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Thermodynamic Analysis of the Design of a Heat Pump for Heat Recovery in a Biomass Heating Network

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ABSTRACT

The efficiency of biomass heating networks has become an increasingly important issue in recent years. Studies have shown that the most effective measure for increasing efficiency is to install a heat pump along with a flue gas condenser in the boiler of a biomass heating system. With a view to further enhancing efficiency, this study focuses on the thermodynamic analysis and optimization of a heat pump design used for heat recovery in a biomass heating network and its impact on the overall system. The study demonstrates that the choice of target exhaust gas temperature has a significant impact on both heat pump sizing and overall system efficiency. There is an optimum exhaust gas temperature at which the maximum overall system efficiency can be achieved. The supply temperature of the heat pump and the choice of refrigerant have little effect on the system overall, but they do affect the size of the heat pump. The supply temperature should be as low as possible. R600a is recommended as a refrigerant for the heating networks studied, due to its physical properties and beneficial behavioral characteristics at different temperatures.

Keywords: biomass district heating, heat pump design, integration, heat recovery

1. INTRODUCTION

By the end of the last century, there was widespread intensive construction of biomass heating plants throughout Europe. However, as fuel costs were relatively favorable, comparatively little importance was placed on the efficiency of the conversion of fuel energy to heating energy in the design of the plant. As a result, there are now many older heating plants and local heating networks operating at low efficiency. The average annual rate of conversion efficiency in biomass heating networks, from the initial fuel to the final energy supplied to the consumer, is currently between 55% and 80% on [1,2]. Due to the increase in biomass energy prices, it has become increasingly important to increase the efficiency of biomass heating plants so as to reduce the biomass costs.

The combination of a flue gas condenser and a heat pump is recommended in [3,4], to compensate for the disadvantage in biomass boiler efficiency. By cooling either the flue gas in the condenser or the network return temperature, the heat pump ensures that the flue gas temperature falls reliably below the dew point temperature, enabling the latent heat to be used. The heat gained at a low-temperature level is raised to a higher temperature level such that it can be used by the local heating network.

Zajacs et al. 2020 [5] studied the theoretical feasibility of this integration concept with a natural gas boiler in the form of a simple model and evaluated the efficiency increase. This showed that the concept can be effectively applied in district heating systems to improve flue gas condenser efficiency. In the case of biomass heating, the potential increase in energy depends greatly on the fuel water content [6]. The higher the water content, the greater the potential of efficiency increase from exhaust gas. In this way, the biomass boiler differs from the gas boiler. Hebenstreit et al. 2014 [7] evaluated the condensation system with an integrated heat pump together with a quench for heat recovery in a biomass boiler, using a simple heat pump model with a fixed Carnot factor. However, the study did not analyze the effects of the heat pump cycle design or its influence on the heating system as a whole.

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In a standard heating plant with no heat pump, the system efficiency is higher when the exhaust gas temperature is lower. However, the power consumption of an integrated heat pump also has an effect on the system's efficiency. The aim of this study is to analyze the technical parameters that have a significant effect on heat pump design and system efficiency, on the basis of a detailed thermodynamic model.

2. METHODOLOGY

2.1 Technical concept

To enable an in-depth analysis, a Python-based model of a biomass heating network was proposed, which enables a heat pump to be integrated into a biomass heating network. The main components of the model are the boiler (including the combustion chamber), heat exchangers (such as an air preheater, economizer, and flue gas condenser), and the heat pump. The development of these modules was based on the thermodynamic behavior of a number of substances in biomass fuel and flue gas. For simulation purposes, the modules can be connected as shown in the integration concepts and hydraulic circuits.

