losses can be achieved in decentralized DHW systems (with temperature of 48°C) comparing with centralized DHW systems with ultrafiltration for legionella treatment.

In this paper, the vapour compression low temperature residential heat pump was analysed in real operating conditions in the changeover regimes in the summer period. The focus is on transition from domestic water reheating regime to space cooling regime and vice versa. The scientific contribution of this manuscript is reflected in the application of the second law of thermodynamics method (exergy analysis) to determine the quality of energy and the possibility of improving the efficiency of residential air-to-water heat pumps.

2. Short description of residential air-to-water heat pump system

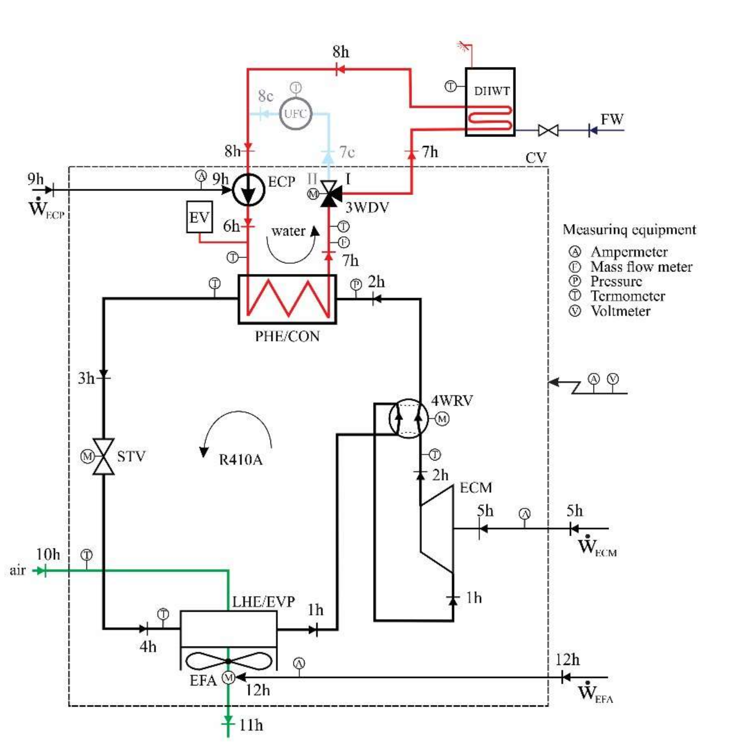
A schematic of a vapour compression residential air-to-water heat pump for domestic water reheating and space cooling is shown in figure 1a and figure 2a, respectively. The energy system consists of refrigerant cycle and water cycle. The refrigerant is R410A. The heat pump in the split version consists of outdoor and indoor units. The units interconnected with two insulated pipes of refrigerant (liquid and vapour phases). Outdoor unit have four main components: inverter type compressor (ECM), lamella Cu-Al refrigerant-air heat exchanger (LHE) with an electronic speed control axial fan on air side (EFA), four-way refrigerant reversing valve (4WRV) and electronic throttling refrigerant valve (ETV). Indoor unit have compact plate refrigerant-water heat exchanger (PHE), three-way water diverting valve (3WDV), expansion vessel (EV), electronic speed control circulating pump (ECP) and domestic hot water tank (capacity 180 litters).

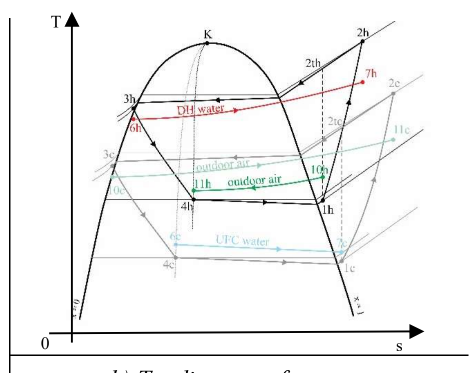
In the summer period two different regimes are analysed: space cooling regime with underfloor cooling system (UFC) at temperature t_{7c} = $t_{UFC,sp}$ = 17°C and domestic water reheating regime at two temperature levels ($t_{DHW,sp}$ = 48°C during the night, $t_{DHW,sp}$ = 43°C during the day).

For the changeover of regimes used four-way refrigerant reversing valve (4WRV) and three-way water diversion valve (3WDV), figures 1a and 2a. When the 4WRV in heating position and 3WDV in position I, DHW preparation is enabled (refrigerant cycle 1h-2h-3h-4h and water cycle 6h-7h-8h), figure 1, while when 4WRV in cooling position and 3WDV in position II, space cooling is running (refrigerant cycle 1c-2c-3c-4c and water cycle 6c-7c-8c), figure 2. The priority is preparation of domestic hot water.

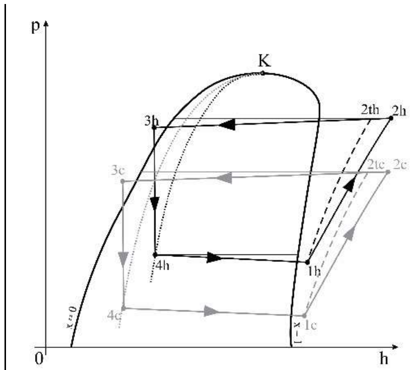
The flow of water provides circulating pump has an electronic speed control function.

Circulating pump, axial fan and compressor using 230V AC power supply from electricity company, which represented with energy flows 5h(c), 9h(c) and 12h(c), measured with amperemeters, figures 1a, 2a.



