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High-Temperature Two-Stage Subcritical Heat Pump Running on Environmentally Friendly Refrigerants

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ABSTRACT

The paper presents the design of a two-stage subcritical compressor heat pump operating in an air-to-water system and running on environmentally friendly refrigerants. The pump is dedicated to buildings where there is a high-temperature central heating system and wall-mounted radiators are used as heat receivers. The first stage of the unit was supplied with R290 refrigerant and the second stage with R1234 ze(E) refrigerant. The nominal heating capacity of the unit is 10 kW for a lower source temperature in the range -20÷10 °C. The unit›s refrigeration system was designed and simulation tests were carried out to determine the unit›s operating parameters for the lower source temperature in the range -20÷10 °C, the condensation temperature of the stage I refrigerant for values: 20 °C, 25 °C, 30 °C and the condensation temperature of the stage II refrigerant in the range 30÷60 °C. The value of the coefficient of performance (COP) was determined in relation to the heating capacity generated in stages I and II to the electricity consumed, which value, depending on the operating parameters, ranges from 2.55 to 5.6. The value of the COP related to the heating capacity received from the stage II condenser to the total capacity consumed by the stage I and II compressor, depending on the operating parameters, ranges from 1.5 to 2.9. It was found that by using a stepless adjustment of the volumetric efficiency of the stage I and II compressor, it is possible to match the heating capacity of stage I to the cooling capacity of stage II and thus increase the COP value by 50%.

Keywords: two-stage heat pump, COP, R290, R1234ze(E).

INTRODUCTION

Since 1st January 2021, technical conditions have been in force according to which, in the case of newly constructed single-family buildings, the overall energy demand may not be covered by non-renewable energy carriers in an amount greater than 70 kWh/m^{2}year, and, in the case of multi-family buildings, in an amount greater than 65 kWh/m^{2}year of the usable area of the building [1]. In practice, it is not possible to comply with TC2021 in buildings in which the central heating (CH) system and domestic hot water (DHW) preparation system were made as monovalent and based on heating devices supplied by non-renewable energy carriers, such as natural gas, heating oil or coal. In order to meet TC2021, it is necessary to build hybrid systems based on renewable or renewable and fossil energy carriers [2]. In practice, building hybrid supply systems compared to monovalent systems involves significant investment costs. The analyses carried out show that the smaller the usable area of the building, the more difficult it is to meet TC2021. The analyses also show that any building as a monovalent system will meet TC2021 if the CH and DHW system is based on a pellet boiler [2]. The main objective of the introduced TC2021 is to reduce CO_2 emissions to the target neutral level of the EU Member States in 2050 [3]. One of the most preferred solutions for supplying CH and DHW systems in newly constructed buildings in recent years has been a hybrid supply system based on the cooperation of an air-to-water compressor heat pump with a photovoltaic installation [4]. Since the EU energy policy targets not only decarbonisation but also the efficient use of electricity [3], in Poland this solution was subsidised by numerous aid programmes e.g.: Clean Air, My Electricity. Therefore, in recent years, this solution has been dedicated not only to newly constructed buildings but also to existing buildings where thermal modernisation has been carried out and an off-grade heat source has been replaced with a compressor heat pump. Very often, in such cases, a single-stage compressor heat pump does not meet the established expectations, which in practice involves high operating costs due to the compressor's electricity consumption and results in an increase in CO_2 emissions instead of a decrease.

REFRIGERATION SYSTEM PARAMETERS

The first constraint on the use of a compressor heat pump operating in an air-water system with economic justification is the design supply temperature of the heating circuits. This is due to the design of the single-stage refrigeration system and the technical capabilities of the compressors [5, 6]. In the case of single-stage refrigeration units, which include compressor heat pumps, the lowest electricity consumption occurs when the difference between the temperature of the lower source and the temperature of the upper source is the smallest [7]. In practice, this means that due to the low values of the upper source temperature (supply temperature of the heating circuits), the dedicated heat receivers in the CH system for compressor heat pumps are surface radiators (floor radiators), where the maximum temperature of the medium supplying the floor radiator should not exceed 40 °C [8]. Typically, in buildings constructed before the year 2000, and due to the technical parameters of the heating equipment, the central heating (CH) system was designed as a high-temperature system with wall-mounted radiators as the heat receivers. In accordance with PN-EN 442-2 [9], the heating capacity of a wall-mounted radiator is determined for the following operating parameters of a CH system: supply temperature of the heating circuits 75 °C, return temperature of the medium from the heating circuits 65 °C at a room air temperature of 20 °C. Such a system is not suitable for operation with a single-stage subcritical compressor heat pump. To adapt the system, it is necessary to reduce the supply temperature of the heating circuits to a maximum medium temperature of 40 °C. There is an attempt to solve the problem of reducing the supply temperature of the heating circuits, while retaining the existing heat transfer surface of the wall-mounted radiators, by thermo-modernisation of the building involving a reduction of the heat load through decreasing the heat transfer coefficients of building envelopes [10]. As a result, the heat load of the building due to central heating will be reduced; however, is this sufficient to make the use of a single-stage compressor subcritical heat pump, with a maximum heating medium temperature of 40 °C and the existing heat transfer surface of the wall-mounted radiators, economically justifiable in such cases? It is easy to answer this question. The heat transfer surface of a wall-mounted radiator is related to its capacity and the heat transfer coefficient of the material from which it is made according to equation 1 [10].

$$A = \frac{Q_g}{U \cdot \Delta t_g \cdot \varepsilon} \tag{1}$$

where: A – area [m^2], Q_g – heating capacity [W], U – heat transfer coefficient [W/m^{2}K], Δt_g – arithmetic mean temperature difference [K], ε – correction factor [-]

The heating capacity Q_g is the capacity of the radiator, the value of which is given catalogued by manufacturers in accordance with PN-EN 442-2:2015-02 [7]. The first temperature value is the supply temperature of the heating circuits, the second the return temperature and the third the air temperature in the room. The catalogues also give the remaining coefficients of equation 1. For example, let's assume that we are heating a room with an area of 20 m^2 in a building before thermo-modernisation, whose specific heat load is 70 W/m^2. It is easy to calculate that the heating capacity of the wall-mounted radiator must be 1400 W, at a supply temperature of 75 °C. For the assumed operating parameters, the calculated heat transfer surface of the radiator is 0.48 m^2. The building underwent thermo-modernisation and the heat load of the building was reduced to 25 W/m^2, so the radiator capacity was reduced to 500 W. This means, as one can easily calculate from Equation 1, that with 500 W of nominal heating capacity and a retained heat transfer surface of the wall-mounted radiator of 0.48 m^2, the supply temperature of the heating circuits must be 51.5 °C. Such operating parameters of the system in reality lead to high electricity bills. The second constraint also affecting the increase in electricity bills is due to the operating envelope of the compressors. Although in the catalogue the compressor operating envelopes contain a very wide range of refrigerant saturation and condensation temperatures, in practice the difference between the lower and upper source temperatures, depending on the compressor type, will be no more than 50 K. At these operating parameters, the operation of the refrigeration system is assisted by an electric heater consuming electricity at a ratio of 1:1 to the heat generated (Figure 1) [12]. In addition, with a constant volumetric efficiency of the compressor and an increase in the upper source temperature, the capacity of the heat pump will decrease and the electricity consumption required to drive the compressor will increase (Figure 2). Figures 3 and 4 show the operating parameters of the heat pump, which were recorded during operational tests on a test bench held by the Institute of Mechanical Engineering, the design and operating principle of which are presented in [13].

