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Analysis of Efficiency of Finned Heat Exchanger Fed with Synthetic and Natural Refrigerant

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ABSTRACT

This paper presents a comparative analysis of a finned heat exchanger fed with synthetic refrigerant R410a and environmentally friendly refrigerant R290. Analyses of the performance of the heat exchanger operating as an evaporator in a refrigeration system were carried out. Simulation studies were performed using SOLIDWORKS software with the Flow Simulation library. The analysis was conducted for various design parameters of the device. The performance of the exchanger was examined for different fin materials (aluminium alloys: alloy 1060, 1060-H12, alloy 2014, alloy 2024), number of fins (78 and 39), and fin spacings (2.5 mm and 5 mm). The studies were conducted under various operational parameters of the exchanger (varying volume flow of air and refrigerant mass flow). Based on the results obtained, it was found that the material of the exchanger does not significantly affect its performance. The next conclusion is that when the heat exchanger is fed with an environmentally friendly refrigerant, the heat transfer surface area can be reduced by 50%. As a result, the reduction in heat transfer surface area can be achieved by increasing the fin spacing, which will reduce the electricity consumed in the defrost process under operating conditions, or by reducing the size of the exchanger, which will positively translate into a reduction in the weight of the refrigerant in the system. The possibility of reducing the dimensions of the heat exchanger by up to 50% indicates that replacing R410a refrigerant with environmentally friendly R290 is a promising direction for development. It advocates for ongoing investigations to refine these systems and maximize the operational advantages presented by natural refrigerants.

Keywords: evaporator, finned heat exchanger, R290, R410a, heat pump.

INTRODUCTION

In accordance with Regulation (EU) No 2024/573 of the European Parliament and of the Council on fluorinated greenhouse gases, due to their high ODP and GWP values, the HCFC and HFC refrigerants commonly used in appliances must be removed in accordance with the current schedule. According to the legislation, possible replacements for the withdrawn agents are natural agents belonging to the HC group and characterised by low (close to 0) ODP (Ozone Depletion Potential) and GWP (Global Warming Potential) values. The substitution of synthetic refrigerants by environmentally friendly refrigerants is associated with a positive environmental impact; however, it is also associated with a number of design and operational problems for refrigeration systems and equipment. The main disadvantage of green refrigerants is their flammability, so the main limitation is the filling level of the refrigeration system with the green refrigerant (the mass of refrigerant in the system). Ensuring the same refrigerant capacity with reduced refrigerant mass in the system requires solving a number of design problems of key system components, e.g. condenser, evaporator, as well as the refrigeration system itself, e.g. selecting pipe diameters in the refrigeration system, which is a challenge for designers of refrigeration equipment. Therefore, the aim of this study is to compare the cooling capacity of a finned heat exchanger operating as an evaporator in a refrigeration system, supplied with synthetic refrigerant R410a (GWP – 2088, ODP – 0) and natural refrigerant R290 (GWP – 3, ODP – 0). Through this comparison, we aim to elucidate the design adaptations and performance variations necessitated by the switch to a greener, yet inherently more challenging, refrigerant. This research not only underscores the environmental imperative but also highlights the innovative engineering solutions required to achieve sustainable refrigeration without compromising on efficiency or safety. The aim of this study is thus to conduct an in-depth analysis and evaluation of both the theoretical and practical aspects associated with the implementation of natural refrigerants, in the context of their impact on the design and operation of refrigeration systems.

HEAT EXCHANGER OPERATING PARAMETERS

The main technical parameter of a heat exchanger for calculating the heat transfer surface area is the heat transfer coefficient [1–3]. The value of the heat transfer coefficient depends on many factors, including: the design of the heat exchanger, the materials of which it is made and the operating parameters (operating temperatures, flow rate of the working fluid) [4–6]. Therefore, numerous theoretical studies and operational tests have been conducted to assess the influence of the main constructional parameters, e.g. thickness, spacing, height and material of the fins [7–9] and the influence of operational parameters – mass flow of the refrigerant, operating temperatures [10, 11] on the heat transfer coefficient and efficiency of heat exchangers. The topic of performance research is widely addressed across various industries and fields due to its crucial importance for optimizing processes and resources [12]. Fakheri [13] defined the heat exchange efficiency of a heat exchanger as the ratio of the actual heat exchange in the heat exchanger to the optimum heat exchange rate. Lotfi et al. [14, 15] investigated the efficiency of a finned heat exchanger in relation to tube and fin design. The results showed that the shape of the tubes and the design of the fins play an important role in the efficiency of the heat exchanger. Carija et al. [16] studied the efficiency of a finned heat exchanger using smooth and corrugated fins. The results of their simulation studies were confirmed by experimental studies. When using corrugated fins, the heat transfer coefficient value increased by 58% compared to the performance of the heat exchanger using smooth fins. The geometry of the individual key structural components of the finned heat exchanger is crucial, as a reduction in their dimensions has a beneficial effect on limiting the size of the exchanger, reducing the amount of materials used in its construction, which translates into a reduction in the weight of the exchanger and its price [17–19]. Sertkaya et al. [20] studied the efficiency of a standard finned heat exchanger in comparison with the efficiency of an exchanger with the same dimensions but made of aluminium foam. The pressure loss and heat transfer rate were compared. The study showed that the pressure loss and friction coefficient for the open-cell aluminum foam heat exchanger are higher compared to the finned heat exchanger. The optimum fin spacing for the exchanger design studied was also determined. M. Mehrtash et al. [21] experimentally investigated the heat flow in a finned heat exchanger as a function of different heat exchanger fin angles and fin spacing. The work investigated a wide range of fin inclination angles, from -60° to +80°. It was found that the optimum distance between the fins for all the inclinations studied varied from 9 mm for a downward-facing horizontal fin to 13 mm for an upward-facing horizontal fin, with the optimum distance for the vertical position of the fins being in the middle. The aim of the study by Bejan et al. [22] was to find an answer to the question of what the optimal size of a heat exchanger should be, as well as what design features it should have. Different heat exchanger configurations were studied as a function of temperature distribution and heat flow. The form factor of the exchangers was determined to ensure maximum thermal efficiency depending on the exchanger's operating parameters. The reduction in exchanger dimensions is associated with a reduction in the mass of refrigerant in the system, which is beneficial for improving safety when operating systems filled with flammable refrigerants. Tassou et al. [23, 24] analysed the volume of the finned heat exchanger that translates directly into the weight of the refrigerant in the system, the efficiency and the safety of the refrigeration system. Reducing the volume of the heat exchanger made it possible to minimise the risks associated with possible refrigerant leakage, which is particularly important when the system is filled with flammable or toxic refrigerant. In turn, reducing the amount of refrigerant in the refrigeration system adversely affects the performance of the refrigeration system. Limiting the mass of refrigerant in the system significantly reduces the achievable heating capacity of the refrigeration system over a wide range of variations in the system's operating parameters, which is a serious operating problem for heat pumps operating in air-to-water or air-to-air systems. Therefore, currently the main design problem of finned heat exchangers, and at the same time a challenge for designers, is to strive for an increase in the heat transfer coefficient with a reduction in the size of the exchanger, as this makes it possible to achieve an increase in the efficiency of cooling equipment, while reducing the mass of the refrigerant in the system [25–27]. In spite of numerous studies, the problem of the correct choice of heat exchanger surface area for the required capacity of refrigeration systems is still relevant. The problem is particularly relevant for evaporators. An evaporator surface area that is too small in relation to the required cooling capacity will result in incorrect operation of the refrigeration system. Therefore, there is a need for further research.

