Original scientific paper

Ključne reči: toplotna pumpa; grejanje; toplotni izvor; eksergijska analiza; energijska analiza

Key words: heat pump; heating; heat source; exergy analysis; energy analysis

TERMODINAMIČKA ANALIZA RADA ČETIRI RAZLIČITA GREJNA SISTEMA TOPLOTNE PUMPE

THERMODYNAMIC ANALYSIS OF THE OPERATION OF FOUR DIFFERENT HEAT PUMP HEATING SYSTEMS

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U radu je prikazana termodinamička (energijska i eksergijska) analiza rada četiri različita sistema toplotnih pumpi: voda-voda, voda-vazduh, vazduh-voda i vazduh-vazduh, koji se koriste za potrebe grejanja. Pretpostavljeno je da su snage grejanja sve četiri toplotne pumpe međusobno jednake, kao i temperaturni režim rada grejnog sistema. U zavisnosti od uticajnih parametara, i to temperature toplotnog izvora i promene temperature toplotnog izvora na isparivaču toplotne pumpe, analizirane su promene koeficijenta grejanja i eksergijskog stepena korisnosti toplotnih pumpi. Ustanovljeno je da toplotne pumpe sa podzemnom vodom kao izvorom toplote imaju veći COP i eksergijski stepen korisnosti od toplotnih pumpi sa okolnim vazduhom kao izvorom toplote. Sprovedena analiza pruža smernice pri odabiru odgovarajuće toplotne pumpe i optimizaciji njenog rada.

The paper presents a thermodynamic (energy and exergy) analysis of the operation of four different heat pump systems: water-to-water, water-to-air, air-to-water and air-to-air, which are used for heating purposes. It was assumed that the heating capacities of all four heat pumps are equal to each other, as well as the operating temperature regime of the heating system. Depending on the influencing parameters, namely the temperature of the heat source and changes in the temperature of the heat source on the evaporator of the heat pump, the changes in the heating coefficient and the exergy degree of usefulness of the heat pumps were analyzed. It was found that heat pumps with groundwater as a heat source have a higher COP and exergy efficiency than heat pumps with ambient air as a heat source. The conducted analysis provides guidelines for selecting a suitable heat pump and optimizing its operation.

1.Introduction

More than 50% of the total energy consumed by end users, primarily households and industrial facilities, is used for heating and cooling purposes [1]. In this context, heat-related CO_2 emissions from buildings reached 4.9 Gt globally in 2020 [2], accounting for about 15% of total global carbon emissions [3]. Consequently, countries around the world have strengthened policy goals and introduced financial incentives to support the transition to clean heating, focusing on reducing dependence on fossil fuels and promoting technologies that utilise renewable energy sources. One of the most

promising technical solutions in this regard is the application of heat pumps. This technology enables a significant reduction in energy consumption for heating and cooling, as well as a decrease in harmful gas emissions, thereby contributing to both economic and environmental sustainability.

The first functional heat pump was developed in the mid-20th century, but its commercial application became widespread only toward the end of the 20th and the beginning of the 21st century, driven by rising energy prices and the growing need to improve energy efficiency. The primary purpose of heat pumps is for heating, using electrical energy to transfer heat from the environment into the indoor space of a building. Their operation requires significantly less input power or energy to achieve the same heating output compared to conventional systems of equivalent heating capacity. The ratio between the amount of heat delivered to the heated space and the input energy required to drive the heat pump is known as the Coefficient of Performance (COP), which for modern heat pumps can reach values between 3 and 5, or even higher.

Depending on the heat source they use, heat pumps are classified into several types. Air-to-water and air-to-air heat pumps use outdoor air as the heat source. They are the most commonly used systems due to their simple installation and lower investment costs, but they perform somewhat less efficiently under extremely cold conditions. Water-to-water and water-to-air heat pumps utilise water from underground aquifers, lakes, or rivers as the heat source. Their operation is highly efficient but requires access to water resources and regulated water usage rights. Ground-to-water heat pumps transfer heat from the earth through geothermal collectors. They provide a very stable and high coefficient of performance throughout the year, but require significant investment and construction work.