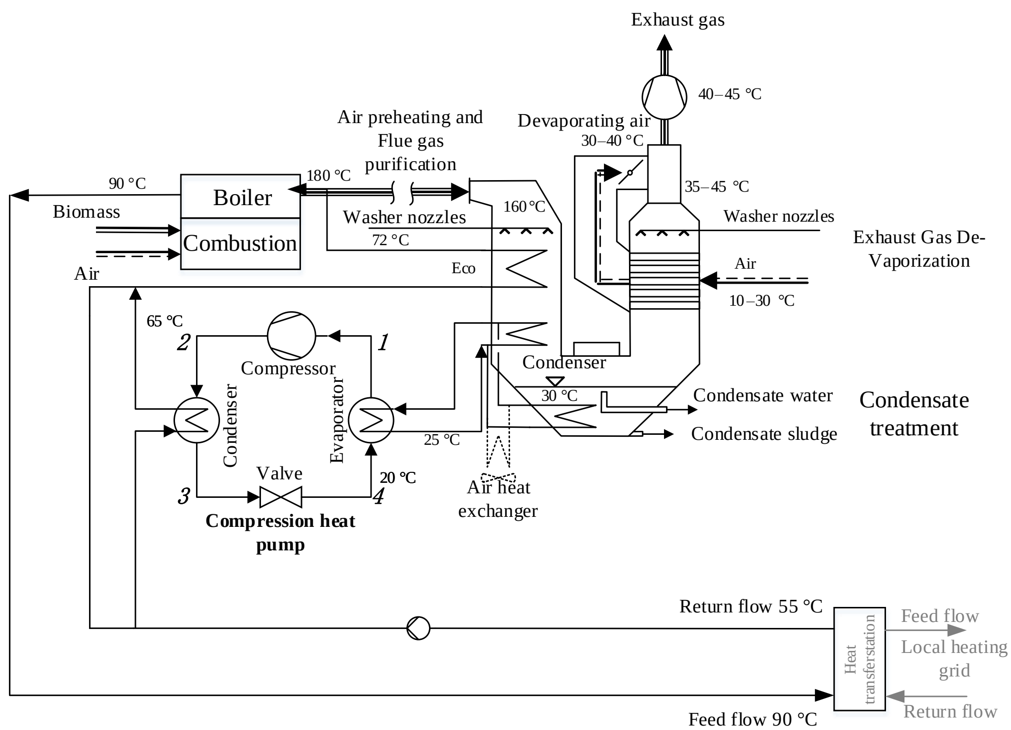

Figure 1: Technical concept of integrating an electrically driven compression heat pump into a flue gas condenser

This paper investigates the concept of the integration of an electric compression heat pump into a flue gas condenser, as illustrated in Figure 1. To enable the energy and mass flows of the heating plant to be simulated as realistically as possible, a detailed combustion formula that considers CO and NOx in the flue gas was used.

2.2 Heat pump design

The design of a heat pump should take into account the thermodynamic cycles of various refrigerants at different temperature levels. The heat pump system was simplified into four main components, as shown in Figure 1. To solve the thermodynamic cycle, the thermal properties of different refrigerants were imported from the database of CoolProp [8].

Table 1: Procedure for calculating the thermodynamic cycle

Nodes	Known parameters	Solved parameters
init	$T_e = T_{out}^{Ex.gas} - \Delta T_e$ $T_c = T_{supply}^{hp} + \Delta T_c$ *refrigerant*	p_e, p_c
1'	$T_e,\ p_e$, q=1	-
1	$p_e, T_1 = T + \Delta T_{sh}$	$h_1,\ s_1,\ q_1$
2	$p_c,\ s_1,\ \eta_{is}$	$h_2, T_2,\ q_2$
2"	p_c,q=1	$h_{2"},\ s_{2"}$
2'	p_c,q=0	$h_{2'},\ s_{2'}$
3	$p_c, T_3 = T_e + \Delta T_{sc}$	$h_3,\ s_3,\ q_3$
4	$h_3,\ T_e,\ \eta_{iso}$	$h_4,\ s_4,\ q_4$

The procedure for calculating the thermodynamic cycle is summarized in Table 1, where the numbers correspond to the points in the heat pump cycle in Figure 1.

For a defined cycle, the refrigerant flow $\dot{m}_{ref}$, electricity demand P_{el}, heat output $\dot{Q}_{hp}$, and COP of the heat pump is determined as follows:

$$\dot{m}_{ref} = \frac{\dot{Q}_{cond} * \eta_{hex}}{h_1 - h_4} \quad (1)$$

$$P_{el} = \frac{\dot{m}_{ref} * (h_2 - h_1)}{\eta_m * \eta_{el}} \quad (2)$$

$$\dot{Q}_{hp} = \dot{m}_{ref} * (h_2 - h_3) * \eta_{hex} \quad (3)$$

$$COP = \frac{\dot{Q}_{hp}}{P_{el}} \quad (4)$$

2.3 System performance analysis

The “figure of merit”, suggested by Rosen et al. [9], was adopted in this paper to indicate the overall efficiency of the system. This figure describes the overall energy performance of a system, with both fuel and electrical expenditures, determined by the ratio of the end energy supply and the total energy consumption of the system. For the systems under consideration, this can be defined as η_{sys}, as follows.