RESEARCH PROBLEM

The problem of incorrectly sizing the evaporator surface area for the refrigeration system and the resulting consequences will be presented using the example of performance testing of a compressor heat pump operating in a brine-to-water system, fed with R290. The pump is based on a scroll compressor whose catalogue cooling capacity, for a saturation temperature of 0 °C and a condensation temperature of 35 °C, is 9.23 kW. The heat pump's evaporator was selected for such operating parameters from a heat exchanger manufacturer's catalogue. The constructed heat pump (Fig. 1) was subjected to performance tests on a test stand located at the Institute of Mechanical Engineering of the Warsaw University of Life Sciences (Fig. 2). The hydraulic structure of the test stand is compliant with PN-EN 14511, and enables operational tests to be conducted in the range of heat source system temperature changes of -15 °C÷50 °C and in the range of heat distribution system temperature changes of 15 °C ÷ 70 °C. The stand›s construction is based on two 300 dm^3 buffer tanks (Figure 2). Tank Z1, which is the heat source system for the compressor heat pump, was filled with ethylene glycol. The tank is connected to the evaporator by galvanised steel pipes with a nominal diameter of 1 ¼'. Between the tank and the evaporator there is a three-way mixing valve ZM1, which allows the temperature of the heat source system to be regulated by adjusting the position of the valve insert using an analogue signal in the range of 0-10V DC. The heat load of the condenser includes a Z2 tank with a volume of 300 dm^3 filled with water. The medium flow through the evaporator and condenser is forced by circulating pumps with infinitely variable capacity control in the range 0–50 dm^3. Regeneration of the heat source system of the heat pump (tank Z1) is possible via a hydraulic connection between tank Z2 and tank Z1. The separation between the Z1 and Z2 tanks

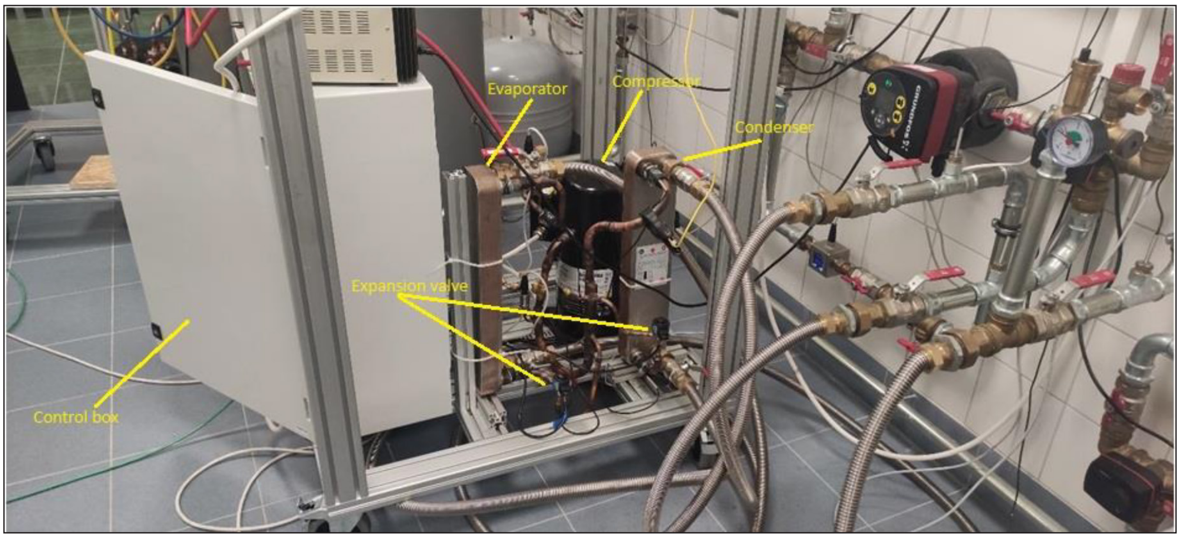

Figure 1. Compressor heat pump operating in a brine-to-water system connected to a test stand