Numerous studies have examined and compared the efficiency of two or more types of heat pumps, as well as their performance relative to conventional heating systems. After reviewing various geothermal heat pumps, Self et al. [4] analysed the efficiency, cost, and CO_2 emissions of different heating systems. They compared geothermal heat pumps, air-to-water heat pumps, electric floor heaters, and natural gas furnaces, and found that geothermal heat pumps consume significantly less energy for space heating than alternative systems. Therefore, heat pumps are considered highly efficient and economically viable heating solutions. Luo et al. [5] evaluated the potential of different heat pump systems in the urban area of Wuhan, China. They analysed geothermal heat pumps using groundwater and subsurface soil layers as heat sources, heat pumps using surface water as the heat source, and air-to-water heat pumps. Their findings indicated that geothermal heat pumps operate with higher efficiency due to the more stable temperatures found in deeper soil layers compared to shallow aquifers or ambient air. Çakır et al. [6] also compared different types of heat pumps using an experimental setup installed at the Faculty of Mechanical Engineering, Atatürk University in Erzurum, Turkey, which could operate in four different modes: air-to-air, air-to-water, water-to-air, and water-to-water. For the same heating capacity across all four systems, it was found that the water-to-air heat pump achieved the highest *COP* and the highest exergy efficiency. Maddah et al. [7] conducted an energy, exergy, environmental, and economic assessment of ground-source (GSHP) and air-source (ASHP) heat pumps. The results showed that the coefficient of performance and exergy efficiency values were higher for the GSHP cycle than for the ASHP cycle, while for both cycles, the refrigerant R134a yielded the best results. The GSHP system also demonstrated advantages in terms of both economic and environmental aspects. Akpinar and Hepbasli [8] presented an exergy-based performance evaluation model for two geothermal heat pump (GSHP) systems, which can serve as a useful tool for assessing other GSHP systems worldwide.

In this study, an energy and exergy analysis was carried out for four different heat pump heating systems: water-to-water, water-to-air, air-to-water, and air-to-air. It was assumed that all four heat

pumps have the same heating capacity and operate under identical temperature conditions of the heating system. The variations in the Coefficient of Performance (*COP*) and the exergy efficiency of the heat pumps were analysed as functions of key influencing parameters, the temperature of the heat source, and the temperature variation at the evaporator of the heat pump.

2. Thermodynamic analysis of the heat pump system

The heating system of a heat pump consists of three subsystems: the external subsystem of the plant, which connects the system to the heat source; the heat pump itself, serving as the central subsystem, and the internal subsystem, which represents the building's heating system. The analysis was carried out on the basic model of a heat pump, whose main components include two heat exchangers, the evaporator and the condenser, as well as a compressor and an expansion valve (Figure 1). The external fluid, such as groundwater or ambient air, flows through the evaporator, transfers heat to the working fluid of the heat pump, cools down, and is then returned to the environment.

On the other side, the return heating fluid from the building's heating system passes through the condenser, absorbs heat from the working fluid of the heat pump, and becomes heated. Once heated to the supply temperature, this fluid circulates through the internal subsystem of the plant, transferring heat to the building. Depending on the type of heat pump system, either water or air can be used as the heating fluid. The compressor raises the pressure of the refrigerant from the evaporation pressure to the condensation pressure by performing work input, while the expansion valve reduces the refrigerant pressure from the condensation pressure to the evaporation pressure.