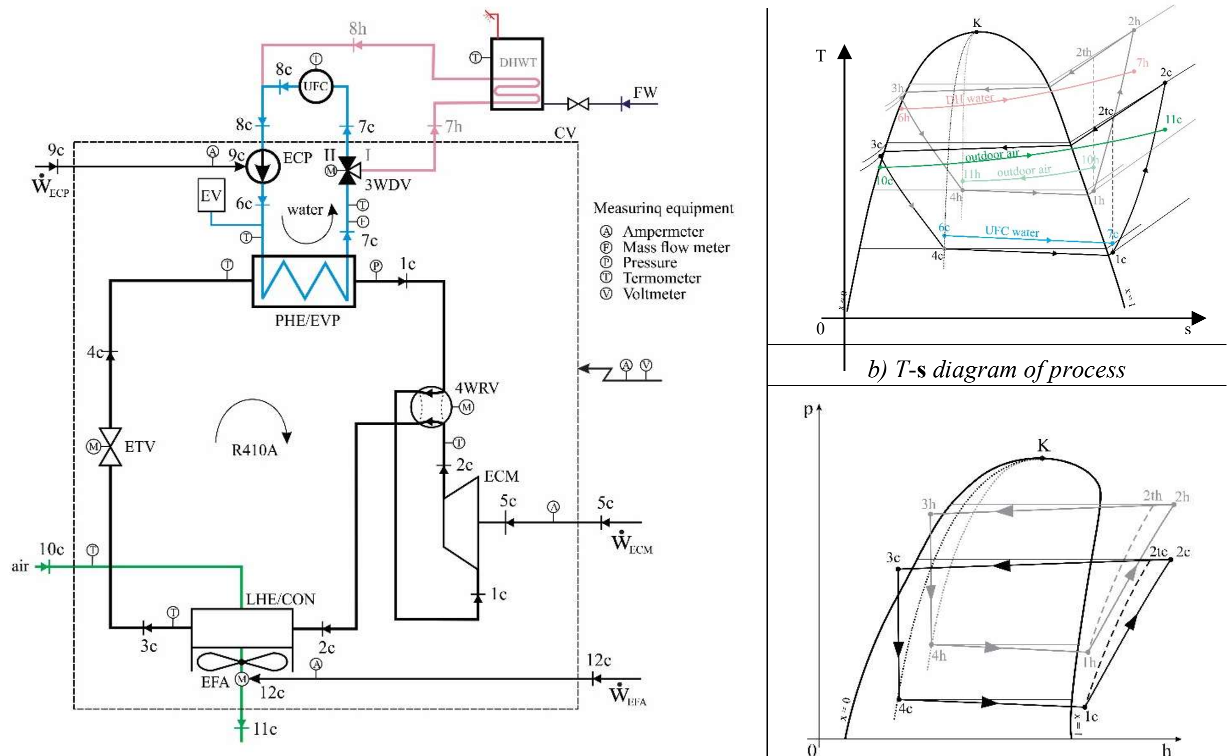


b) T-s diagram of process



a) Scheme in heat pump regime | c) log p-h diagram of process

Figure 1. Scheme and diagrams for heat pump domestic water reheating regime



b) *T*-**s** *diagram of process*

a) Scheme in space cooling regime | c) log p-h diagram of process

Figure 2. Scheme and diagrams for space cooling regime

Parameters of process fluids (temperature, pressure, mass flow rate and electricity), are directly measured at a total of 13 positions with original sensors installed at the heat pump factory, figures 1a and 2a. Other characteristic values were calculated.

3. Modelling and methodology

The concept of exergy is based on the second law of thermodynamics and provides information about the quality of energy. Every energy conversion process destroys exergy. The exergy analysis can help to minimize energy losses and reduce the footprint of exergy destruction [16].

In this paper for computation exergy parameters, standard values of temperature and pressure were applied, i.e. $T_0 = 298.15$ K, $p_0 = 1.01325$ bar. Accordingly, the more interesting component of total exergy is a physical exergy which we determine according to the following expression:

$$e_j^{\mathrm{PH}} = e_j = (h_j - h_0) - T_0(s_j - s_0). \tag{1}$$

In the steady state conditions at a component level, equations for entropy (2) and exergy balance (3) have the following forms [17]:

$$0 = \sum_j \left(\frac{\dot{Q}_j}{T_j}\right)_k + \sum_i (\dot{m}_i s_i)_k - \sum_e (\dot{m}_e s_e)_k + \dot{S}_{\mathrm{gen},k} \tag{2}$$

$$0 = \sum_j \left(\left(1 - \frac{T_0}{T_j}\right)\dot{Q}_j\right)_k - \dot{W}_{\mathrm{cv},k} + \sum_i (\dot{m}_i e_i)_k - \sum_e (\dot{m}_e e_e)_k - \dot{E}_{\mathrm{D},k}. \tag{3}$$

The entropy generation rate and exergy destruction rate of the component represents on the last terms on the right sides of equations (2) and (3). The exergy destruction rate can be calculated as a product of absolute surrounding temperature and entropy generation rate:

$$\dot{E}_{\mathrm{D},k} = T_0 \dot{S}_{\mathrm{gen},k}. \tag{4}$$

The total exergy destruction rate for the whole energy system (heat pump) is determined as a summing of exergy destruction of each component of the energy system:

$$\dot{E}_{\mathrm{D,tot}} = \sum_{k=1}^{n} \dot{E}_{\mathrm{D},k}. \tag{5}$$

In the case where the refrigerant is considered as an ideal gas in the superheating region with a constant specific heat, the specific entropy generation due to the pressure losses through one heat exchanger of the heat pump system (a condenser in the heating and cooling mode) can also be expressed [9]:

$$\left(s_{\mathrm{gen},23}\right)_{\Delta p} = (s_3 - s_2) + 2\left(\frac{h_{2"} - h_3}{T_{2"} + T_3}\right) + c_{\mathrm{p,r}} \ln\left(\frac{T_2}{T_{2"}}\right). \tag{6}$$

The specific entropy generation due to the pressure losses through another heat exchanger of the heat pump system (an evaporator in the heating and cooling mode) can be expressed in the same manner as [9]:

$$\left(s_{\mathrm{gen},41}\right)_{\Delta p} = (s_1 - s_4) - 2\left(\frac{h_1 - h_4}{T_1 + T_4}\right). \tag{7}$$

The entropy generation rate due to heat transfer is equal to the difference between the total rate of entropy generation and the value of entropy generation due to pressure losses:

$$\left(\dot{S}_{\mathrm{gen}}\right)_q = \dot{S}_{\mathrm{gen}} - \left(\dot{S}_{\mathrm{gen}}\right)_{\Delta p}. \tag{8}$$

The total rate of entropy generation for each component of the heat pump system is determined based on entropy balance equation (2). The maximal theoretical energy-based efficiency is reached when the energy system operates between two reservoirs (cold or low temperature reservoir and hot temperature reservoir) during the reversible Carnot cycle [18]:

$$\mathrm{COP_C} = \frac{T_{\mathrm{H,h}}}{T_{\mathrm{H,h}} - T_{\mathrm{C,h}}}; \qquad \mathrm{EER_C} = \frac{T_{\mathrm{C,c}}}{T_{\mathrm{H,c}} - T_{\mathrm{C,c}}}. \tag{9}$$