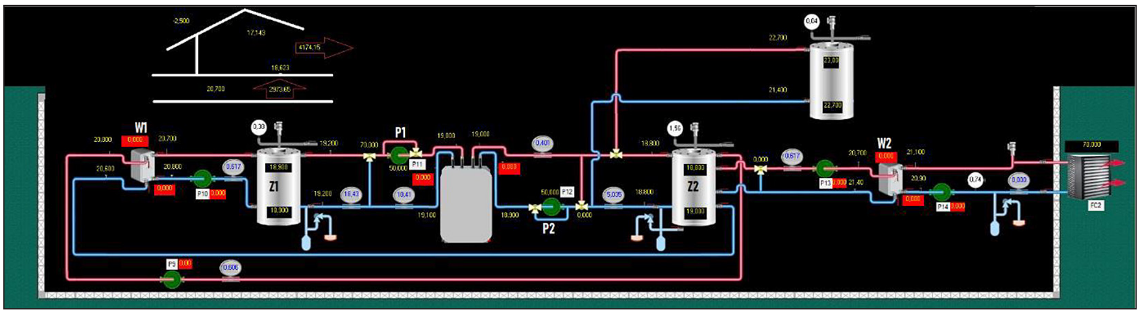

Figure 2. Visualisation of test stand operation

is implemented using a plate heat exchanger, and the media flow is forced by pumps P9 and P10. Excess heat on the heat distrubution system's side is dissipated into the surroundings by means of fan cooler FC2, which is hydraulically connected to tank Z2 via plate heat exchanger W2. The hydraulic system is metered. The medium flow is measured using ultrasonic flowmeters equipped with analogue outputs with a voltage range of 0–10 V DC. Temperature measurement is carried out using PT1000 sensors, made in a 4-wire system, placed in capillaries. Control of the capacity of the circulating pumps forcing the media through the evaporator and through the condenser is carried out using PID controllers, for which the setpoint is the difference between the inlet temperature and the outlet temperature of the working fluid. The regulation of the temperature of the heat source system and heat distribution system is carried out using PID controllers that regulate the position of the mixing valve inserts ZM1 and ZM2. The system control algorithm is implemented in a PLC S7-1500 controller. The controller is equipped with expansion modules for discrete and analogue inputs and outputs, to which all adjusters and measuring transducers are connected.

The operation of the system is monitored using SCADA software enabling the acquisition of measurement data installed on a PC (Fig. 2). Communication between the controller and the computer is carried out using the Ethernet TCP-IP protocol.

Performance tests were performed in accordance with PN-EN 14511 for brine-to-water heat pumps. According to the standard, the temperature of the lower source was 0 °C and the temperature of the upper source was 35 °C. The flow of the medium through the condenser and evaporator was controlled by PI controllers so that the gradient between the outlet and inlet temperatures for the evaporator was 3 K and for the condenser was 5 K. Figure 3 shows the temperature curve of the media – glycol on the heat source system side and water on the heat distribution system side. The temperature setpoint for the heat source system was 0 °C and the temperature setpoint for the heat distribution system was 35 °C. When the heat pump under test is switched on, the temperature of the heat distribution system increases and the temperature of the heat source

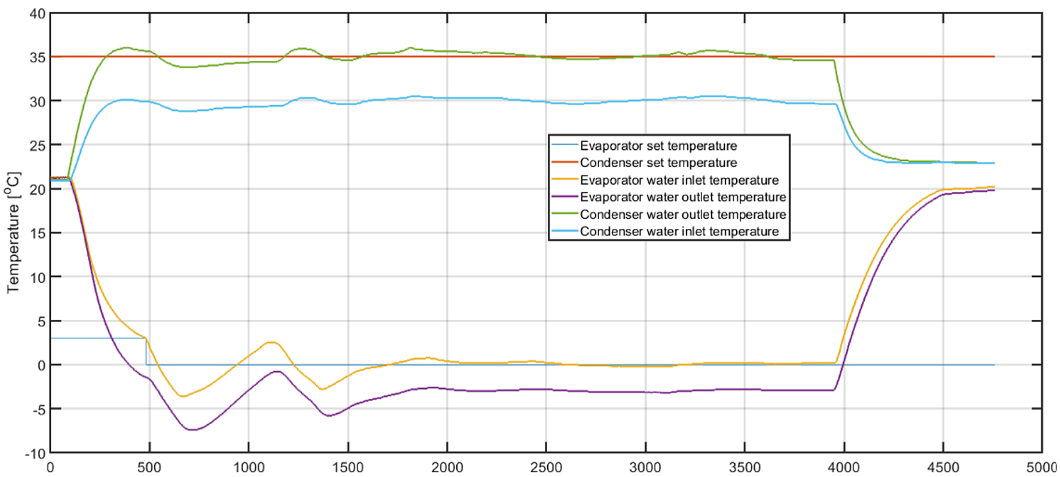

Figure 3. Temperature curve of the heat source and heat distribution system of the heat pump under test during the experiment

system decreases to the set values. It took 2,500 seconds to stabilise the temperature of the heat source system and heat distribution system. When steady state was reached, the heating capacity value which was 8 kW and the cooling capacity value which was 4.6 kW was read (Fig. 4).

The experiment was repeated at heat source system temperatures of 7 °C and 10 °C (Fig. 5). Once the heat source system temperature was stabilised at 7 °C and the heat distribution system temperature at 35 °C, the cooling capacity was read at 6 kW (Fig. 6). For these operating parameters, the heating capacity increased to a value of 9.2 kW. The temperature of the heat source system was then increased to a value of 10 °C, while the temperature of the heat distribution system was kept constant at 35 °C. The increase in the temperature of the heat source system resulted in an increase in the demand for cooling capacity. It can be seen from the heating and cooling capacity curves shown in Figure 6 that, despite the increase in the heat source system temperature, the available cooling capacity could not be utilised, which would result in an increase in heating capacity (Fig. 6). The available cooling capacity could not be used due to the evaporator's insufficient heat transfer surface area. In this case, an increase in the mass of refrigerant fed into the evaporator through the expansion valve reduced the refrigerant overheating to 0 K, resulting in the compressor being flooded with liquid refrigerant.

The analysis presented here shows that the problem of selecting heat exchangers using new, environmentally friendly refrigerants which, according to Regulation (EU) No 2024/573 of the European Parliament and of the Council, are to replace synthetic refrigerants, is still relevant. For plate heat exchangers used as evaporators and condensers in ground-source heat pumps, the problem is somewhat less complicated, as they are homogeneous in terms of the material used. The problem is more complicated in the case of finned heat exchangers acting as evaporators in air-to-water compressor heat pumps, as the exchangers are constructed from different materials (tubes – copper, fins – aluminium) and have a different design in terms of length and arrangement of the number of fluid circuits in relation to each other. It is well known that the refrigerant flow velocity has the greatest influence on the value of the heat transfer coefficient; however, one should bear in mind that an increase in the refrigerant flow velocity significantly increases the flow resistance, which is a disadvantage from an operational point of view. Therefore, this article attempts to assess the influence of design and operating parameters on the performance of a finned heat exchanger fed with synthetic (R410a) and natural (R290) refrigerants.