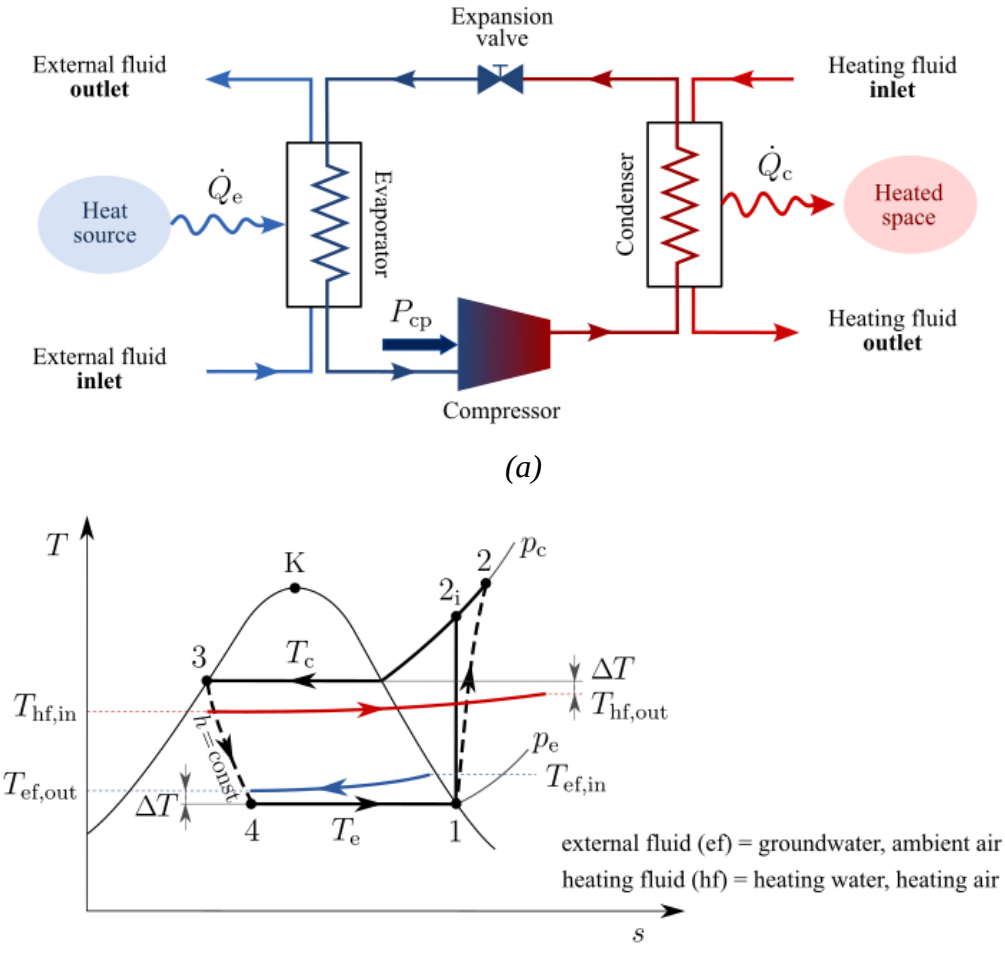

Figure 1. (a) Schematic representation of the heat pump cycle and (b) its T–s diagram

For all four analysed heat pump systems, identical heating capacities of 50 kW and the same operating temperature regime of the heating system, 35/30°C, were adopted. The operating parameters of the heat pump were defined based on the selected temperature conditions of the external and internal subsystems. It was assumed that the evaporation temperature of the heat pump's working fluid is Δt = 2°C lower than the outlet temperature of the external fluid from the evaporator, while the condensation temperature of the working fluid is Δt = 2°C higher than the outlet temperature of the heating fluid from the condenser. The working fluid of the heat pump is R-410A refrigerant. The flow of the external and internal heating fluids is ensured by the operation of circulation pumps and/or fans. Since the electrical power required for their operation is negligibly small compared to the power consumed by the compressor, it was neglected in the present analysis.

2.1. Energy analysis

The heat flow rate transferred from the external fluid, groundwater (gw) or ambient air (aa), acting as the heat source ($\dot{Q}_{gw}$, $\dot{Q}_{aa}$), to the working fluid in the evaporator of the heat pump is determined using the following expression:

$$\left|\dot{Q}_{gw}\right| = \dot{m}_{gw} c_{p,gw} (T_{gw,in} - T_{gw,out}) \quad (1)$$

$$\left|\dot{Q}_{aa}\right| = \dot{m}_{aa} c_{p,aa} (T_{aa,in} - T_{aa,out}) \quad (2)$$

where gde $\dot{m}_{gw}$, $c_{p,gw}$, $T_{gw,in}$ and $T_{gw,out}$, or $\dot{m}_{aa}$, $c_{p,aa}$, $T_{aa,in}$ and $T_{aa,out}$, denote the mass flow rate (kg/s), the specific heat capacity at constant pressure (kJ/(kg·K)), and the inlet (K) and outlet (K) temperatures of the external fluid at the evaporator, respectively.