$$\eta_{sys} = \frac{\dot{Q}_{boiler} + \dot{Q}_{hexs} + \dot{Q}_{hp}}{\dot{m}_{fuel} lhv_f + P_{,el}} \quad (5)$$

Here, the end energy supply is primarily part of the network’s total heat demand, which is supplied by the boiler $\dot{Q}_{boiler}$, heat exchangers $\dot{Q}_{hexs}$, and heat pump $\dot{Q}_{hp}$. The energy expenditures are from the fuel energy input $\dot{m}_{fuel} lhv_f$ and the electricity consumption P_{el} of the heat pump.

2.4 Case description

To analyze the influence of certain parameters on the integration of a heat pump, an anonymized biomass heating network in Germany was selected. The furnace’s nominal power and boiler nominal power were 3800 kW and 3000 kW, respectively. The air preheater is installed in the heating plant before the economizer. The fuel used in the heating plant is loose wood chips with a water content of 50%.

It is assumed that the CO and NOx components of the flue gas are 220 mg/m3 (i.N.tr) and 200 mg/m3 (i.N.tr), respectively, in accordance with the limit values of the 44th BImSchV in Germany [10], with a reference O2 content of 11%*vol*. The flue gas temperature directly after the boiler is set to 180 °C. There is a temperature reduction of 20 K after the air preheater and 60 K after the economizer. The flue gas then enters the condenser at a temperature of 60 °C, where it is cooled by the heat pump to the target exhaust gas temperature. The network flow and return are set to 90 °C and 55 °C respectively. The average ambient temperature is 15 °C.

3. RESULTS AND DISCUSSION

This section analyzes the influence of the target exhaust gas temperature, heat pump supply temperature, and choice of refrigerant on heat pump design and overall system efficiency.

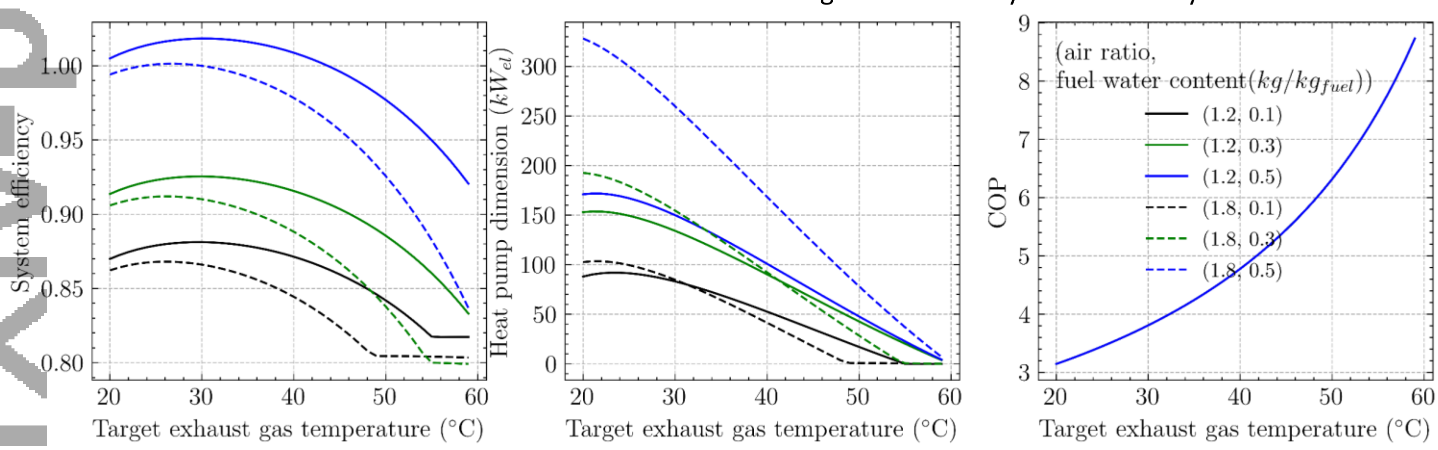

Figure 2 LHV-based system efficiencies, required heat pump dimensions, and COP, as a function of target exhaust gas temperature for various air ratios and fuel water contents (at a constant heat pump supply temperature of 65 °C)