RESEARCH OBJECT AND METHODOLOGY

Finned heat exchangers most often act as an evaporator in air-to-water heat pumps and serve as an evaporator and condenser in air-to-air heat pumps. They consist of coils made of copper tubes that form the refrigerant circuits and fins designed to increase the heat transfer surface area. The air flows between the open spaces between the fins giving up heat to the

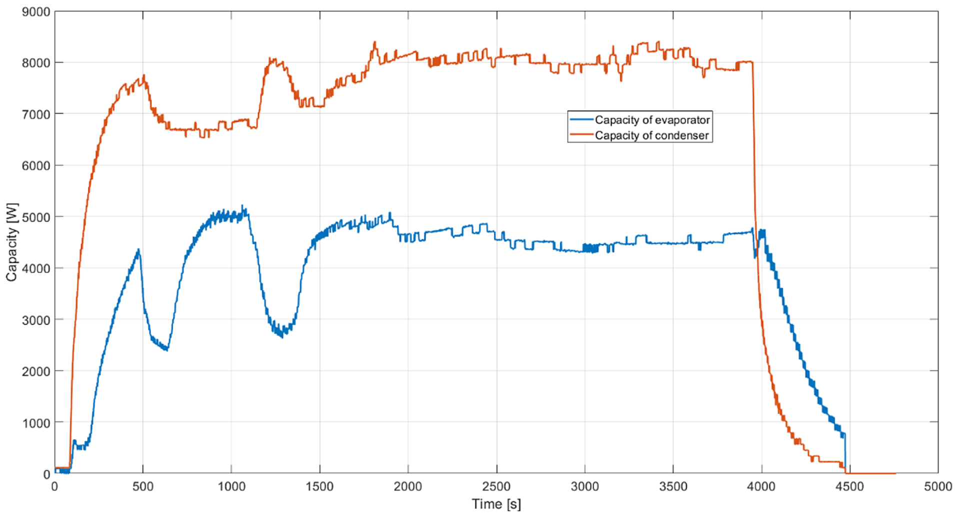

Figure 4. Heating and cooling capacity values of the tested heat pump during the experiment with a heat source system temperature of 0 °C and a heat distribution system temperature of 35 °C

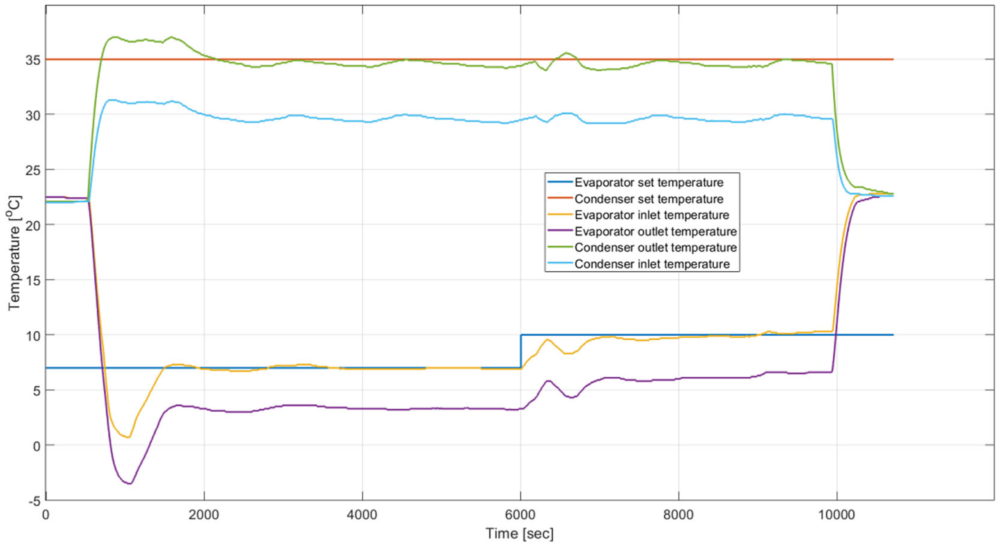

Figure 5. Temperature curve of the heat source and heat distribution system of the heat pump under test during the experiment

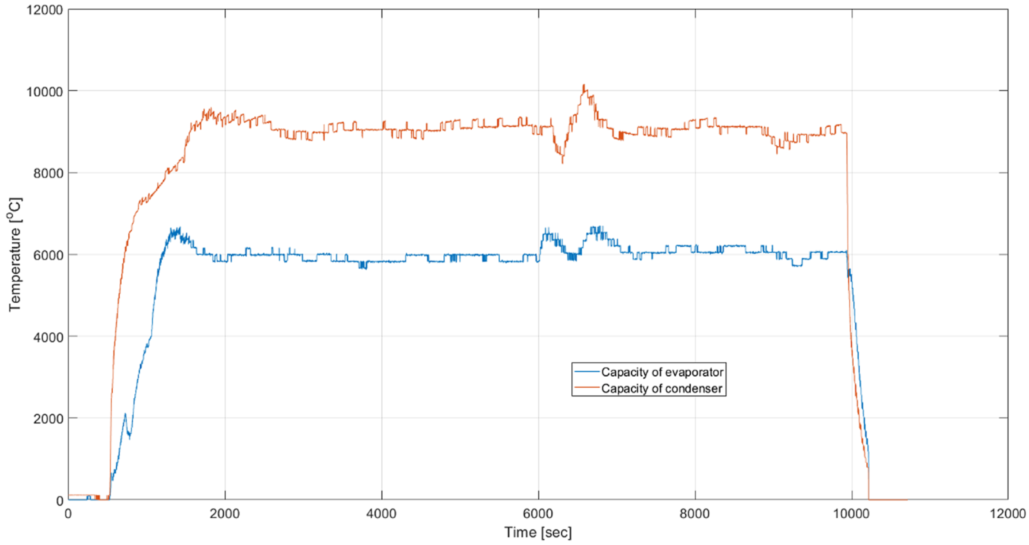

Figure 6. Heating and cooling capacity values of the tested heat pump during the experiment with a heat source system temperature of 7 °C and 10 °C and a heat distribution system temperature of 35 °C

refrigerant flowing in the tubes on which the fins are mounted. The main design parameters of a finned heat exchanger are: the diameter of the coil, the length of the medium circuit, the thickness and the spacing of the fins. The capacity of a finned heat exchanger identical to that of a plate heat exchanger can be calculated using Equation 1.