On the other hand, that same heat flow rate is supplied to the working fluid in the evaporator for complete evaporation ($\dot{Q}_e$) and is given by:

$$\dot{Q}_e = \dot{m}_R (h_1 - h_4) \quad (3)$$

where $\dot{m}_R$, h_1 and h_4 denote the mass flow rate (kg/s), the specific enthalpy at state 1 (kJ/kg), and the specific enthalpy at state 4 (kJ/kg) of the refrigerant, respectively.

For the compressor operation, the (electrical) power input is:

$$P_{cp} = \dot{m}_R (h_2 - h_1) \quad (4)$$

where h_2 denotes the specific enthalpy of the refrigerant at state 2 (kJ/kg). The internal efficiency of the compressor ($\eta_{S,cp}$) is defined as the ratio of the technical work in isentropic compression (1-2i) to the technical work in the actual adiabatic, non-quasi-static compression (1-2), both performed from the same initial state to the same final pressure:

$$\eta_{S,cp} = \frac{h_{2i} - h_1}{h_2 - h_1} \quad (5)$$

Its value is determined as a function of the condensation pressure (p_c) and the evaporation pressure (p_e) using the following expression [9]:

$$\eta_{S,cp} = 0,85 - 0,0467 \frac{p_c}{p_e} \quad (6)$$

The heat flow rate received by the heating fluid, either heating water (hw) or heating air (ha), while passing through the condenser ($\dot{Q}_{\mathrm{hw}}, \dot{Q}_{\mathrm{ha}}$), and subsequently transferred to the heated space as a heat sink, is given by:

$$\dot{Q}_{\mathrm{hw}} = \dot{m}_{\mathrm{hw}} c_{p,\mathrm{hw}} (T_{\mathrm{hw,out}} - T_{\mathrm{hw,in}}) \tag{7}$$

$$\dot{Q}_{\mathrm{ha}} = \dot{m}_{\mathrm{ha}} c_{p,\mathrm{ha}} (T_{\mathrm{ha,out}} - T_{\mathrm{ha,in}}) \tag{8}$$

Where $\dot{m}_{\mathrm{hw}}$, $c_{p,\mathrm{hw}}$, $T_{\mathrm{hw,in}}$ and $T_{\mathrm{hw,out}}$, or $\dot{m}_{\mathrm{ha}}$, $c_{p,\mathrm{ha}}$, $T_{\mathrm{ha,in}}$ and $T_{\mathrm{ha,out}}$, denote the mass flow rate (kg/s), the specific heat capacity at constant pressure (kJ/(kg·K)), and the inlet (K) and outlet (K) temperatures of the heating fluid leaving the condenser, respectively.

That same heat flow rate is removed from the working fluid ($\dot{Q}_{\mathrm{c}}$) during its cooling and complete condensation in the condenser:

$$\left|\dot{Q}_{\mathrm{c}}\right| = \dot{m}_{\mathrm{R}} (h_2 - h_3) = \dot{Q}_{\mathrm{e}} + P_{\mathrm{cp}} \tag{9}$$

The Coefficient of Performance (*COP*) of the basic heat pump cycle represents the ratio between the heat flow removed from the working fluid, that is, the heat transferred to the heated medium, heating power ($\dot{Q}_{\mathrm{h}}$), and the power required to drive the compressor:

$$COP = \frac{\dot{Q}_{\mathrm{h}}}{P_{\mathrm{cp}}} = \frac{\left|\dot{Q}_{\mathrm{c}}\right|}{P_{\mathrm{cp}}} \tag{10}$$

2.2. Exergy analysis

Energy analysis does not provide information about the quality of energy flows within a system. Therefore, to perform a comprehensive thermodynamic evaluation, it is necessary to conduct an exergy analysis as well. The change in exergy (per unit time) of the external fluid, groundwater ($\Delta\dot{E}x_{\mathrm{gw}}$) or ambient air ($\Delta\dot{E}x_{\mathrm{aa}}$) is expressed as:

$$\Delta\dot{E}x_{\mathrm{gw}} = \dot{m}_{\mathrm{gw}} (c_{p,\mathrm{gw}} (T_{\mathrm{gw,in}} - T_{\mathrm{gw,out}}) - T_0 c_{p,\mathrm{gw}} \ln \frac{T_{\mathrm{gw,in}}}{T_{\mathrm{gw,out}}}) \tag{11}$$