3.1 Analysis of target exhaust gas temperature

Figure 2 shows that the system efficiency based on the lower heating value does not always increase as the exhaust gas temperature falls. At an air ratio of 1.2, the highest efficiency points appear at an exhaust gas temperature of about 30 °C. If the exhaust gas temperature decreases further, the total efficiency will also decrease. This is due to the unavoidable power consumption of the heat pump. At a very low exhaust gas temperature, the potential saving is already so low that the power consumed for heat recovery actually has a negative effect on the system as a whole. The diagram also shows that the maximum achievable total efficiency depends on the air ratio and the fuel water content. At a higher air ratio λ, the maximum efficiency decreases because the higher air volume in turn increases the exhaust gas volume. Some of the heat is dissipated by the increased exhaust volume. By increasing the λ value, the optimum exhaust gas temperature shifts to the left. Thus, the air ratio also needs to be considered when selecting the exhaust gas temperature.

The analysis clearly shows that the maximum efficiency is significantly higher at a higher fuel water content than at a lower one. This is due to the high latent heat of the steam in the flue gas. The greater the quantity of water in the fuel, the more energy can be recovered. Overall efficiency therefore also increases with the increase in water content.

The evaporation temperature of the heat pump depends greatly on the target exhaust gas temperature after the flue gas condenser. A lower exhaust gas temperature means a high recovery potential, but also a higher temperature lift for the heat pump. Therefore, the electricity required for the recovery of waste heat at a lower temperature level also increases as shown in Figure 2. The graph also indicates that the dimensions required for the heat pump increase rapidly as the exhaust gas temperature falls, before the highest efficiency value is reached. This is accompanied by a sharp drop in the COP. Higher power requirements coupled with a lower COP value also mean higher investment costs and poor operating conditions, even though more energy can be recovered from the exhaust gas.

3.2 Analysis of heat pump supply temperature

Once heat has been recovered from the exhaust gas, it can be fed into either the network return or the network supply. The main difference lies in the heat pump heating supply temperature. Different supply temperatures are required, depending on how the heat pump condenser is integrated.

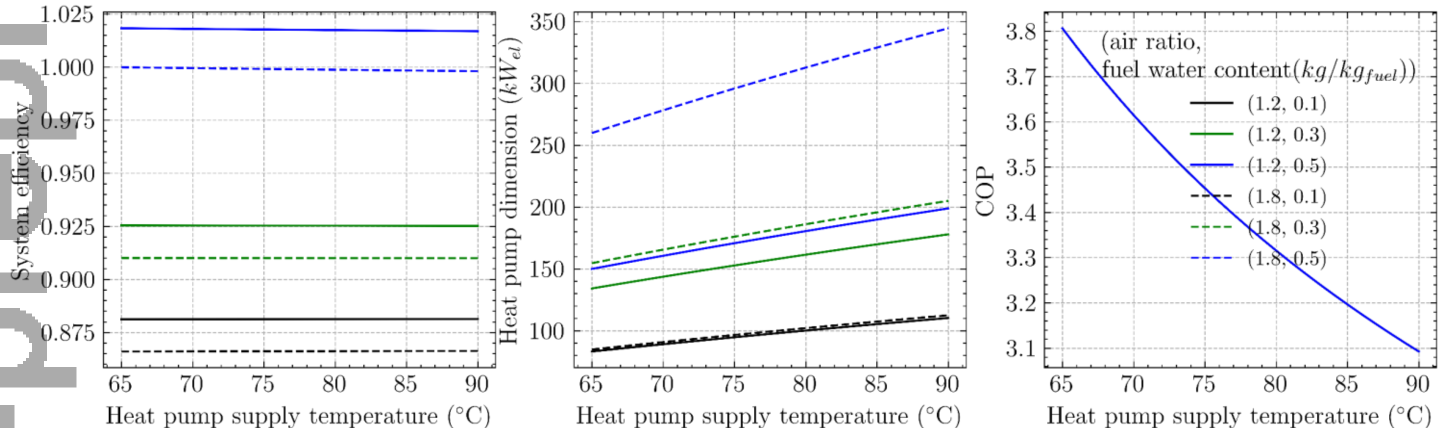

Figure 3 LHV-based system efficiencies, required heat pump dimensions, and COP, as a function of heat pump supply temperature for various air ratios and fuel water contents (at a constant exhaust gas temperature of 30 °C)