$$\dot{Q} = A \cdot k \cdot \Delta t_m [W] \quad (1)$$

where: A – heat transfer surface area [m^2], $\dot{Q}$ – capacity [W], k – heat transfer coefficient [$\frac{W}{m^2 \cdot K}$], Δt_m – logarithmic mean temperature difference [K].

The heat transfer coefficient k depends on both the design parameters of the heat exchanger, such as the diameter of the coil, the thickness and spacing of its fins and the materials used in its construction, as well as on its operating parameters, air velocity and refrigerant flow rate. Determining the heat transfer coefficient value is the main problem in selecting the heat transfer surface area of a finned heat exchanger, as compared to a plate heat exchanger it is constructed from at least two materials (copper, aluminium) with different thermal conductivity values. For this reason, an attempt was made to carry out simulation tests and, based on the test results, comparative analyses of the operating parameters of a

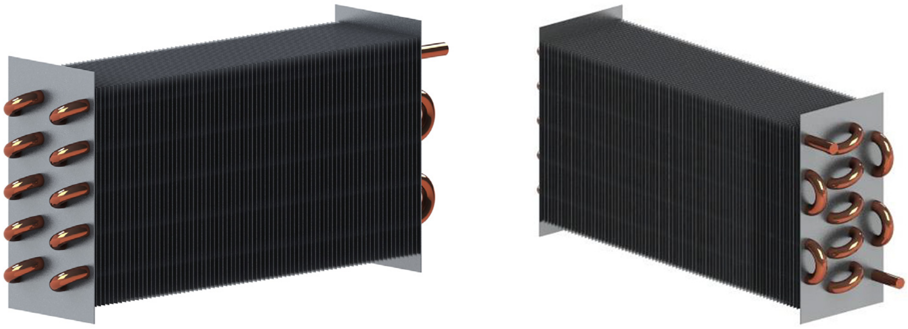

Figure 7. Modelled heat exchanger

finned heat exchanger supplied with synthetic refrigerant R410a and environmentally friendly refrigerant R290. A model of the finned heat exchanger was made in SolidWorks software (Fig. 7) and simulation studies were carried out in SolidWorks software using the Flow Simulation library. The SolidWorks software, similar to the ANSYS software used in the research by Prasetyo et al. [28–29], is based on fundamental principles of physics, such as the Navier-Stokes equations and the heat transfer equation, to simulate flow and heat exchange in 3D models. It employs numerical methods, such as the finite element method (FEM), to approximate solutions to the governing equations in discretized models. Simulation studies were carried out for several design variants. The number, spacing and material of fins were varied (Table 1). Simulation tests were carried out for different operating conditions. The air volume flow was varied with a step of 0.5 m^3/s and the refrigerant mass flow with a step of 0.01 kg/s (Table 2).

COMPARATIVE ANALYSIS OF THE OPERATING PARAMETERS OF A FINNED HEAT EXCHANGER FED WITH R410A AND R290

The simulation studies included more than 500 simulations. Due to the huge amount of data, this paper presents the key results.

Effect of fin material on heat exchanger output

Figure 8 shows an example of a simulation of a modelled finned heat exchanger. The tests were carried out with an inlet temperature of R410a refrigerant of 0 °C, a refrigerant mass flow rate of 0.05 kg/s and an air temperature of 10 °C and a volume flow rate of 2 m^3/s. For the case studied, the fins were made of 1060-H12 alloy. For the specified initial operating conditions, the outlet temperature of the refrigerant

Table 1. Design parameters of the tested finned heat exchanger

Parameter	Value
Number of fins	$n_1 = 78$, $n_2 = 39$
Fin spacing [mm]	$l_1 = 2.5$, $l_2 = 5$
Fin thickness [mm]	d = 0.2
Tube diameter [mm]	φ = 6.35 (¼ inch)
Fin material	Aluminium alloys: alloy 1060, 1060-H12, alloy 2014, alloy 2024

Table 2. Operating parameters of the tested finned heat exchanger

Parameter	Value	Notes
Volume flow of air [m^3/s]	$\dot{V}$ = (1 ÷ 4)	With a step of 0.5
Refrigerant mass flow [kg/s]	$\dot{m}$ = 0.01 ÷ 0.05	With a step of 0.01
Refrigerant	R410a, R290	–
Refrigerant inlet temperature [°C]	0	–
Air inlet temperature [°C]	10	–

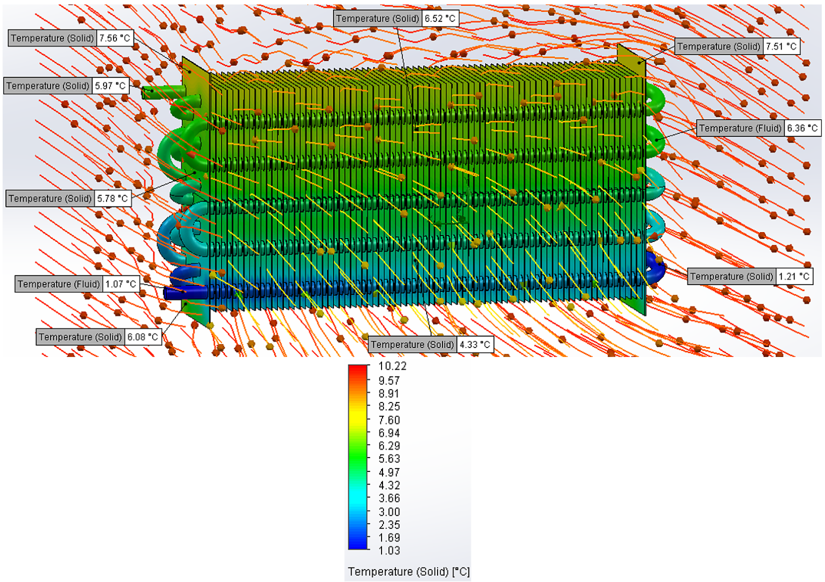

Figure 8. Example analysis of the temperature distribution on the structural components of the exchanger under test

reached 5.97 °C, the outlet air temperature dropped to 8.87 °C and the cooling capacity of the heat exchanger reached 184 W.