$$\Delta\dot{E}x_{\mathrm{aa}} = \dot{m}_{\mathrm{aa}} (c_{p,\mathrm{aa}} (T_{\mathrm{aa,in}} - T_{\mathrm{aa,out}}) - T_0 c_{p,\mathrm{aa}} \ln \frac{T_{\mathrm{aa,in}}}{T_{\mathrm{aa,out}}}) \tag{12}$$

while the change in exergy (per unit time) of the heating water ($\Delta\dot{E}x_{\mathrm{hw}}$) or air in the heating system ($\Delta\dot{E}x_{\mathrm{ha}}$) is determined using the following expression:

$$\Delta\dot{E}x_{\mathrm{hw}} = \dot{m}_{\mathrm{hw}} (c_{p,\mathrm{hw}} (T_{\mathrm{hw,out}} - T_{\mathrm{hw,in}}) - T_0 c_{p,\mathrm{hw}} \ln \frac{T_{\mathrm{hw,out}}}{T_{\mathrm{hw,in}}}) \tag{13}$$

$$\Delta\dot{E}x_{\mathrm{ha}} = \dot{m}_{\mathrm{ha}} (c_{p,\mathrm{ha}} (T_{\mathrm{ha,out}} - T_{\mathrm{ha,in}}) - T_0 c_{p,\mathrm{ha}} \ln \frac{T_{\mathrm{ha,out}}}{T_{\mathrm{ha,in}}}) \tag{14}$$

For the environmental reference state, outdoor air at an atmospheric pressure of p_0 = 1 atm and temperature t_0 is adopted. When groundwater is the heat source, t_0 = 0°C, while in the case when ambient air is the heat source, t_0 is taken equal to the ambient air temperature. In determining the

specific heat capacity of groundwater and heating water, its temperature dependence was taken into account [10], whereas for ambient air and supply air, a constant value was adopted, $c_{p,aa}=c_{p,ha}=1004.9$ J/(kgK).

The exergy efficiency of the heat pump system at the boundary of the building's heating system (η_{ex}) is determined using the following expression:

-The water-to-water type $$\eta_{ex,gw-hw} = \frac{\Delta \dot{E}x_{hw}}{\Delta \dot{E}x_{gw} + P_{cp}} \quad (15)$$

-The water-to-air type $$\eta_{ex,gw-ha} = \frac{\Delta \dot{E}x_{ha}}{\Delta \dot{E}x_{gw} + P_{cp}} \quad (16)$$

- The air-to-water type $$\eta_{ex,aa-hw} = \frac{\Delta \dot{E}x_{hw}}{\Delta \dot{E}x_{aa} + P_{cp}} \quad (17)$$

- The air-to-air type $$\eta_{ex,aa-ha} = \frac{\Delta \dot{E}x_{ha}}{\Delta \dot{E}x_{aa} + P_{cp}} \quad (18)$$

2.3. Methodology

The analysis was carried out through a series of independent calculations in which variations of the specified energy and exergy parameters of the system were monitored. The parameters whose influence on the efficiency factors was analysed are:

- temperature of the groundwater at the evaporator inlet ($t_{gw,in}$ = 10°C; 12°C; 15°C),
- temperature of the ambient air at the evaporator inlet ($t_{aa,in}$ = -5°C; 0°C; 5°C),
- temperature change of the groundwater and ambient air across the evaporator of the heat pump (Δt_{gw}, Δt_{aa} = 2.5°C; 5°C).

Based on the known inlet temperature of the external fluid at the evaporator and the prescribed temperature change across the heat pump evaporator, the outlet temperature of the external fluid from the evaporator was determined. The evaporation temperature of the heat pump working fluid was taken to be 2°C lower than the outlet temperature of the external fluid from the evaporator. Conversely, the condensation temperature was taken to be 2°C higher than the known temperature of the heating fluid at the condenser outlet. In all calculation series, the condensation temperature had a constant value of t_{kd} = 37°C. From the known phase-change temperatures, the thermodynamic state properties of the refrigerant at the characteristic states were obtained. Once these are known, and given the specified heating capacity of the heat pump, it is possible to determine: the mass flow rates of the external fluid, the heating fluid, and the refrigerant, the compressor power required to drive the heat pump, and subsequently the change in exergy of the external fluid and the heating fluid. On the basis of these quantities, the Coefficient of Performance (*COP*) and the exergy efficiency of the analysed heat pump system were evaluated.