As can be seen from the operating parameters of the heat pump, the higher the temperature of the upper source at a constant temperature of the lower source, the lower the heating capacity of the unit and the higher the electricity consumption required to drive the compressor, with a simultaneous decrease in the COP value. The COP value for the operating parameters: upper source 50 °C, lower source -4 °C reached 1.74.

A solution to the presented constraints and disadvantages of single-stage heat pumps could be the use of a high-temperature subcritical cascade compressor heat pump operating in an air-to-water system [14–16]. Such a solution would make it possible to cover the heat load of buildings with high-temperature central heating systems without having to modernise them. Cascade compressor systems are known and described in the literature [17–19]. Numerous studies have focused mainly on the selection of first- and second-stage refrigerants to ensure the highest possible

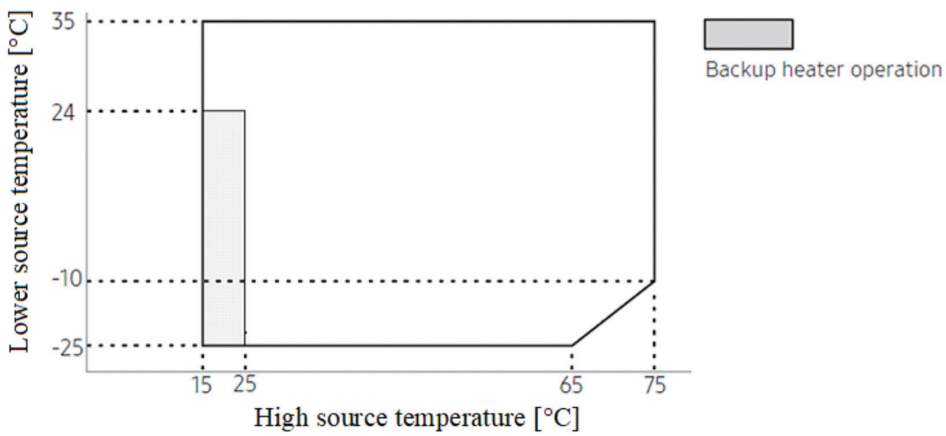

Figure 1. Example operating envelope of a compressor heat pump operating in an air-water system