The maximal real energy-based efficiency is reached when the energy system operates between heat source and heat sink temperatures (evaporation/condensation temperatures) during the reversible cycle [18]:

$$\mathrm{COP_{max}} = \frac{T_{\mathrm{con,h}}}{T_{\mathrm{con,h}} - T_{\mathrm{evp,h}}}; \qquad \mathrm{EER_{max}} = \frac{T_{\mathrm{evp,c}}}{T_{\mathrm{con,c}} - T_{\mathrm{evp,c}}}. \tag{10}$$

The COP of a heat pump is the ratio of heat delivered to the hot reservoir to the work put into the system. For the whole system when the heat pump is working in DHW regime can be calculated [8]:

$$\mathrm{COP_{tot}} = \frac{\dot{Q}_{\mathrm{DHW}}}{\dot{W}_{\mathrm{tot,h}}}; \qquad (\mathrm{COP_{tot}} < \mathrm{COP_{max}} < \mathrm{COP_C}). \tag{11}$$

Also, the Energy efficiency ratio is the ratio of a unit`s cooling output relative to its input power. For the whole system when the heat pump working in space cooling regime can be calculate:

$$\mathrm{EER_{tot}} = \frac{\dot{Q}_{\mathrm{UFC}}}{\dot{W}_{\mathrm{tot,c}}}; \qquad (\mathrm{EER_{tot}} < \mathrm{EER_{max}} < \mathrm{EER_C}). \tag{12}$$

The dividend in equations (11) and (12) represents the delivered heat to end-user. In the domestic water reheating regime expressed as ($\dot{Q}_{\mathrm{DHW}} = \dot{m}_{\mathrm{w}}(h_{7\mathrm{h}} - h_{8\mathrm{h}})$), and in space cooling regime as ($\dot{Q}_{\mathrm{UFC}} = \dot{m}_{\mathrm{w}}(h_{8\mathrm{c}} - h_{7\mathrm{c}})$). The divisor in equations (11) and (12) represent the total input electrical power for all electronic consumers in heat pump: vapor-compressor, outdoor air axial fan and water circulation pump, all devices with electronic speed control options. In case of heat pump operating in domestic water heating regime ($\dot{W}_{\mathrm{tot,h}} = \dot{W}_{5\mathrm{h}} + \dot{W}_{12\mathrm{h}} + \dot{W}_{9\mathrm{h}}$) or in case operating in space cooling regime ($\dot{W}_{\mathrm{tot,c}} = \dot{W}_{5\mathrm{c}} + \dot{W}_{12\mathrm{c}} + \dot{W}_{9\mathrm{c}}$).

On the component or overall system level the exergetic efficiency is defined as the ratio between exergy of products and exergy of fuel [19]:

$$\varepsilon_k = \frac{\dot{E}_{\mathrm{P},k}}{\dot{E}_{\mathrm{F},k}} = 1 - \frac{\dot{E}_{\mathrm{D},k} + \dot{E}_{\mathrm{L},k}}{\dot{E}_{\mathrm{F},k}}; \qquad \varepsilon_{tot} = \frac{\dot{E}_{\mathrm{P,tot}}}{\dot{E}_{\mathrm{F,tot}}} = 1 - \frac{\dot{E}_{\mathrm{D,tot}} + \dot{E}_{\mathrm{L,tot}}}{\dot{E}_{\mathrm{F,tot}}}. \tag{13}$$

In the steady state conditions, exergy of fuel is a sum of exergy of product, exergy destruction and exergy loss on the component or overall system level:

$$\dot{E}_{\mathrm{F},k} = \dot{E}_{\mathrm{P},k} + \dot{E}_{\mathrm{D},k} + \dot{E}_{\mathrm{L},k}; \qquad \dot{E}_{\mathrm{F,tot}} = \dot{E}_{\mathrm{P,tot}} + \dot{E}_{\mathrm{D,tot}} + \dot{E}_{\mathrm{L,tot}}. \tag{14}$$

The exergy loss of components has a zero value ($\dot{E}_{\mathrm{L},k} = 0$) when the boundary of component set at surrounding temperature, but remains the exergy loss of overall energy system ($\dot{E}_{\mathrm{L,tot}} \neq 0$).

The exergy destruction ratio and exergy loss ratio can be calculated when the rate of component exergy destruction and total exergy loos, respectively, divided with exergy of fuel for overall system [17]:

$$y_{\mathrm{D},k} = \frac{\dot{E}_{\mathrm{D},k}}{\dot{E}_{\mathrm{F,tot}}}; \qquad y_{\mathrm{L,tot}} = \frac{\dot{E}_{\mathrm{L,tot}}}{\dot{E}_{\mathrm{F,tot}}}. \tag{15}$$

These coefficients (15) can determine how every component of the system and total exergy loss have an influence on the system's exergetic efficiency ($\varepsilon_{\mathrm{tot}} = 1 - \Sigma y_{\mathrm{D},k} - y_{\mathrm{L,tot}}$).

The coefficient of total exergy destruction of components is given when the component exergy destruction rate compared with the total exergy destruction rate of overall system [19]:

$$y^*_{\mathrm{D},k} = \frac{\dot{E}_{\mathrm{D},k}}{\dot{E}_{\mathrm{D,tot}}}. \tag{16}$$

4. Results and discussions

The results presented in this manuscript are related to a low-temperature residential air-to-water heat pump in the split version, with the nominal capacity of the outdoor unit of 6.0 kW in the heating mode and 6.76 kW in the cooling regime, and vertical type of indoor unit with integrated hydro-module and 180 litters

DHW tank, both units with single-phase electricity connection (nominal power input 1.27 kW in heating mode (COP_{nom} = 4.74), and 1.96 kW in cooling regime (EER_{nom} = 3.45)) [20].