3. Results and discussion

For the same heating capacity and identical operating temperature regime of the heating system, the type of heating fluid, whether heating water or heating air, has no influence on the energy and exergy characteristics of the heat pump. Considering this, the presentation of results is divided into two groups of heat pump systems: those using groundwater as the heat source (water-to-water and

water-to-air heat pumps) and those using ambient air as the heat source (air-to-water and air-to-air heat pumps).

First, an energy analysis was performed for both groups of heat pump systems. Based on the known heating capacity of the heat pump and the enthalpy values of the refrigerant at characteristic states, the mass flow rate of the working fluid was determined. In the next step, the power required to drive the compressor was calculated, followed by the determination of the Coefficient of Performance (*COP*) of the heat pump cycle. The results of the energy analysis are shown in Figure 2.

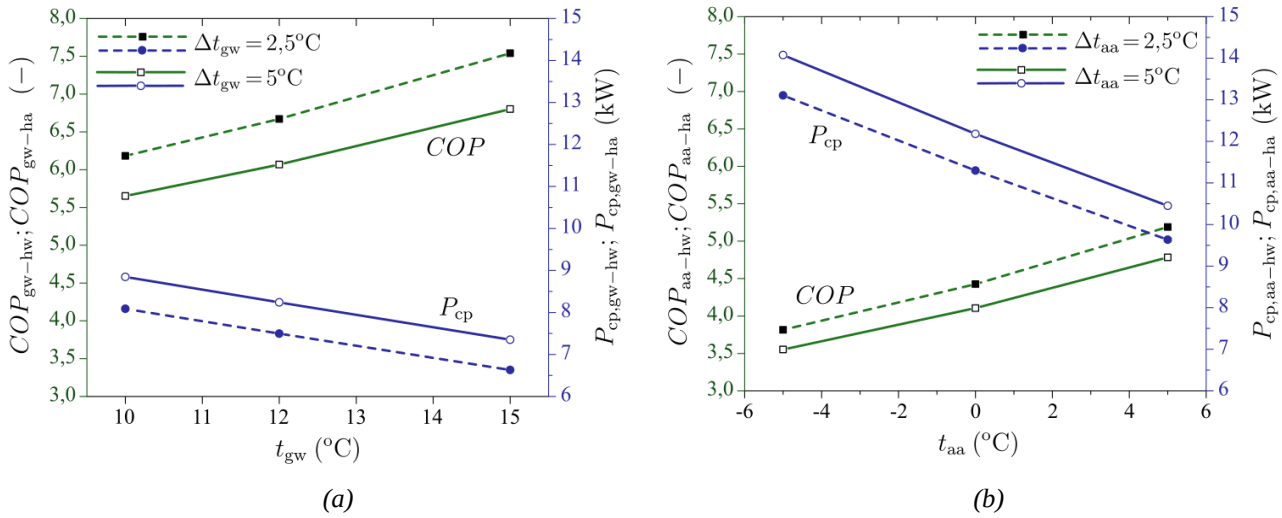

Figure 2. Coefficient of performance of heat pumps with (a) groundwater and (b) ambient air as the heat source

As the temperature of the heat source increases, the evaporation temperature of the refrigerant also rises. At a constant condensation temperature and heating capacity, this results in a decrease in compressor power demand, leading to an increase in the heat pump's Coefficient of Performance (*COP*). From the figure, it can also be observed that a reduction in the temperature difference of the external fluid across the evaporator leads to a further decrease in compressor power and an increase in *COP*. In the analysed temperature ranges, heat pumps using groundwater as the heat source exhibit higher *COP* values than those using ambient air. When the groundwater temperature increases from 10°C to 15°C, the *COP* of the water-to-water and water-to-air heat pumps increases from 5.65 to 6.80 Δt_{gw}=5°C, and from 6.18 to 7.54 for Δt_{gw}=2.5°C. On the other hand, when the ambient air temperature increases from −5 °C to 5 °C, the *COP* of the air-to-water and air-to-air heat pumps increases from 3.55 to 4.78 for Δt_{aa}=5°C, and from 3.82 to 5.19 for Δt_{aa}=2.5°C.