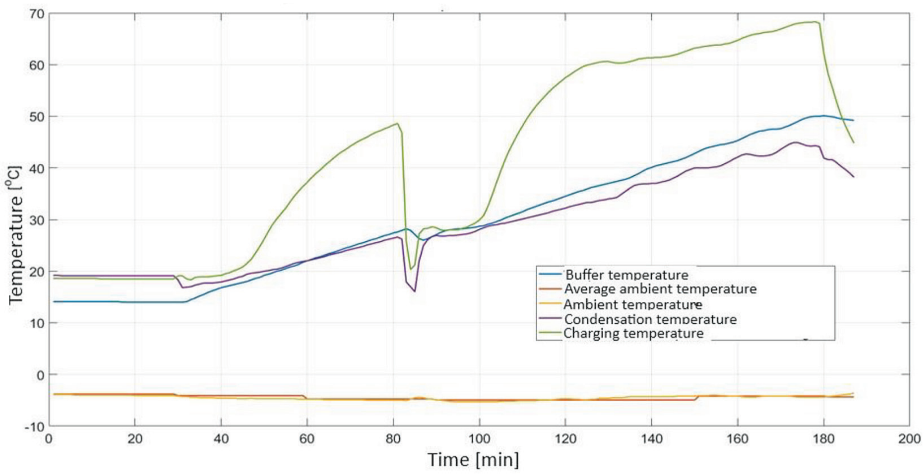

Figure 2. Lower and upper source temperatures during operational tests

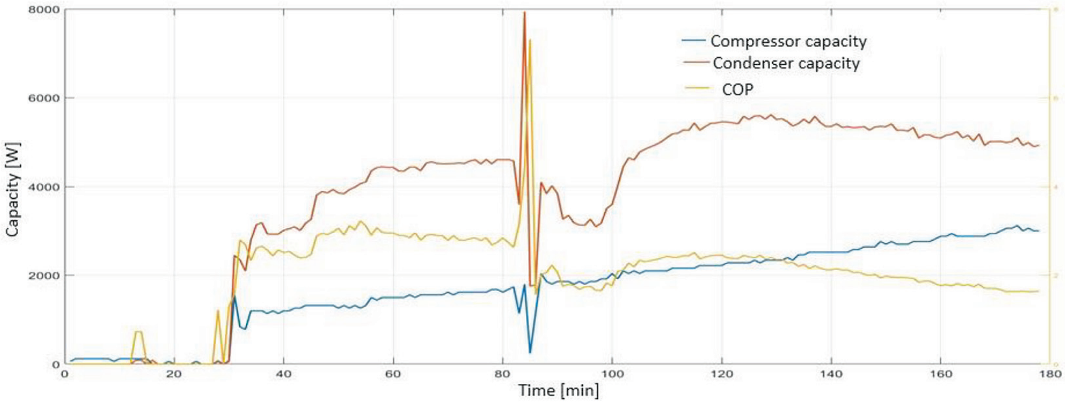

Figure 3. Heating capacity, cooling capacity and COP during the experiment

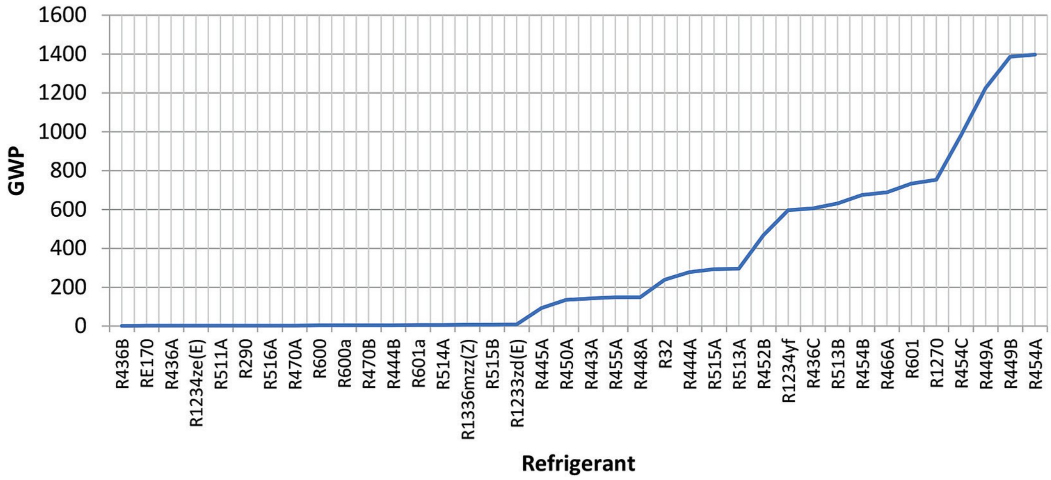

Figure 4. Ranking of refrigerants according to GWP

unit efficiency. Performance tests have been carried out on cascade heat pumps supplied with: R600 in the lower stage and R245fa in the upper stage [20], R404a in the lower stage and R134a in the upper stage [21–23]. The refrigerants analysed, with the exception of R600, are synthetic refrigerants with high GWP values: R245fa (GWP = 1030), R404a (GWP = 3922), R134a (GWP = 1430). According to the phase-out schedule for fluorinated refrigerants in Regulation (EU) No 517/2014 of the European Parliament and of the Council [24], commonly used synthetic refrigerants must be replaced by refrigerants characterised by DOP = 0 and GWP < 149. This raises a problem with selecting the refrigerants used in the first and second stage of the cascade. In addition, the use of new refrigerants involves the need to develop the design of the unit's refrigeration system, e.g. the choice of heat transfer surfaces of exchangers, pipe diameters and components.

HIGH-TEMPERATURE CASCADE HEAT PUMP

The main design consideration for the cascade heat pump was the unit's nominal heating capacity of 10 kW for a lower source temperature in the range of -20÷10 °C and an upper source temperature in the range of 30÷60 °C. The key design problem was selecting the heat transfer surfaces of the exchangers – the evaporator in stage I, the condenser in stage II of the unit and the exchanger between stages, which acts as a condenser for stage I and as an evaporator for stage II. The second problem was designing the refrigeration system so that evaporator defrosting could be carried out using the hot gas method, with the aim of minimising defrosting time and thus reducing the electricity consumed in the process. The third design problem was selecting the refrigerants that could be used in stages I and II of the unit within the assumed boundary conditions. As already mentioned, pursuant to EU Directive 517 [24], synthetic refrigerants are being phased out and must be replaced with environmentally friendly refrigerants. An environmentally friendly refrigerant is characterised by ODP = 0 and GWP < 149. Therefore, the available refrigerants were reviewed in terms of four criteria: value of ODP = 0, value of GWP < 7, refrigerant pressure as a function of saturation temperature > 2 bar, and refrigerant pressure as a function of critical point temperature < 25 bar.

Refrigerant selection

Figure 4 shows the technical parameters of the refrigerants that can be used in the system. All the selected refrigerants have an ODP value of zero. The GWP values of the selected refrigerants range from 1 to 1397. According to Directive 517/2014 [24], refrigerants with GWP values in the range 150–1500 can be used in the first stage. In the upper stage of the unit, only refrigerants with GWP < 150 will be able to be used, which means that the number of refrigerants that can be used is reduced to 21 (Fig. 4). When selecting the refrigerants used in the lower as well as the upper stage, their saturation and condensation temperatures as a function of pressure are important. The saturation temperature of the first stage refrigerant must be adequate for the possible variations in the temperature of the lower source, and the condensation temperature of the second stage refrigerant must be adequate for the required operating temperatures of the upper source. The condensation temperature of the first stage refrigerant and the saturation temperature of the second stage refrigerant must be coherent with respect to each other. Figure 5 shows a summary of the selected refrigerants as a function of the critical point temperature. For technological reasons, due to the lower critical point temperature required, a refrigerant with a critical point temperature of less than 80 °C can be used in the lower stage. For this reason, agents in the range R6001 – R449B can be used in the lower stage. In the second stage, due to the higher condensation temperature required, agents in the range R448A – R290, whose critical point temperature is lower than 100 °C, should be used (Fig. 5). Based on the pressure for a saturation temperature of 0 °C, the group of refrigerants that can be used in the first stage narrows considerably (Fig. 6), as the pressure of the refrigerant cannot be close to atmospheric pressure. For this reason, refrigerants in the range R1233zd(E) – R170 have been eliminated. The same is true of refrigerants that can be used in the second stage, for which the critical point temperature as a function of pressure is relevant (Fig. 7). Refrigerants whose critical point pressure exceeds 40 bar (R454 C – R170) have been eliminated. Based on the analyses carried out with regard to the parameters of the refrigerants in terms of the required operating parameters (temperature of the lower source -20 °C –/+ 15 °C, temperature of the upper source 30–60 °C), it was decided that the lower stage will be supplied with R290, whose ODP is 0 and GWP 3, and the upper stage will be supplied with R1234ze(E), whose ODP is O and GWP 7. The other operating parameters of the selected refrigerants are shown in the pressure – enthalpy diagrams (Figs. 8, 9).