The monitoring of the system operation was carried out through a system device (D-checker) for servicing the equipment and reading parameters from all heat pump sensors for a total of 11 days, starting from 2024-08-13 21:54 to 2024-08-24 10:16 (a total of 15,109 minutes or N = 15,109 readings/measurements). For further analysis, three characteristic days (2024-08-14 00:01 to 2024-08-17 00:00) were selected, figure 3. Bearing in mind that a periodic change was registered, the changes from space cooling to domestic water heating and vice versa were analysed in detail during one representative hour (2024-08-14 in the period from 00:57 p.m. to 1:56 p.m., figure 4. The reading of process variables was performed with a step of one minute.

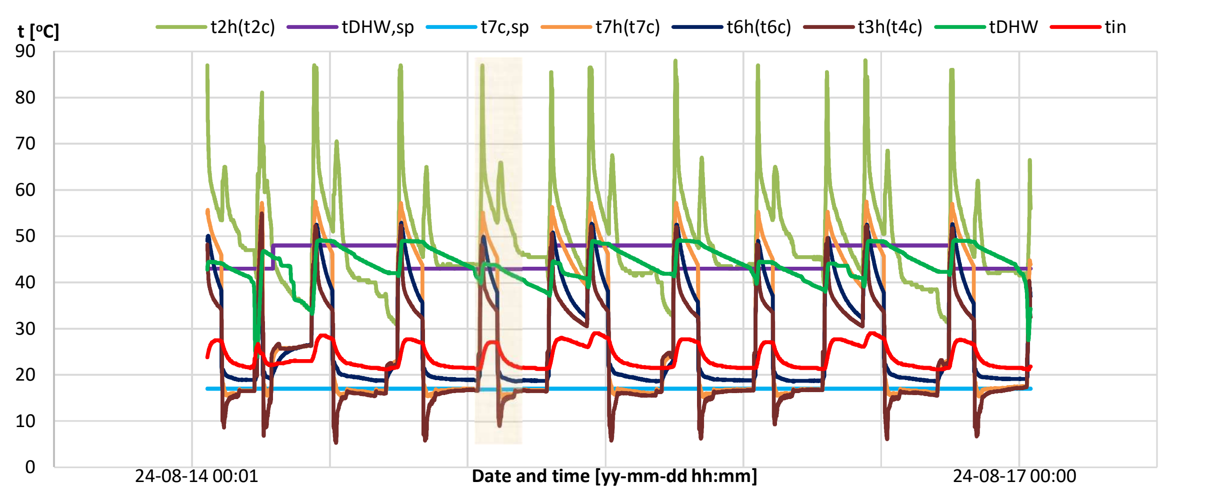


Figure 3. Profiles of temperature for process fluids for three days working

The transition from the space cooling regime to the domestic water reheating regime takes three minutes, and the transition from the domestic water reheating regime to the space cooling regime takes four minutes, figure 4. During the transition period, the compressor is not in operation, as well as the axial fan on the heat exchanger of the outdoor unit. In that period, the mode change is essentially done via 4WRV and 3WDV. The 4WRV changes the direction of the refrigerant through the two heat exchangers and through the ETV, and the 3WDV redirects the water from the floor panels to the DHW tank and vice versa.

During DHW preparation, figures 1, 3 and 4 the goal was to heat the water in the tank ($t_{DHW,sp}$ = 43°C during the day and $t_{DHW,sp}$ = 48°C during the night, figure 3). The temperature in the was 38.6°C and in the first few minutes of operation it dropped by 0.4°C due to sensor position. Bearing in mind that the priority goal was to heat the DW to the set temperature as soon as possible, the compressor started with full capacity and discharge temperature increasing from 45.0°C to 84.0°C, figure 3. When the temperature of water in the tank approaches one degree below the set temperature, the compressor reduced the frequency and entered modulation. Due to the inertia of the system, the set point temperature was reached in the next 60 seconds, and the compressor was switched off. The domestic water reheating regime ran for about 13 minutes.

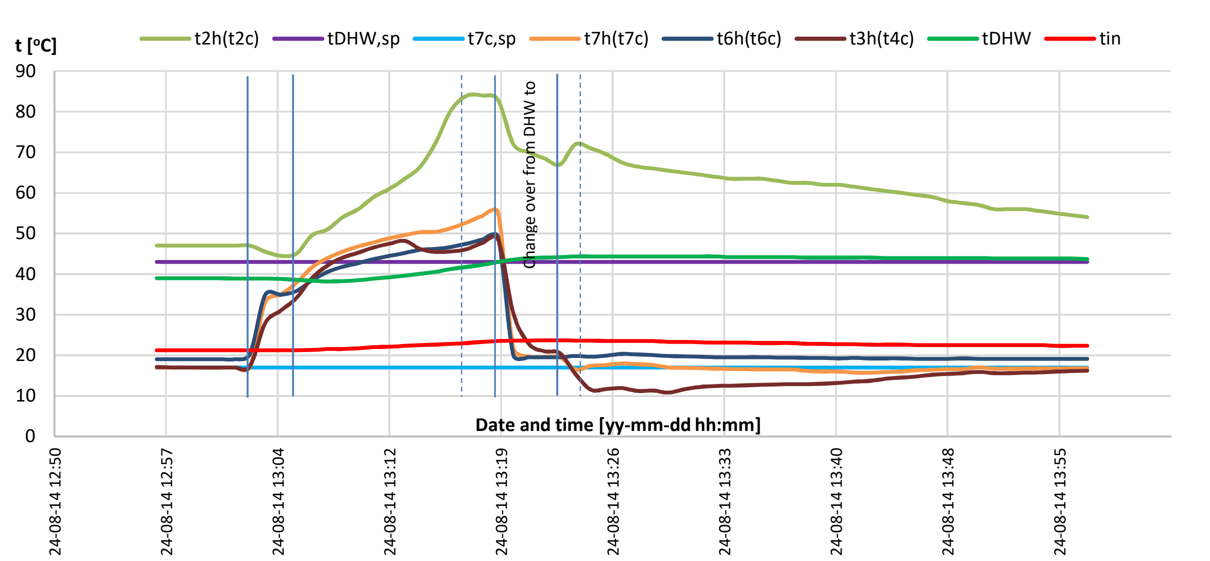


Figure 4. Profiles of temperature for process fluids during representative hour

Table 1. Characteristic equations for heating and cooling regimes, and real operation conditions