In order to perform the exergy analysis of the heat pump heating systems, it was first necessary to determine the mass flow rates of the external and heating fluids (Figure 3).

From Figure 3, it can be seen that the mass flow rates of the heating fluids remain constant for all analysed cases. This is a consequence of the constant heating capacity of the heat pump and the constant temperature change of the heating fluid at the condenser (Δt_{hw}=Δt_{ha}=5°C). Due to the difference in specific heat capacities, the mass flow rate of heating water is lower than that of heating air ($\dot{m}_{hw} = 2.39\,\text{kg/s}$, $\dot{m}_{ha} = 9.95\,\text{kg/s}$). As the refrigerant mass flow rate increases with rising heat source temperature, the heat transfer rate in the evaporator increases, which in turn causes a rise in the mass flow rate of the external fluid. The same trend occurs when the temperature difference of the external fluid across the evaporator decreases.

Thus, within the analysed temperature range, the mass flow rate of groundwater varies between 1.96 kg/s and 2.04 kg/s for Δt_{gw}=5°C, and between 4.00 kg/s and 4.14 kg/s for Δt_{gw}=2.5°C. The mass

flow rate of ambient air lies in the range of 7.15–7.87 kg/s for Δt_{aa}=5°C, and 14.69–16.07 kg/s for Δt_{aa}=2.5°C.

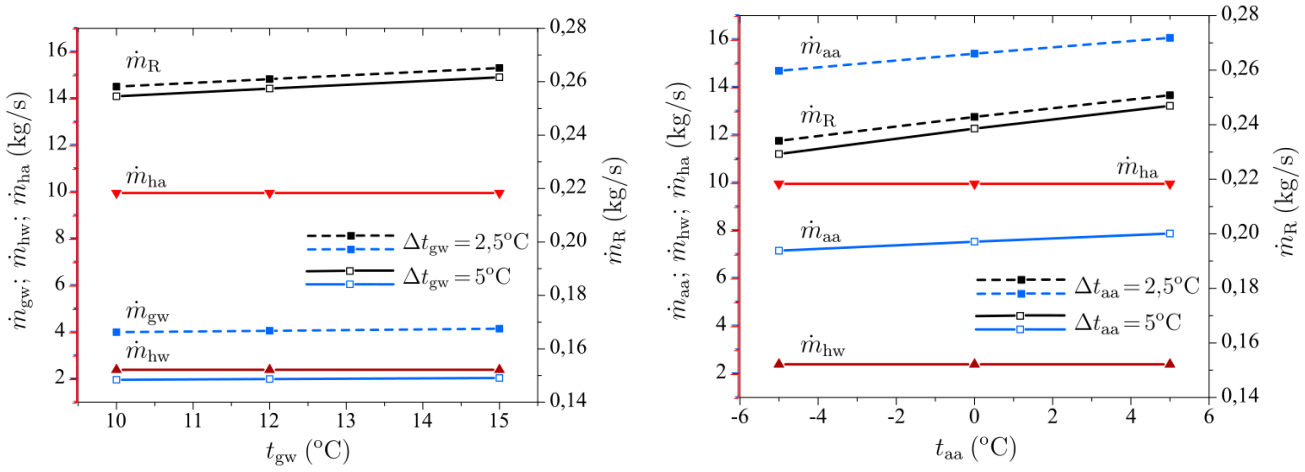

Figure 3. Mass flow rates of the external and heating fluids of heat pumps with (a) groundwater and (b) ambient air as the heat source

Figure 4 presents the exergy variation of the external and heating fluids, as well as the exergy efficiency of the analyzed heat pump system.