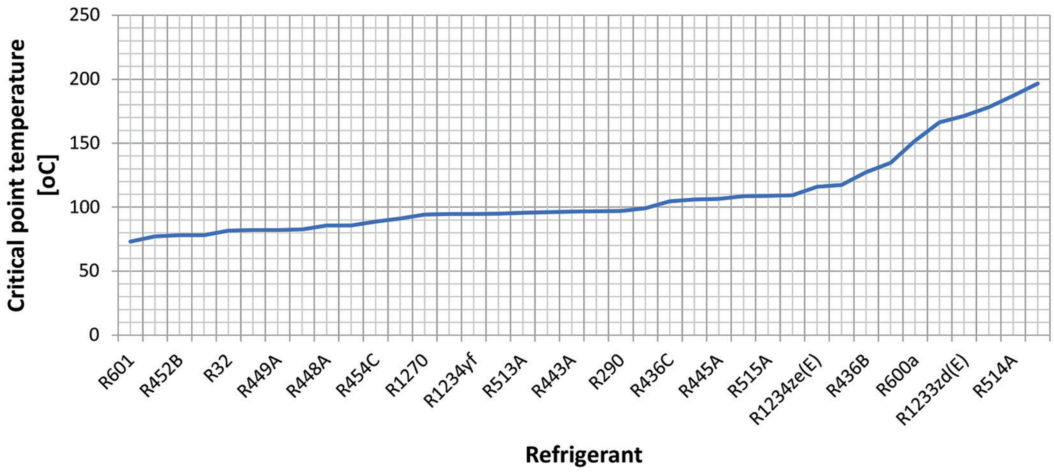

Figure 5. Ranking of refrigerants according to critical point temperature

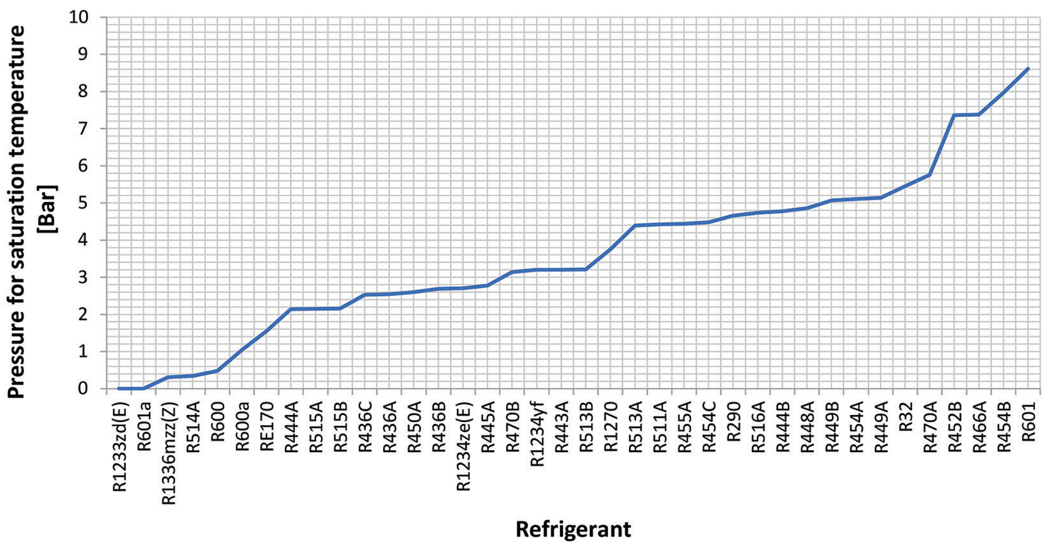

Figure 6. Ranking of refrigerants according to pressure for saturation temperature of 0 °C

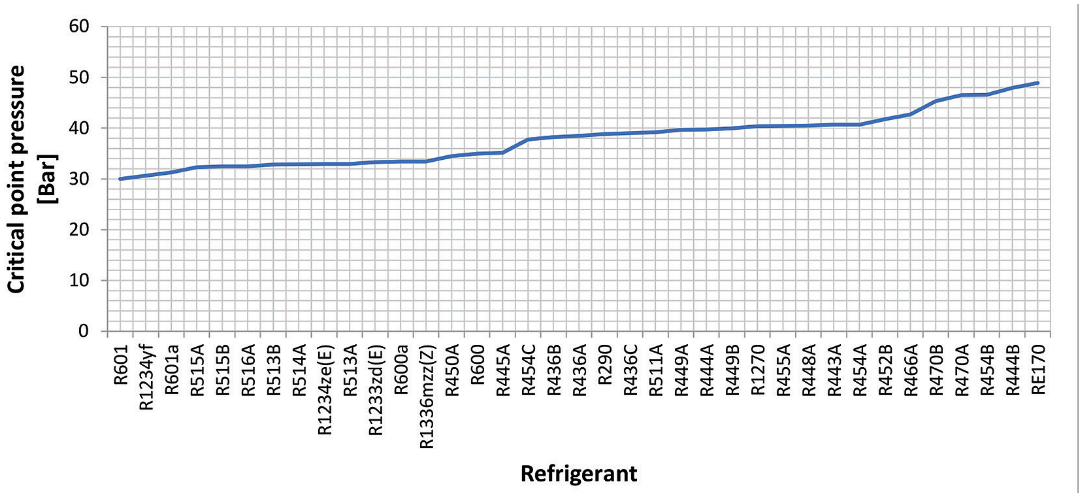

Figure 7. Ranking of refrigerants according to critical point pressure

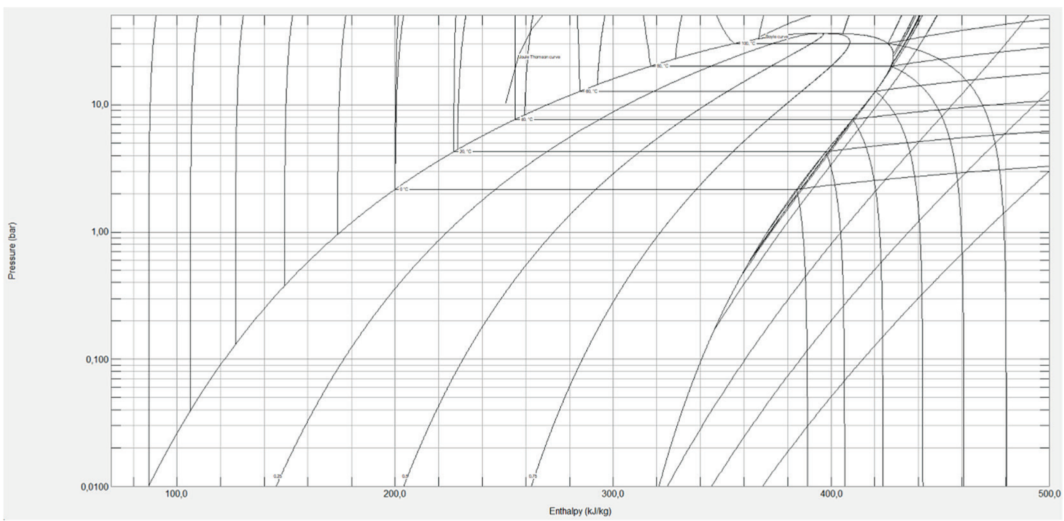

Figure 8. p-h diagram of the R1234ze(E) refrigerant

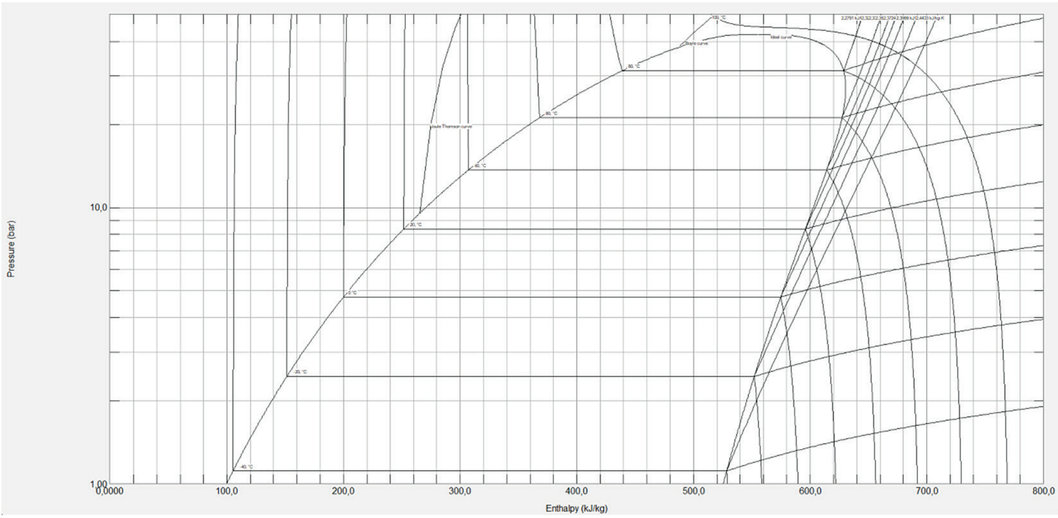

Figure 9. p-h diagram of the R290 refrigerant

DESIGN OF THE REFRIGERATION SYSTEM OF A TWO-STAGE HIGH-TEMPERATURE COMPRESSOR HEAT PUMP AND BOUNDARY OPERATING PARAMETERS