Component	Exergy of fuel		Exergy of product		Real operation conditions
	Heating	Cooling	Heating	Cooling	
Compressor	$\dot{W}_{5h}$	$\dot{W}_{5c}$	$\dot{E}_{2h}$ - $\dot{E}_{1h}$	$\dot{E}_{2c}$ - $\dot{E}_{1c}$	$\eta < 0.85$; $Q_L \neq 0$
Condenser	$\dot{E}_{2h}$ - $\dot{E}_{3h}$	$\dot{W}_{12c} + (\dot{E}_{11c}$ - $\dot{E}_{10c})$	$\dot{E}_{7h}$ - $\dot{E}_{6h}$	$\dot{E}_{2c}$ - $\dot{E}_{3c}$	$\Delta p' = 0$; $\Delta p'' \neq 0$; $Q_L \neq 0$
Throttling valve	$\dot{E}_{3h}$	$\dot{E}_{3c}$	$\dot{E}_{4h}$	$\dot{E}_{4c}$	$h = const.$; $Q_L \neq 0$
Evaporator	$\dot{W}_{12h} + (\dot{E}_{10h}$ - $\dot{E}_{11h})$	$\dot{E}_{4c}$ - $\dot{E}_{1c}$	$\dot{E}_{1h}$ - $\dot{E}_{4h}$	$\dot{E}_{7c}$ - $\dot{E}_{6c}$	$\Delta p' = \Delta p'' = 0$; $Q_L \neq 0$
Circulation pump	$\dot{W}_{9h}$	$\dot{W}_{9c}$	$\dot{E}_{6h}$ - $\dot{E}_{8h}$	$\dot{E}_{6c} - \dot{E}_{8c}$	$\eta = 0.80$
Overall system	$\dot{W}_{tot,h} + (\dot{E}_{10h}$ - $\dot{E}_{11h})$	$\dot{W}_{tot,c} + (\dot{E}_{11c}$ - $\dot{E}_{10c})$	$\dot{E}_{7h}$ - $\dot{E}_{6h}$	$\dot{E}_{7c}$ - $\dot{E}_{6c}$	
(Heat pump)	$\dot{W}_{tot,h(c)} = \dot{W}_{5h(c)} + \dot{W}_{9h(c)} + \dot{W}_{12h(c)}$				

During UFC operation, figures 2., 3. and 4. the goal was to prepare the supply water temperature on 17°C ($t_7 = t_{UFC,sp} = 17$°C). During start-up, this temperature was 17.6°C. The compressor started with a reduced capacity (cca 45% of full capacity), and it was permanently reduced to a technological minimum, which was 20% of full capacity. The heat pump system permanently performed quantitative and qualitative regulation of work (adjustment of temperature and flow rate of refrigerant, as well as adjustment of temperature and flow rate of water). After 24 minutes of entering the space cooling mode, the heat pump had fully adjusted its operation to the needs of the consumer, with the temperature of the refrigerant vapour continuing to move in the range of 47-57°C, the flow rate of the refrigerant was 0.011-0.015 kg/s, the temperature difference of the inlet and output water was in the range 3-5°C, and the water mass flow rate was 0.18-0.21 kg/s. The transition from the space cooling regime to the domestic water reheating regime was done automatically when the water temperature in the storage tank dropped at least five degrees below the set temperature. ($t_{DHW} < t_{DHW,sp}$ -5°C).

For reading all the thermodynamics parameters of process fluids, the CoolProp software is used [21].

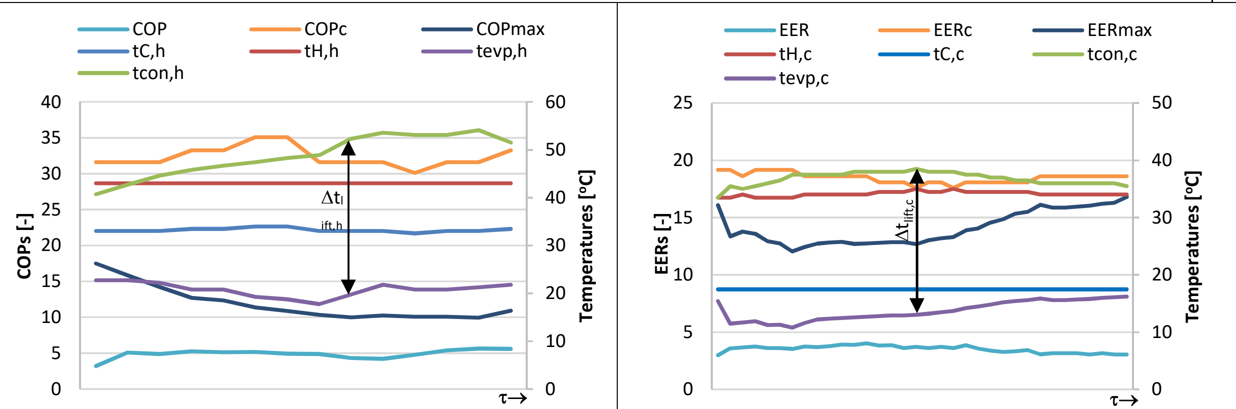
For each component and for the overall system in table 1. shown equations for calculating the exergy of fuels, the exergy of products, and parameters for real operating conditions.

For calculating thermal, energy and exergy variables for both regimes were used the real values for all process fluids. The basic statistics have been applied for every parameter: maximal, minimal, average, median and standard deviation [22]. The average values were used for all further analysis.

The overview of statistical values of main thermodynamic, energy and exergy parameters of the overall system for both regimes of operation, provides on table 2. Energy-based performance (COP) during the DHW operation varies around average value of 4.75 (very close to the value given in the manufacturer`s catalogue COP_{nom} = 4.74 [15] and within the limits of the results in paper [11]) and the median value 4.84, table 2 and figure 5a.