Figure 9 shows the results of simulation tests to assess the influence of the fin material on the cooling capacity of the heat exchanger with fins made of different aluminium alloys, whose thermal conductivity coefficient α was in the range 140 W/mK ÷ 230 W/mK. Based on the results, it can be concluded that the cooling capacity of the heat exchanger is not significantly affected by changing the fin material. The cooling capacity of the exchanger is decisively influenced by the operating parameters: the mass flow of the refrigerant and the volume flow of air through the fins of the exchanger.

At a constant air volume flow of 1m^3/s and a refrigerant mass flow of 0.01 kg/s, changing the fin material does not change the cooling capacity of the exchanger. Small changes in the cooling capacity of the exchanger due to an increase in the thermal conductivity coefficient of the fin material at constant air volume flow occur with an increase in the refrigerant mass flow rate, but these are small differences. The largest increase in cooling capacity of 2 W occurred for a mass flow of 0.05 kg/s (Fig. 9).

Figure 10 shows an example of the temperature distribution on a single fin made of different aluminium alloys and located in the central part of the modelled exchanger. As can be seen from the temperature distribution, the largest differences occur on the upper and lower parts of the fin. In the central part, the temperature distribution regardless of material is uniform. However, when analysing the minimum and maximum temperature values (Table 3) determined on the surface area of individual fins (Fig. 10), it is possible to observe a certain regularity that the higher the conductivity coefficient α of the material from which the exchanger fins are made, the more uniform the temperature distribution on the examined fin is, as evidenced by the calculated differences between the maximum temperatures occurring on the fins and the minimum ones. The difference between the maximum and minimum temperatures for fins made of alloy 2024 and those made of alloy 1060-H12 is 13.59 %. For

Table 3. Minimum and maximum temperature of fins made of different aluminium alloys

Material	Minimum temperature [°C]	Maximum temperature [°C]	$\Delta T = T_{max} - T_{min}$ [°C]	Average temperature [°C]
Alloy 2024	3.7747	9.3634	5.5887	7.4650
Alloy 2014	3.8278	9.3126	5.4848	7.4540
Alloy 1060	4.0328	9.0830	5.0502	7.4166
1060 - H12	4.1333	8.9623	4.8290	7.4025

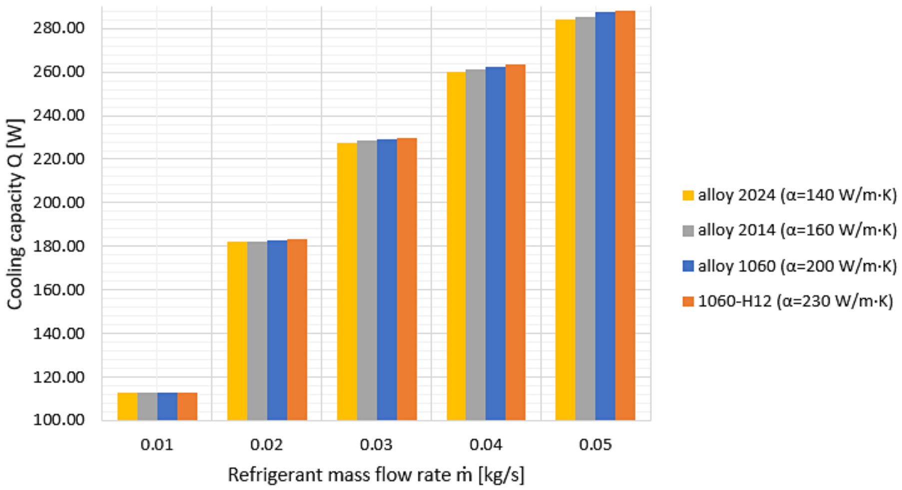

Figure 9. Cooling capacity as a function of the refrigerant flow for 1 m^3/s, for R410a

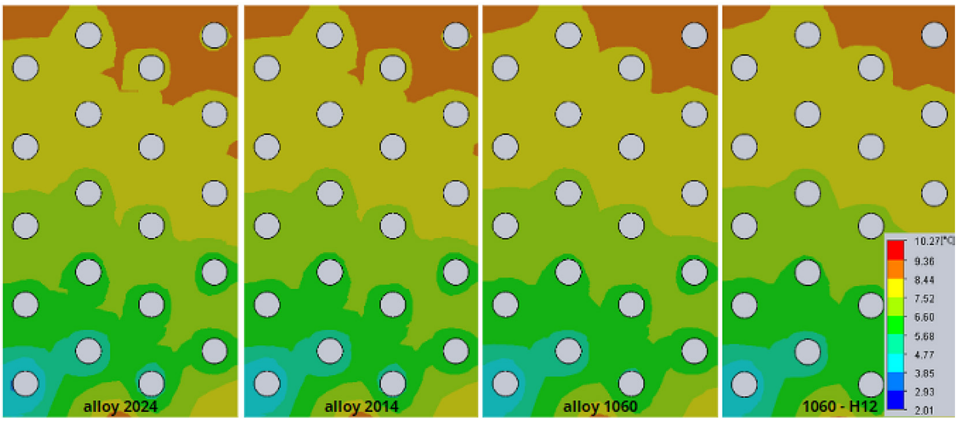

Figure 10. Temperature distribution for fins in the central part of the heat exchanger

fins with larger α coefficients, a more uniform heating of the fin surface area occurs, but these differences are not significant. The difference between the average temperature of the exchanger made of alloy 2024 and the exchanger made of alloy 1060-H12 is only 0.84 %, which means that the material does not significantly affect the performance of the exchanger.