The design of the two-stage subcritical compressor heat pump is based on two refrigeration systems separated by a heat exchanger acting as condenser for the lower stage and evaporator for the upper stage. A schematic diagram of the unit's refrigeration system is shown in Figure 10. A finned (lamellar) heat exchanger (21) with fan-forced airflow is used as the evaporator in the first stage of the unit. The exchanger (22) is the condenser for the first stage and the evaporator for the second stage. The exchanger (23) acts as the condenser for the second stage. The key design problem was selecting the heat transfer surface area of the individual exchangers so that, over the entire range of variation in the temperature of the lower source at the assumed range of refrigerant condensation, the refrigerant supplying stage I as well as stage II in the evaporator changes its physical state from liquid to superheated steam, and in the condenser from superheated steam to liquid. Both stages use scroll compressors (1), which have been thermally secured by mounting a temperature sensor (22) on the head. Both stages use regenerative exchangers (15) that allow superheating and subcooling of the refrigerant and act as a liquid refrigerant reservoir. The refrigeration system is pressure-protected by low (12) and high (9) pressure switches and a safety valve (16) connecting the high-pressure liquid line to the suction line. A four-way valve is fitted in both stage I and II to enable reversible operation of the unit. In addition, in both the first and second stages there are auxiliary fittings in the form of sight glasses (11), filters (14), expansion valves (7), service valves (10) as well as liquid refrigerant reservoirs (16).

The following boundary conditions were assumed when designing the unit: nominal unit heating capacity of 10 kW over the entire lower source temperature range, lower source temperature range of the first stage: -20÷10 °C, first-stage refrigerant condensation temperature values of 20 °C, 25 °C and 30 °C, second-stage refrigerant condensation temperature range of 30÷60 °C. In addition, it was assumed that the saturation temperature of the first-stage refrigerant as well as the second-stage refrigerant is 10 K lower than the source temperature. With the p–h diagrams of the refrigerants linking the key refrigerant parameters (Figs. 8, 9), including: enthalpy, pressure, temperature, entropy and specific volume of the steam, and using equation 1 for the assumed boundary conditions as well as based on the parameters of the R1234ze(e) refrigerant shown in the p-h diagram (Fig. 8) and based on the R290 refrigerant parameters shown in the p-h diagram (Fig. 9), the refrigerant mass flow rate was calculated from equation (2), allowing the volumetric efficiency of the compressors to be calculated from equation 3. The volumetric efficiencies of the compressors are 26.84 m^3/h and 13.42 m^3/h respectively.

$$m = \frac{Q_g}{(h_2 - h_3)} \tag{2}$$

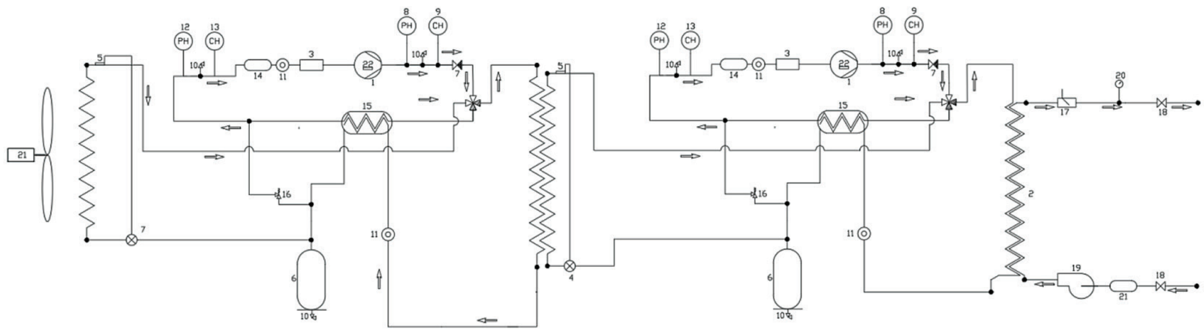

Figure 10. Diagram of the refrigeration system of a two-stage high-temperature subcritical heat pump.

where: m – mass flow of the refrigerant [kg/s], Q_g – heating capacity [W], h_2 – enthalpy of refrigerant discharge [J/kg], h_3 – enthalpy of refrigerant condensation [J/kg]

$$V = m \cdot v \tag{3}$$

where: V – compressor volumetric efficiency [m^3/s], m – mass flow of the refrigerant [kg/s], v – specific volume of the refrigerant at the suction connection [m^3/kg]

Having obtained the volumetric efficiency of the two compressors, simulation tests were carried out to determine the operating parameters of the designed unit.

Operating parameters of a two-stage high-temperature compressor heat pump

Simulation tests were performed in the Matlab&Simulink package. Using available libraries, a thermodynamic system was constructed for the designed unit. Pressures, temperatures and the physical state of the refrigerant were measured at key points in the refrigeration system. During the simulation tests, it was assumed that the difference between the inlet and outlet temperatures of air flowing through the evaporator would be 1K, while the difference between the outlet and inlet temperatures of water entering the second stage condenser would be 5 K. Simulation tests were performed assuming that the compressors operate in ON/OFF mode. The performance of the fan and the circulation pump was adjusted using PI controllers. A fragment of the thermodynamic system model of the two-stage high-temperature heat pump implemented in the Matlab and Simulink software is shown in Figure 11.