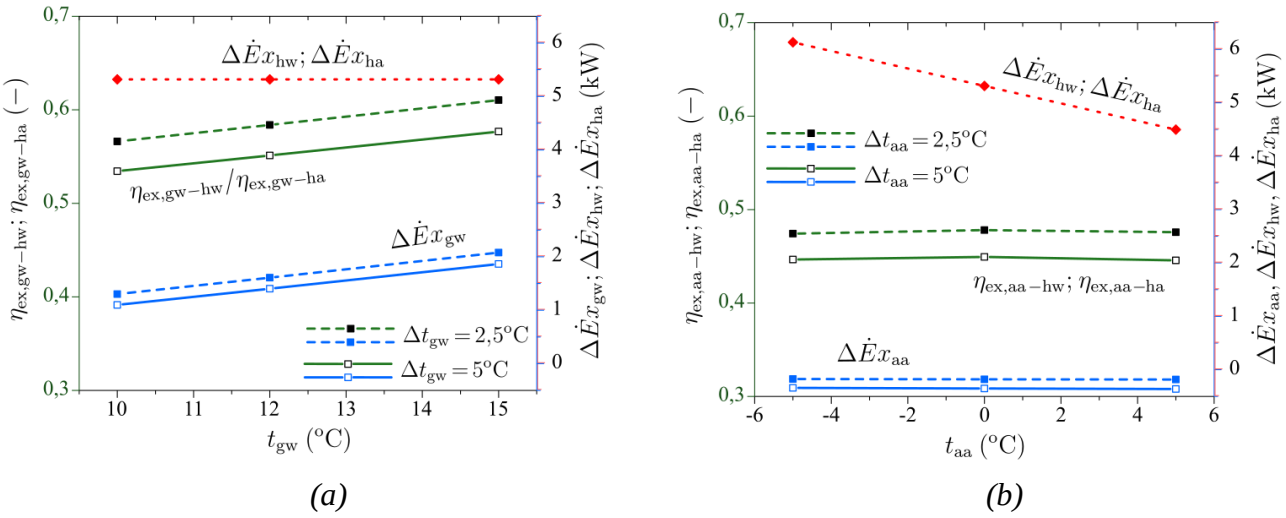

Figure 4. Exergy efficiency of heat pumps with (a) groundwater and (b) ambient air as the heat source

From Figure 4, it can be observed that within the analysed temperature ranges and under the adopted ambient conditions, heat pumps using groundwater as the heat source exhibit higher exergy efficiency, which increases with rising groundwater temperature. The exergy efficiency of heat pumps that use ambient air as the heat source remains approximately constant. In addition to the change in the exergy of the external and heating fluids, the exergy efficiency is also influenced by the compressor power demand (as shown in Figure 2). Because air-source heat pumps require higher compressor power, their exergy efficiency values are lower. For both groups of heat pumps, the exergy efficiency increases as the temperature difference of the external fluid across the evaporator decreases. When the groundwater temperature increases from 10 °C to 15 °C, the exergy efficiency η_{ex} of water-to-water and water-to-air heat pumps rises from 0.534 to 0.577 for Δt_{gw} = 5°C, and from 0.566 to 0.610 for Δt_{gw}

= 2.5°C. For air-to-water and air-to-air heat pumps the maximum exergy efficiency value is 0.449 for Δt_{aa} = 5°C, and 0.478 for Δt_{aa}=2.5°C, at an ambient air temperature of 0°C.

4. Conclusion

In this study, a thermodynamic analysis was conducted for four different heat pump heating systems: water-to-water, water-to-air, air-to-water, and air-to-air. Based on the performed thermodynamic analysis, the following conclusions can be drawn:

- For the same heating capacity and identical operating temperature regime of the heating system, the type of heating fluid (water or air) has no significant influence on the energy and exergy performance characteristics of the heat pump.
- As the temperature of the heat source increases, the power required to drive the compressor decreases, resulting in a higher Coefficient of Performance (*COP*) of the heat pump.
- A reduction in the temperature difference of the external fluid across the evaporator leads to a decrease in compressor power and an increase in the *COP*.
- Within the analysed temperature range, heat pumps using groundwater as the heat source exhibit higher *COP* values than those using air as the heat source.
- Under the analysed temperature conditions and for the adopted ambient temperatures, heat pumps with groundwater as the heat source achieve a higher exergy efficiency, which increases with rising groundwater temperature.
- The exergy efficiency of heat pumps using air as the heat source remains approximately constant within the analysed temperature range.

5. Acknowledgement

The results presented in this paper are part of research supported by the Ministry of Science, Technological Development and Innovation of the Republic of Serbia, under Contract No. 451-03-137/2025-03/200105, dated February 4, 2025.

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