Table 2. Overview of statistical values for both regimes

Para-meters	Unit	Domestic water reheating regime*					Underfloor space cooling regime**				
		Average	Median	St.dev.	Max.	Min.	Average	Median	St.dev.	Max.	Min.
$t_{2h(c)}$	[°C]	64.96	62.25	13.19	84.00	45.00	61.62	62.00	4.67	72.00	54.00
$t_{7h(c)}$	[°C]	48.29	49.20	4.68	55.20	37.80	16.66	16.65	0.55	18.00	15.70
$t_{con,h(c)}$	[°C]	48.77	48.60	4.17	54.10	40.70	36.81	37.00	1.12	38.50	33.50
$t_{evp,h(c)}$	[°C]	20.79	20.75	1.43	22.75	17.75	13.72	13.35	1.71	16.20	10.80
$\Delta t_{lift,h(c)}$	[°C]	27.99	29.65	4.18	32.85	17.95	23.09	23.80	2.34	26.70	18.10
$\dot{m}_{r,h(c)}$	[kg/s]	0.036	0.037	0.007	0.047	0.017	0.015	0.015	0.003	0.020	0.011
$\dot{m}_{w,h(c)}$	[kg/s]	0.350	0.351	0.005	0.354	0.338	0.214	0.211	0.032	0.251	0.180
COP/EER	[-]	4.75	4.84	0.60	5.53	3.08	3.52	3.61	0.30	4.05	2.98
$\dot{E}_{F,tot,h(c)}$	[kW]	1.25	1.27	0.22	1.63	0.89	0.83	0.82	0.12	1.25	0.65
$\dot{E}_{P,tot,h(c)}$	[kW]	0.43	0.44	0.16	0.71	0.11	0.06	0.06	0.01	0.08	0.04
$\dot{E}_{D,tot,h(c)}$	[kW]	0.74	0.70	0.10	0.93	0.63	0.47	0.47	0.06	0.81	0.42
$\dot{E}_{L,tot,h(c)}$	[kW]	0.08	0.07	0.05	0.16	0.01	0.29	0.31	0.07	0.38	0.18
$y_{L,tot,h(c)}$	[%]	5.97	5.26	3.58	13.00	1.17	35.14	37.32	5.01	41.33	26.03
$\varepsilon_{tot,h(c)}$	[%]	33.38	33.53	8.40	48.03	21.91	6.93	6.72	0.75	8.19	5.74
* - subscript h & COP; ** - subscript c & EER											



a) Domestic water reheating *b) Underfloor space cooling*

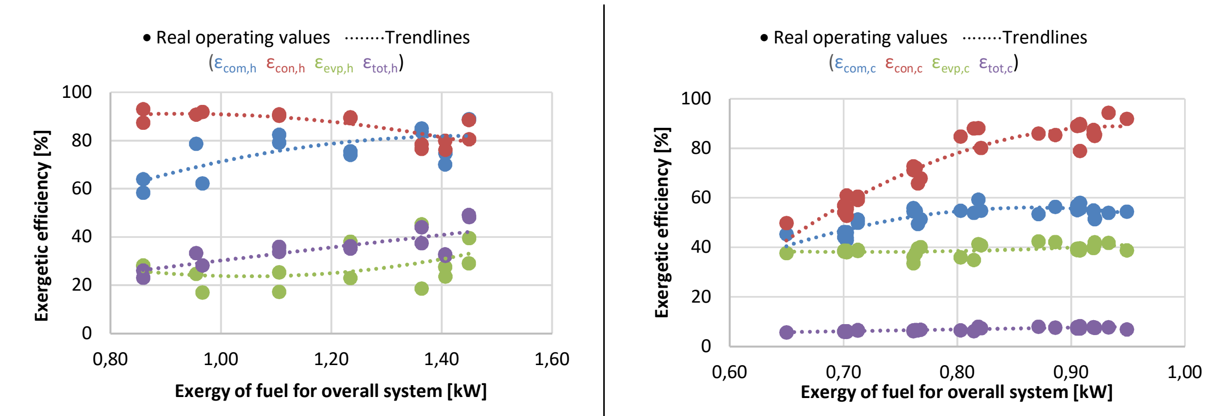
Figure 5. The main temperatures profiles and energy-based efficiency measure of the overall system

Considering that the standard deviation is 0.60 or 12.63%, experimental results are concentrated around average value. Also, during underfloor space cooling regime energy-based performance (EER) varies around significantly lower average value 3.52 (also very close to the value given in the manufacturer`s catalogue EER_{nom} = 3.45 [15]) and the median value of 3.61, with the standard deviation of 0.30 or 8.52%, table 2 and figure 5b. DHW operation of heat pump was 25.89% more efficient compared to space cooling operation, from the energy-based point of view.

The exergetic efficiency of the observed heat pump system recorded an increase with fuel exergy increase during the domestic water reheating operation, figure 6a, as expected because with an increase in total fuel and product exergy of the overall system increases significantly. Figure 6a shows that there was a growing trend in the exergy efficiency of the evaporator and compressor, while the exergy efficiency of the condenser first stagnated and then slightly decreased. In this regime, the average value of exergy efficiency was 33.38%, while the minimum and maximum values were 21.91% and 48.03%, table 2, figure 6a.

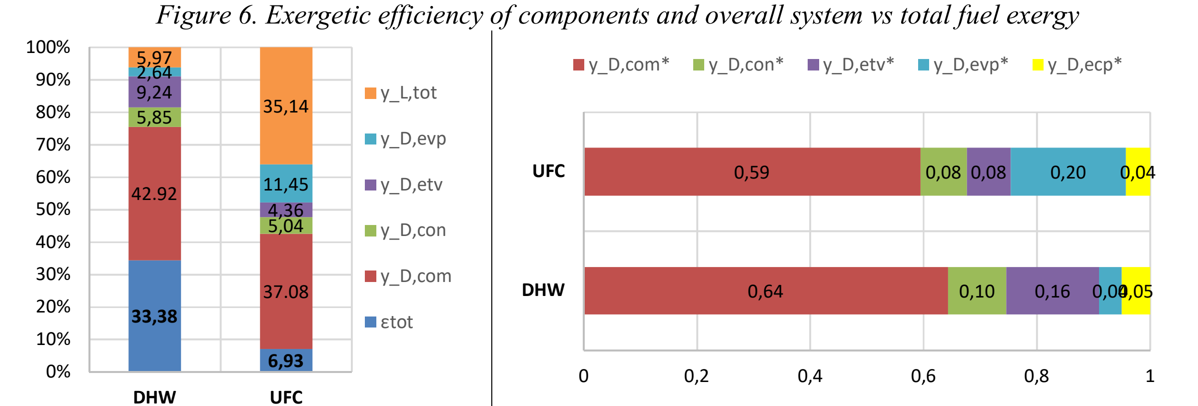
In the underfloor space cooling regime, exergetic efficiency fluctuated very little around the average value ($\bar{\varepsilon}_{\mathrm{tot,c}} = 6.93\%$) during the whole period. Characteristic in this regime was that the exergetic efficiency having a permanently low value ($\varepsilon_{tot,,max,c} = 8.19\%$, $\varepsilon_{tot,min,c} = 5.74\%$), which was a consequence of extremely low values of the product exergy of the whole system ($\bar{\dot{E}}_{\mathrm{P,tot,c}} = 0.06$ kW) compared to the corresponding fuel exergy ($\bar{\dot{E}}_{\mathrm{F,tot,c}} = 0.83$ kW) table 2, figure 6b. This was due to the oversized capacity for cooling purposes.