Figures 12–20 show the operating parameters of the designed high-temperature subcritical heat pump for the assumed boundary conditions. Due to the amount of measurement data, in this article the authors have presented the results they considered crucial. Figure 12 shows the variation in the capacity of stage I evaporator as a function of the lower source temperature with stage II refrigerant condensation in the range 30÷60 °C. It can be seen from the graph that for each of the cases considered, the minimum value of the cooling capacity of the first stage of the unit is for a lower source temperature of -20 °C. As the lower source temperature increases, the cooling capacity increases to reach a maximum for -10 °C. For higher temperatures, the cooling capacity decreases. An analogous trend should be observed in the case of variation in the capacity of the stage II evaporator, which also serves as the stage I condenser (Fig. 13). The trend of variation in the cooling capacity of stages I and II, as shown in Figs. 12 and 13, is the result of two factors: the constant volumetric efficiency of both compressors and the changes in the heating capacity of stage I of the unit, which depends on the difference between the lower and upper source temperatures. The smaller the difference between the lower and upper source temperatures, the greater the heating capacity. This means that as the temperature of the unit's lower source increases, the heating capacity of the stage I condenser – which is also the stage II evaporator – increases, resulting in an increase in refrigerant superheating. For the analysed lower source temperature of stage II, which is the condensation temperature for the stage I refrigerant of 20 °C, the cooling capacity demand of stage II is 9.66 kW. For a stage I operating point of -20 °C/20 °C, the available heating capacity of stage I is 5.56 kW. As a result, the expansion valve in stage I of the unit is fully open, while the expansion valve in stage II of the unit must limit the amount of refrigerant entering the evaporator. As a result, the heating capacity

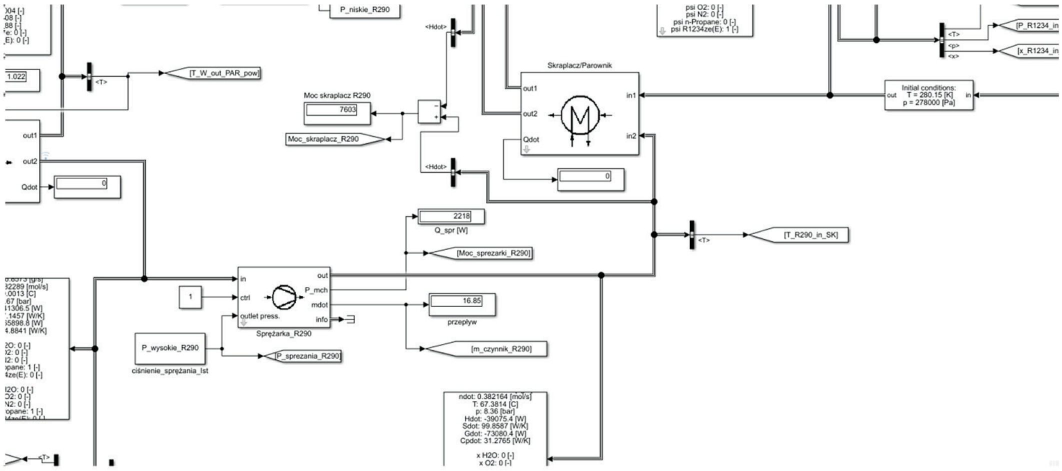

Figure 11. Thermodynamic system model fragment implemented in the Matlab and Simulink software

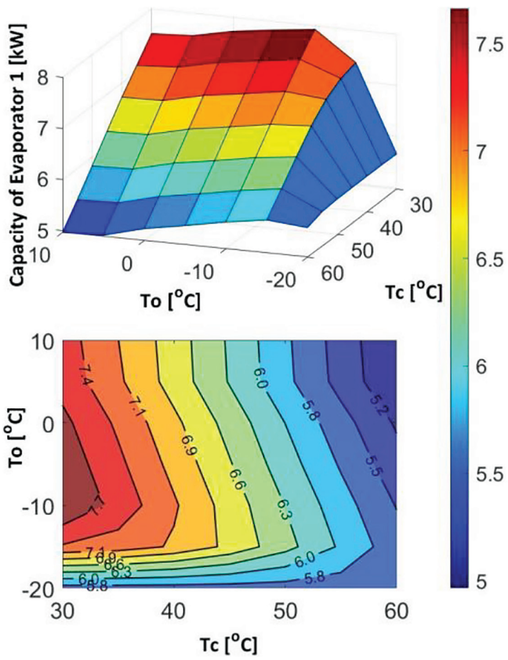

Figure 12. Capacity of stage I evaporator as a function of the lower source temperature – To with stage II refrigerant condensation temperature – Tc in the range 30÷60 °C

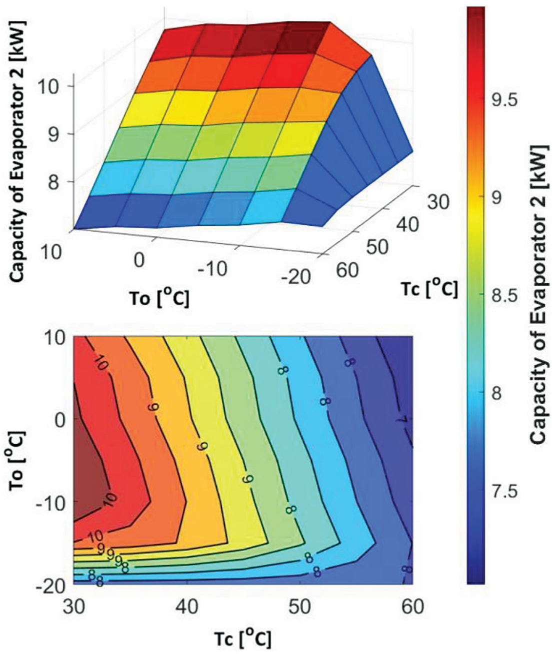

Figure 13. Capacity of stage II evaporator as a function of the lower source temperature – To with stage I refrigerant condensation of 20 °C and stage II refrigerant condensation temperature – Tc in the range of 30÷60 °C

available on the stage II condenser is lower than the assumed design capacity u, depending on the condensation temperature of the refrigerant it ranges from 6.3 kW for a condensation temperature of 30 °C to 9.96 for a temperature of 60 °C (Fig. 14). In other cases, the heating capacity received at the stage II condenser is at the assumed level or higher. To increase the capacity of the unit's first-stage condenser, a compressor in stage I with a higher volumetric efficiency should be used, or a stepless adjustment of the volumetric efficiency of both stage I and stage II compressors should be applied. Then, by adjusting the volumetric efficiency of the stage I and II compressors, it would be possible to match the required inter-stage heating and cooling capacity. An increase in the condensation temperature of the stage I refrigerant to a value of 30 °C translates into an increase in the saturation temperature of the stage II refrigerant and does not result in a drastic increase in the cooling capacity of stage II (Fig. 15). The cooling capacity of stage II increases