Figure 11 shows the temperature distribution of the fins of the exchanger fed with R410a and R290. The Figure shows that the average fin temperature of a heat exchanger supplied with R290 is lower than the average fin temperature of a heat exchanger supplied with R410a (Table 4). The difference between the average fin temperatures of heat exchangers filled with refrigerants R290, and R410a, is 21.27%. As a result, the cooling capacity of an exchanger charged with R290 is higher than that of an exchanger charged with R410a. The minimum, maximum and average temperatures on each fin were determined (Table 5).

Comparative analysis of the capacity of a finned heat exchanger fed with R410a and R290

Figures 12 and 13 show the cooling capacity of a fin evaporator supplied with R410a (Figure 12) and R290 (Figure 13) as a function of changes in refrigerant mass flow in the range of 0.01 kg/s ÷ 0.05 kg/s with a change in air volume flow of 1 m^3/s ÷ 1 m^3/s for a refrigerant saturation temperature of 0 °C and an air temperature of 10 °C.

A comparison of the heat exchanger capacities shown in the Figures indicates that, under the same operating conditions, the use of R290 increases the cooling capacity of the system relative to feeding the exchanger with R410a. In order to make a comparative analysis to evaluate the increase and variation in the capacity of the exchanger supplied with R290 in relation to the heat exchanger supplied with R410a, the coefficient

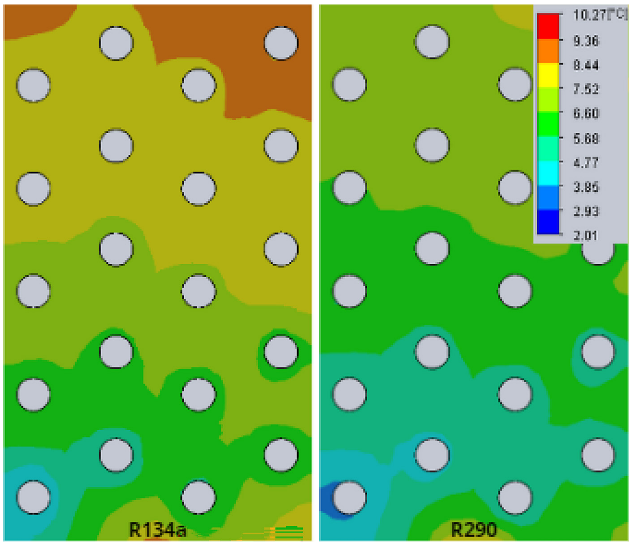

Figure 11. Temperature distribution for fins in the central part of the heat exchanger

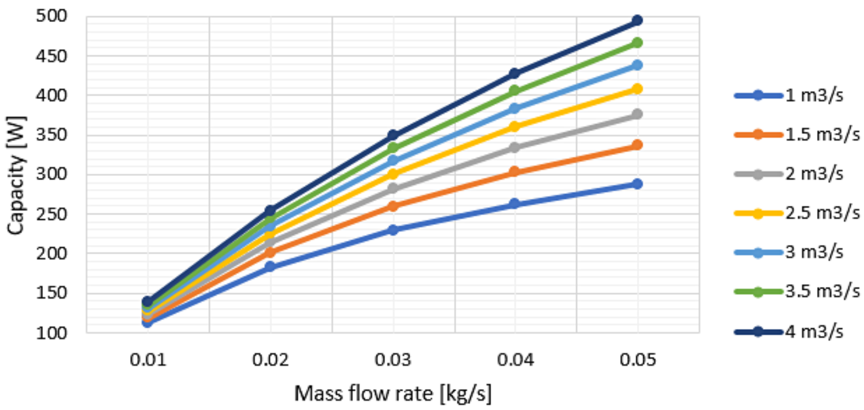

Figure 12. Cooling capacity of a finned heat exchanger supplied with R410a for a refrigerant saturation temperature of 0 °C and an air temperature of 10 °C

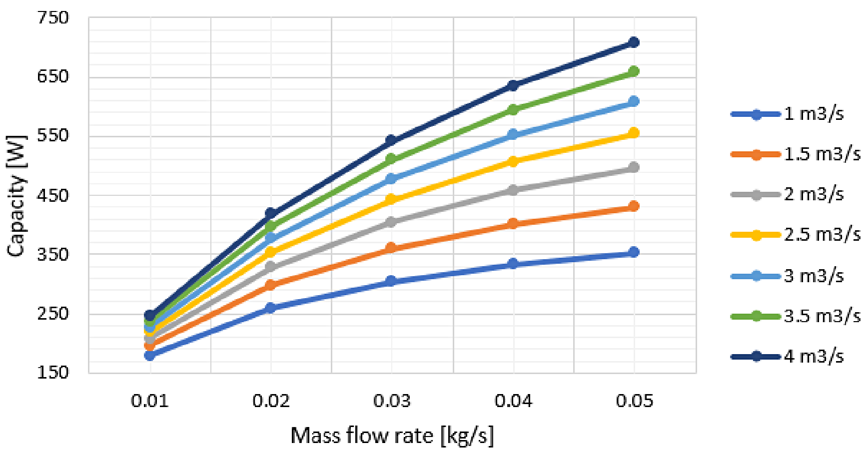

Figure 13. Cooling capacity of a finned heat exchanger supplied with R290 for a refrigerant saturation temperature of 0 °C and an air temperature of 10 °C

Table 4. Fin temperatures

Refrigerant	Minimum temperature [°C]	Maximum temperature [°C]	Average temperature [°C]
R134a	4.0328	9.0830	7.4166
R290	2.8889	8.0061	5.8392

C was introduced. The coefficient C is the ratio of the cooling capacity of the exchanger charged with R290 to the cooling capacity of the exchanger charged with R410a (Equation 2) (Table 5).

$$C = \frac{A}{B} \tag{2}$$

where: C – coefficient [-], A – capacity of exchanger supplied with R290 [W], B – capacity of exchanger supplied with R410a [W].