The average value of the exergetic efficiency during the DHW regime (33.38%) which corresponds to the results in the paper [11].



a) Domestic water reheating *b) Underfloor space cooling*

Figure 6. Exergetic efficiency of components and overall system vs total fuel exergy



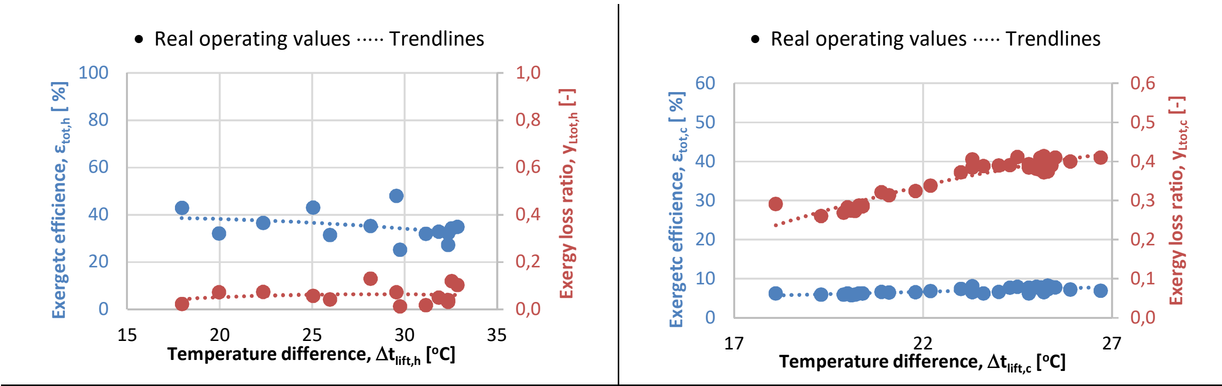
a) Exergy destruction ratios and efficiency *b) Total exergy destruction ratios*

Figure 7. Main second-law parameters for both regimes

On the other hand, that value was 26.45% larger compared to the exergetic efficiency during the UFC regime (6.93%), figure 7a. The conclusion is that air-to-water heat pump during DHW regime operates with more favorable operating conditions. During that operation, the compressor had the biggest impact on reducing exergetic efficiency, with the exergy destruction ratio of 42.92%, figure 7a.

During the domestic water reheating regime required small value of total fuel exergy (1.25kW) but the amount of exergy destruction generated was significant (0.74kW). During UFC regime 0.83kW of fuel exergy generated 0.47kW of exergy destruction, table 2. During UFC the exergy destruction ratio (37.08%) and the exergy loss coefficient (35.14%) have dominant effects on heat pump exergetic efficiency reduction, figure 7a. In both regimes, irreversibilities in the compressor have significantly influence on the exergetic efficiency of the heat pump. Influence of condenser, evaporator and electronic trotting valve for both regimes ranges from 2.64% - 11.45%, figure 7a. The circulating pump influence was neglected. The dominant influence of the compressor was also confirmed by the results of total exergy destruction ratios, 0.64 (DHW) and 0.59 (UFC), figure 7b. Figure 8 shows real operating values and trendlines for exergetic efficiency and exergy loss ratio as a function of the temperature lift from evaporation to condensation temperatures, for both observed modes of operation of the heat pump system.

In the DHW mode, the exergetic efficiency has a slight downward trend, and the exergy loss ratio has a slight upward trend with increasing difference between condensation and evaporation temperatures, figure 8a. In the UFC mode, the exergetic efficiency has a slight upward trend, while the upward trend of the exergy loss ratio is more pronounced, figure 8b.The reasons lie in the fact that in heating mode, a larger deviation is achieved between the condensation and evaporation temperatures, as well as a larger difference between the set and reference temperatures ($T_{DHW,sp}$ = 321K, $T_{UFC,sp}$ = 290K, T_0 = 298K).



a) Domestic water reheating *b) Underfloor space cooling*

Figure 8. Exergetic efficiency and exergy loss ratio vs temperature difference for DHW and UFC regimes

General recommendation is to choose the capacity of the air-to-water heat pump so that it can close to its optimal capacity in both regimes, and then reduce the exergy of flows 2, 5, 12 and 9, figures 1a and 2a.

5. Conclusions

The results of the analysis conducted based on first and second laws at component and overall system level show that observed low temperature residential air-to-water heat pump:

- From the energy point of view, in both regimes the heat pump operates with good performance ($\overline{\mathrm{COP}} = 4.75$; $\overline{\mathrm{EER}} = 3.52$),
- From the exergy point of view, in both regimes the heat pump operates with small value of efficiency, especially in the underfloor space cooling regime ($\bar{\varepsilon}_{\mathrm{tot,h}} = 33.38\%$; $\bar{\varepsilon}_{\mathrm{tot,c}} = 6.93\%$),
- The largest influence on exergetic efficiency during the domestic water reheating operation is attributed to the irreversibility in the compressor, and during the underfloor space cooling operations to the irreversibility in the compressor and exergy loss.

The capacity of observed the heat pump is enough for operating in DHW regime but over dimensioned for UFC operation. One solution and a suggestion for future research could be an option with two compressors of different capacities. The greatest impact on reducing overall exergetic efficiency has a compressor with 42.92% of useful work destroyed during DHW regime and 37.08% during UFC regime. Other components contribute to total exergy destruction with 17.73% (DHW) and 20.85% (UFC). Exergy loss reduces exergetic efficiency by 5.97% in the DHW regime and 35.14% in the UFC regime. To increase exergetic efficiency of residential air-to-water heat pump it is important to reduce internal irreversibility of the compressor while operating with lower capacities (especially in the cooling regime), reduce exergy losses to the environment, and operate at nominal conditions during all operating regimes.