Figure 3 presents the system efficiencies, required heat pump dimensions, and COP, at different heat pump supply temperatures. The graphs on the left show that a change in heating temperature has no significant effect on system efficiency. However, as the supply temperature rises, the power requirements of the heat pump increase significantly. The sharp drop in the COP was caused by the increased temperature lift between the evaporator and the condenser. At a fixed exhaust gas temperature, the heat available from the flue gas condenser remains constant. To fully recover the heat at a lower COP value, the required power input increases, as do the dimensions of the heat pump. This trend is more pronounced at higher fuel water contents. From a technical and economic viewpoint, a higher heat pump supply temperature has no significant impact on the overall system but results in larger heat pump dimensions, accompanied by higher investment costs and a more significant electricity input. Therefore, when designing the heat pump, the chosen supply temperature should only be as high as necessary. However, it must also be considered that the heat supply temperature must not fall below the minimum value, so that all the heat can be fed into the grid, due to the pinch rule of heat exchangers [11].

$$\frac{\dot{Q}_{hp}}{\dot{m}_{net}^{main} * c_p} + T_{return}^{main} \leq T_{supply}^{hp} \tag{6}$$

3.3 Analysis of the choice of refrigerant

After a preliminary analysis of their properties, R134a, R717, R236fa, R290, R600a, and R1234yf were selected as the refrigerants to be focused on in this study, as they have suitable boiling temperatures at ambient pressure and critical temperatures.

Figure 4 shows the results of varying the exhaust gas temperatures for different heat pump temperature lifts and refrigerants. It can be seen that the choice of refrigerant has no significant effect on system efficiency. This low influence of the refrigerant on the system is justified, as the resulting heat pump supply capacities are similar.

However, the choice of refrigerant does affect the COP of the heat pump, and, in turn, the electricity demand. Due to the high price and environmental impact of electricity, the choice of refrigerant is also important from an economic and environmental point of view. This analysis enables the right choice of refrigerant to be made for the selected temperature level and allows waste heat to be recovered with the lowest possible power consumption.

Figure 4 reveals the differences in the COP values for both low- and high-temperature increases. R236fa performs well at high target exhaust gas (heat source) temperatures, while R717 performs better at lower temperatures. However, R236fa has the highest GWP value [12] of all the refrigerants considered and can cause environmental damage. R717 is in the safety group denoting increased toxicity (B2L) [13].

As an alternative, R600a behaves well at different exhaust temperatures and temperature increases. The COP values are significantly lower with R290 and R1234yf than with R600a. It is also noticeable that R1234yf does not perform well at high exhaust temperatures with large temperature increases, due to the low critical temperatures of the two refrigerants. Thus, the R600a refrigerant might be a better alternative to R236fa and R717 than R290 or R1234yf. However, even when using R600a, it should be noted that R600a belongs to the "highly flammable"(A3) group 13]. The most commonly used refrigerant, R134a, is not recommended due to its high GWP number and relatively low COP values.