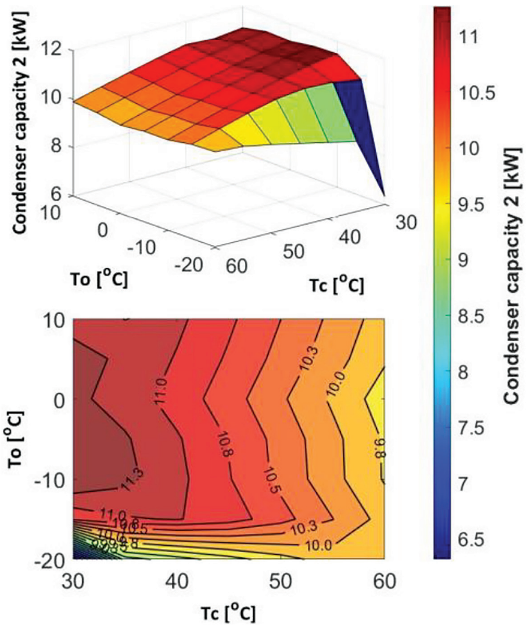

Figure 14. Capacity of stage II condenser as a function of the lower source temperature – to with stage I refrigerant condensation temperature of 20 °C and stage II refrigerant condensation temperature – Tc in the range of 30 ÷ 60 °C

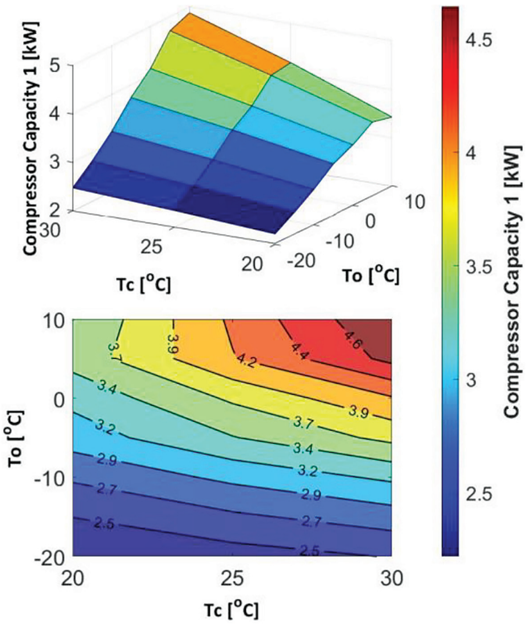

Figure 16. Capacity of stage I compressor as a function of the lower source temperature – To with stage I refrigerant condensation temperature – Tc of 20 °C, 25 °C, 30 °C

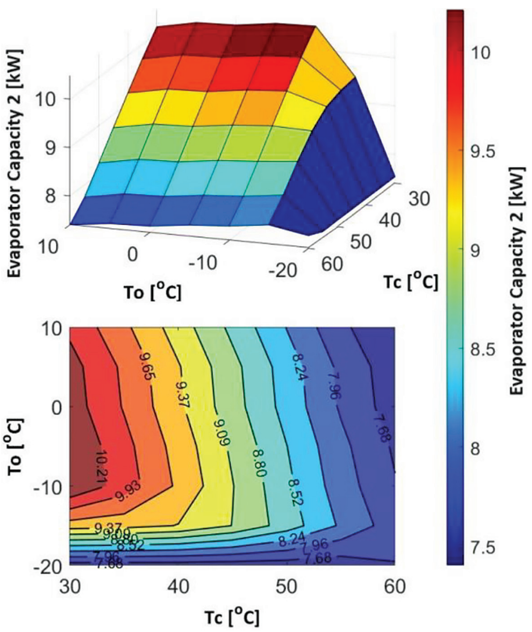

Figure 15. Capacity of stage II evaporator as a function of the lower source temperature – To with stage I refrigerant condensation temperature of 30 °C and stage II refrigerant condensation temperature – Tc in the range of 30 ÷ 60 °C

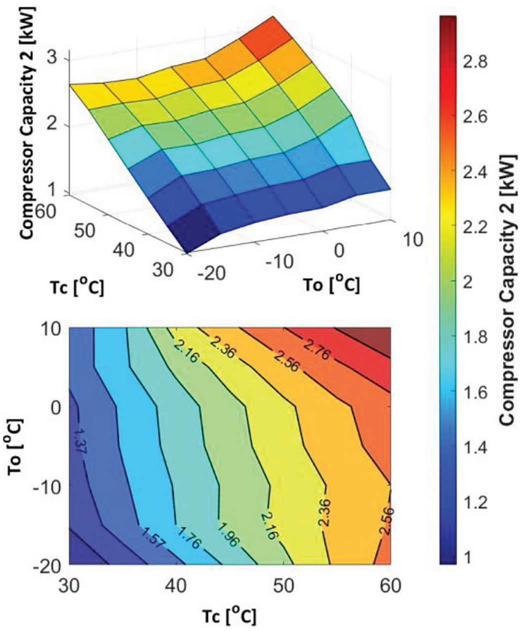

Figure 17. Capacity of stage II compressor as a function of the lower source temperature – To with stage I refrigerant condensation temperature of 20 °C and stage II refrigerant condensation temperature – Tc in the range of 30 ÷ 60 °C

from 9.96 kW to 10.31 kW. Figure 16 shows the capacity of the stage I compressor as a function of the lower source temperature in the range -20÷10 °C at refrigerant condensation temperatures of 20 °C, 25 °C and 30 °C. For the lowest lower source temperature of -20 °C, regardless of the value of the condensation temperature, the compressor capacity is the lowest. This is due to the fact that the heating and cooling capacity is the lowest (Figs. 12, 13). An increase in the temperature of the lower source translates into an increase in the heating capacity available at the stage I condenser, hence increasing the stage I compressor capacity (Fig. 16). Naturally, the higher the refrigerant condensation temperature, the higher the compressor capacity, resulting in an increase in electricity consumption. A change in the temperature of the lower source does not significantly affect the capacity of the stage II compressor at a constant refrigerant condensation temperature (Fig. 17). The capacity of the stage II compressor is significantly influenced by the condensation temperature value of the stage II refrigerant. The higher the condensation temperature of the stage II refrigerant, the more the compressor capacity increases, reaching a maximum value of 3.15 kW for a condensation temperature of 60 °C with a lower source temperature of 10 °C (Fig. 17).

A reduction in electricity consumption is possible by achieving a smooth volumetric efficiency of stage I and stage II compressors, which will enable the heating capacity of stage II to be matched to the assumed value of 10 kW – for most lower and upper source temperatures, the capacity of stage II is greater than 10 kW (Fig. 14). Failure to match the heating capacity of stage I with the cooling capacity demand of stage II significantly affects the COP value of the unit. As the lower source temperature increases, the heating capacity available at the stage I condenser increases, which is not fully utilised due to the constant value of the stage II cooling capacity at the constant condensation temperature of the stage II refrigerant (Fig. 18). The COP value is much less influenced by the volumetric efficiency adjustment of the stage II compressor. In the case of stage II, the decisive factor influencing the COP value is the value of the condensation temperature of the stage II refrigerant (Fig. 19). Analysing the COP value in relation to the heating capacity generated in stages I and II to the electricity consumed by stages I and II shows that the higher the refrigerant condensation temperature in stage II at a constant lower source temperature, the more the COP value decreases (Fig. 20). The increase in the COP value mainly depends on the volume efficiency adjustment of the stage I compressor. Due to the significantly higher available capacity of the stage I condenser than the cooling capacity demand of stage II, the COP value decreases when the condensation temperature of the stage II refrigerant is constant and the lower source temperature increases (Fig. 20). From Figure 20, it can be estimated that the use of stepless adjustment of the volumetric efficiency of the stage I compressor would enable to increase the COP value by at least 50% while maintaining the unit's