Generally, the temperature lift from evaporation to condensation should have minimal values during both operating regimes. For future research, it is recommended how to use space cooling energy to DHW.

The scientific contribution of this manuscript is the confirmation obtained based on energy quality that the efficiency of an air-to-water heat pump depends greatly on the regime in which it operates.

Acknowledgments

This research was financially supported by the Ministry of Science, Technological Development and Innovation of the Republic of Serbia (Contract No. 451-03-65/2024-03).

Nomenclature

c	- specific heat capacity (kJ/kgK)
COP	- coefficient of performance (-)
e	- specific exergy (kJ/kg)
$\dot{E}$	- exergy flow rate (kW)
EER	- energy efficiency ratio (-)
h	- specific enthalpy (kJ/kg)
$\dot{m}$	- mass flow rate (kg/s)
N	- total number of measurements
p	- pressure (bar)
$\dot{Q}$	- heat transfer rate (kW)
s	- specific entropy (kJ/kgK)
$\dot{S}$	- entropy rate (W/K)
t	- temperature (°C)
T	- absolute temperature (K)
$\dot{W}$	- power, work rate (kW)
y	- exergy ratio (-)
Greek symbols	
Δ	- difference
ε	- exergetic efficiency
Subscripts	
c	- cooling
C	- Carnot, cold temperature reservoir
cv	- control volume
con	- condensation
D	- destruction
DHW	- domestic hot water
e	- outlet stream
evp	- evaporation
F	- fuel
gen	- generated
i	- inlet stream
in	- indoor
j	- stream
k	- system component
lift	- from evaporation to condensation
L	- loss
max	- maximal
min	- minimal
nom	- nominal
P	- product
q	- heat transfer
r	- refrigerant
sp	- set point
tot	- overall system
h	- heating
H	- hot temperature reservoir
UFC	- underfloor cooling
w	- water
0	- environment
Superscripts	
PH	- physical
*	- total
$\bar{}$	- arithmetic average

References

[1] Salkuti, S. R. (2022). Emerging and Advanced Green Energy Technologies for Sustainable and Resilient Future Grid. *Energies*, 15(18), 6667. https://doi.org/10.3390/en15186667

[2] Bhowmik, C., Zindani, D., Chatterjee, P., Marinkovic, D., & Šliogerienė, J. (2024). Evaluation of green energy sources: an extended fuzzy-todim approach based on schweizer-sklar and power averaging operators. *Facta Universitatis, Series: Mechanical Engineering*, Online First, doi: 10.22190/FUME240711042B

[3] Romanov D., Leiss B., (2022), Geothermal energy at different depths for district heating and cooling of existing and future building stock, *Renewable and Sustainable Energy Reviews*, 167, art. no. 112727, doi: 10.1016/j.rser.2022.112727

[4] Mishra A.R., Rani P., Cavallaro F., Alrasheedi A.F., (2023), Assessment of sustainable wastewater treatment technologies using interval-valued intuitionistic fuzzy distance measure-based MAICRA method, *Facta Universitatis, Series: Mechanical Engineering*, 21(3), pp. 359 - 386, doi: 10.22190/FUME230901034M

[5] Balo F., Ivanović B., Stević Ž., Ulutas A., Marinković D., Demir H.B., (2024) Optimal Configuration of Spatial Planning for Energy-Efficient Buildings, *Tehnicki Glasnik*, 18, pp. 45-54, doi: 10.31803/tg-20241013235011

[6] https://www.iea.org/energy-system/buildings/heat-pumps

[7] Wang Y., Ye Z., Song Y., Yin X., Cao F., Energy, exergy, economic and environmental analysis of refrigerant charge in air source transcritical carbon dioxide heat pump water heater, *Energy Conversion and Management, 223* (2020), 113209, pp. 1–13

[8] Szargut J., Component efficiencies of vapour-compression heat pump, *Exergy, an International Journal 2 (2002),* pp. 99-104

[9] Shen B., Hu Y., Development and field demonstration of residential air source integrated heat pump using a three-stage compressor, *Energy and Buildings, 328* (2025), 115202, pp. 1-13

[10] Yumrutas R., Kunduz M., Kanoglu M., Exergy analysis of vapor compression refrigeration systems, *Exergy, an International Journal 2 (2002),* pp. 266-272

[11] Bilgen E., Takahashi., Exergy analysis and experimental study of heat pump systems, *Exergy, an International Journal 2 (2002),* pp. 259-265

[12] Morosuk T., Nikulshin R., Morosuk L., Entropy-cucle method for analysis of refrigeration machine and heat pump cycles, *Thermal Science, 10* (2006), 1, pp. 111-124

[13] Ahamed U.J., Saidur R., Masjuki H.H., A review on exergy analysis of vapor compression refrigeration system, *Renewable and Sustainable Energy Reviews, 15* (2011), pp. 1593–1600

[14] Byme P., Ghoubali R., Exergy analysis of heat pumps for simultaneous heating and cooling, *Applied Thermal Engineering, 149* (2019), pp. 414–424

[15] Kropp M., Lammle M., Herkel S., Henning H.M.,Schafer A.V., Enhancing heat pump performance for domestic hot water preparation: A comparative analysis in existing multi-family houses, *328* (2025), 115155, pp. 1-15

[16] https://www.scienceeurope.org/media/0vxhcyhu/se_exergy_brochure.pdf

[17] Bejan A., Tsatsaronis G., Moran M., *Thermal design and optimization*, John Wiley & Sons, Inc., New York, 1996

[18] Moran M., Shapiro H., *Fundamentals of engineering thermodynamics*, John Wiley & Sons Ltd., West Sussex, 2006

[19] Mergenthaler P., Schinkel A.P., Tsatsaronis G., Application of exergoeconomic, exergoenvironmental, and advanced exergy analysis to Carbon Black production, *Energy, 137* (2017), pp. 898-907

[20] https://www.daikin.rs/sr_rs/products/ERLQ-CV3.table.html

[21] http://www.coolprop.org/index.html

[22] Ristić B., *Metrology, Flow-technical measurements (in Serbian),* Prosveta, Niš, 1981.

Paper submitted: 17.01.2025
Paper revised: 03.03.2025
Paper accepted: 11.03.2025