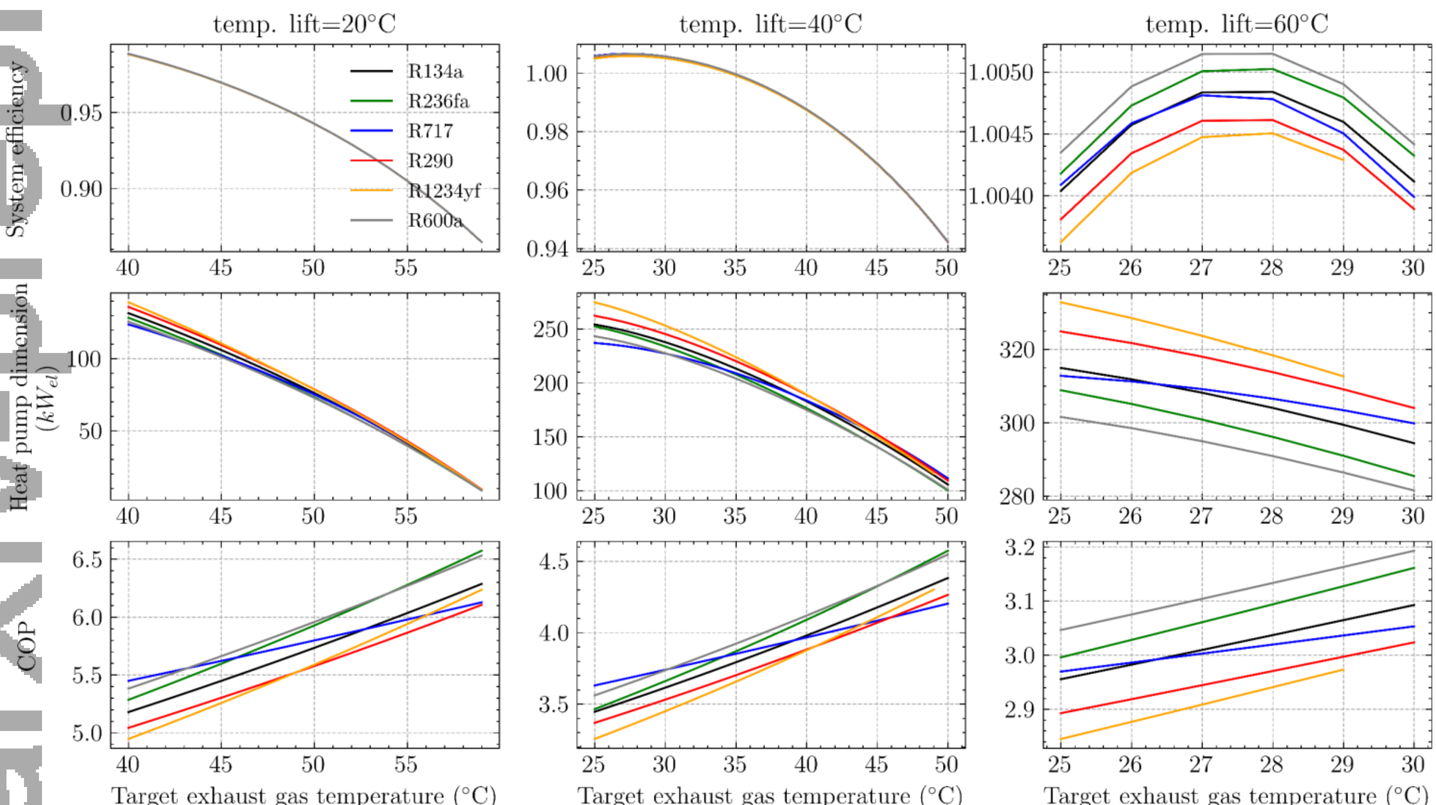

Figure 4 LHV-based system efficiencies, required heat pump dimensions, and COP, as a function of target exhaust gas temperature for various temperature lifts and refrigerants

4. CONCLUSION

This paper presents a thermodynamic analysis of a heat pump design used for heat recovery in biomass heating networks and its impact on the whole system. The most crucial technical parameters due to their significant impact on heat pump design are the target exhaust gas temperature, heat pump supply temperature, and choice of refrigerant.

The analysis was performed under various conditions, enabling several conclusions to be drawn, as follows:

- Due to the dependence of the heat pump evaporation temperature on the target exhaust gas temperature, the overall efficiency of a biomass district heating system with an integrated heat pump does not always increase as the exhaust gas temperature decreases, as it does with standard biomass heating. A maximum point of overall system efficiency is reached at different target exhaust gas temperatures. When selecting the exhaust gas temperature, the fuel water content and air ratio should also be taken into account.
- The supply temperature of the heat pump has little effect on the overall system, but it has a significant effect on the size of the heat pump. A higher flow temperature leads to larger dimensions and higher investment and operating costs. Therefore, when designing a heat pump, the chosen supply temperature should only be as high as necessary.
- The choice of refrigerant has no apparent effect on system efficiency, but it does have an impact on the COP of the heat pump. The analysis reveals that refrigerant R236fa performs best at high target exhaust gas temperatures, while R717 performs better at low temperatures. In contrast, R600a performs well at both temperature levels. Based on its characteristics, it is the one recommended for the case studied here.

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