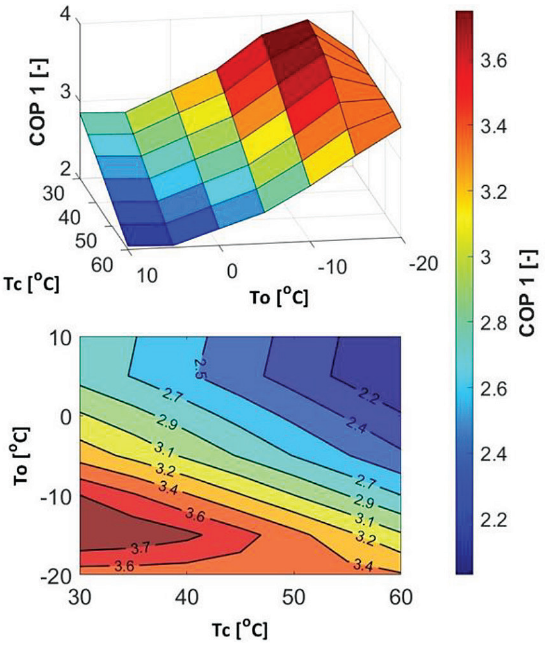

Figure 18. Stage I COP as a function of the lower source temperature – to with stage II refrigerant condensation temperature – Tc in the range 30÷60°

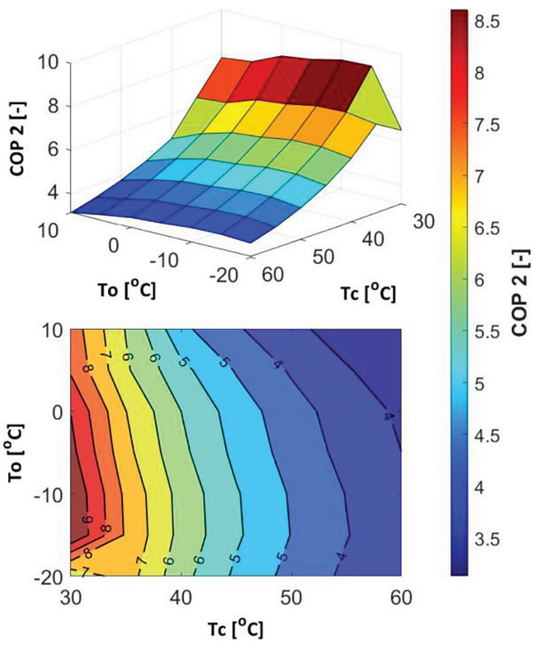

Figure 19. Stage I COP as a function of the lower source temperature – To with stage II refrigerant condensation temperature – Tc in the range 30÷60°

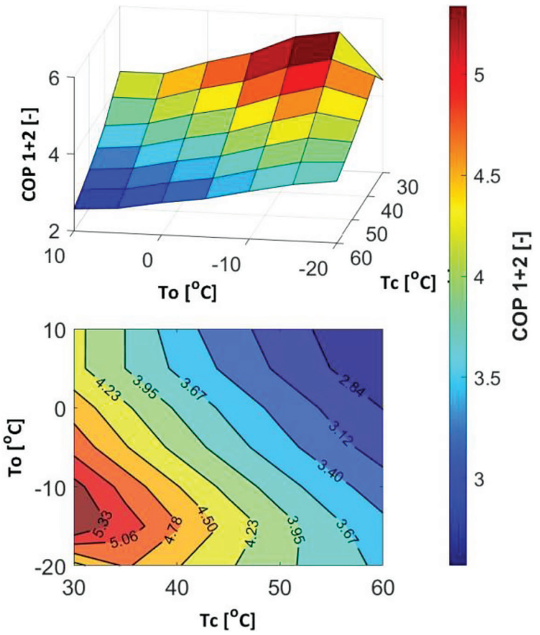

Figure 20. COP in relation to the heating capacity generated in stages I and II to the electricity consumed by stages I and II COP as a function of the lower source temperature – To with stage II refrigerant condensation temperature – Tc in the range 30÷60

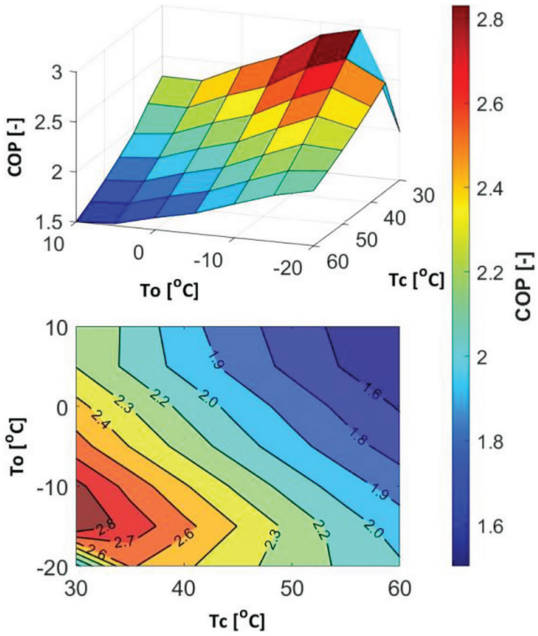

Figure 21. COP in relation to the heating capacity generated in stage II to the electricity consumed by stages I and II COP as a function of the lower source temperature – To with stage II refrigerant condensation temperature – Tc in the range 30÷60°

design heating capacity of 10kW. As a result, the COP value related to the heating capacity received from the stage II condenser to the total capacity consumed by the stage I and II compressor would also be improved by at least 50% (Fig. 21).

CONCLUSIONS

Based on the simulation tests carried out and the analysis of the results obtained, it must be concluded that the designed two-stage subcritical compressor heat pump can constitute an environmentally friendly heat source for single-family buildings intended for thermo-modernisation, in which high-temperature wall-mounted radiators are used as heat receivers. The designed unit provides the assumed nominal heating capacity of 10 kW over the entire range of lower and upper source temperature variation. Reduction in the electricity consumption of stage I and stage II compressors can be achieved by the stepless adjustment of the volumetric efficiency of both compressors. This solution will make it possible to match the heating capacity of stage I to the cooling capacity demand of stage II. In addition, a reduction in electricity consumption can be achieved using adaptive control which adjusts the condensation temperature of stage I and stage II refrigerants to the current heat load of the building as a function of the temperature of the lower source, which is atmospheric air.

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