The results presented in the table show that the greatest increase in performance of a heat exchanger supplied with R290 relative to a heat exchanger supplied with R410a occurs at the lowest refrigerant mass flow rate and at the highest air volume flow rate. With the same heat exchanger surface area for this operating point of the exchanger, the increase in capacity when the exchanger is supplied with R290 compared to the capacity of the exchanger supplied with R410a is 77%. At the highest refrigerant mass flow rate and lowest air volume flow rate, the capacity of the exchanger supplied with R290 refrigerant is 23% higher than when the exchanger is fedwith R410a. The analysis presented shows that, for the same operating parameters, the heat transfer surface area of a finned heat exchanger supplied with R290 can be reduced compared to a heat exchanger supplied with R410a in order to achieve the same cooling capacity. As a result, the size of the exchanger can be reduced, thereby reducing its refrigerant charge rate, or the fin spacing can be reduced, which will reduce the frosting of the heat exchanger and reduce the energy-intensive evaporator defrost process.

Comparison of the performance of a heat exchanger supplied with R290 refrigerant at reduced heat transfer surface area with the performance of a reference heat exchanger supplied with R410a refrigerant

As the process of defrosting the evaporator is energy intensive, this paper presents a comparative analysis of a fin exchanger with twice the fin spacing, compared to a reference exchanger (5 mm) supplied with R290 refrigerant, with a reference exchanger supplied with R410a refrigerant. The reduction in the number of fins resulted in a reduction in the heat transfer surface area from a value of p_1 = 1.0379 m^2 (reference exchanger) to a value of p_2 = 0.6047 m^2, resulting in a 58.26% reduction in heat transfer surface area. Figure 14 shows the output of the exchanger fed with R290 refrigerant with a 58.26% reduction in heat transfer surface area, compared to the reference heat exchanger, with a variable refrigerant mass flow rate of 0.01 kg/s ÷ 0.05 kg/s and a variable air volume flow rate of 1 m^3/s ÷ 4 m^3/s. The tests were performed for a refrigerant saturation temperature of 0 °C and an air temperature of 10 °C. Comparing the capacity of the heat exchanger with reduced heat transfer surface area supplied with R290 refrigerant and the reference exchanger fed with R410a refrigerant, despite the reduced heat transfer surface area, the capacity of the exchanger supplied with R290 refrigerant is still higher than that of the reference exchanger at most operating points, as estimated using the value of the coefficient C (Table 6). The critical

Table 5. Coefficient C values depending on the exchanger's operating parameters

Parameter		Mass flow rate of the medium [kg/s]				
		0.01 kg/s	0.02 kg/s	0.03 kg/s	0.04 kg/s	0.05 kg/s
Volume flow rate [m^3/s]	1 m^3/s	1.59	1.42	1.33	1.27	1.23
	1.5 m^3/s	1.65	1.49	1.39	1.32	1.28
	2 m^3/s	1.69	1.54	1.44	1.37	1.32
	2.5 m^3/s	1.72	1.57	1.47	1.41	1.35
	3 m^3/s	1.74	1.60	1.51	1.44	1.38
	3.5 m^3/s	1.75	1.63	1.53	1.46	1.41
	4 m^3/s	1.77	1.65	1.55	1.49	1.43

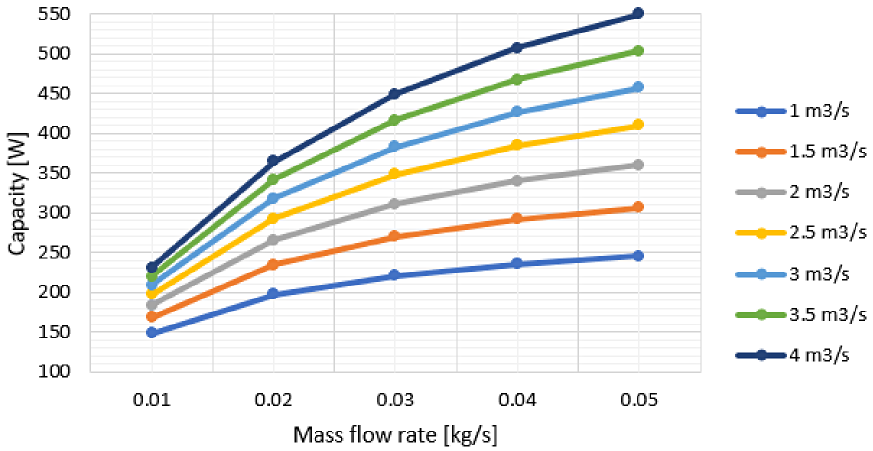

Figure 14. The cooling capacity of a finned heat exchanger supplied with R290 with a reduced heat transfer surface area for a medium saturation temperature of 0 °C and an air temperature of 10 °C

Table 6. Values of coefficient C depending on the exchanger operating parameters for a refrigerant saturation temperature of 0 °C and an air temperature of 10 °C, with reduced heat exchanger surface area supplied with refrigerant R290

Parameter		Mass flow rate of the medium [kg/s]				
		0.01 kg/s	0.02 kg/s	0.03 kg/s	0.04 kg/s	0.05 kg/s
Volume flow rate [m³/s]	1 m³/s	1.32	1.08	0.97	0.90	0.85
	1.5 m³/s	1.42	1.17	1.04	0.96	0.91
	2 m³/s	1.49	1.24	1.10	1.02	0.96
	2.5 m³/s	1.55	1.30	1.16	1.07	1.00
	3 m³/s	1.59	1.35	1.21	1.11	1.04
	3.5 m³/s	1.63	1.40	1.25	1.15	1.08
	4 m³/s	1.66	1.44	1.29	1.19	1.11

Table 7. Values of variable C

Parameter	0.01 kg/s	0.02 kg/s	0.03 kg/s	0.04 kg/s	0.05 kg/s
1 m³/s	0.83	0.76	0.73	0.71	0.70
1.5 m³/s	0.86	0.79	0.75	0.73	0.71
2 m³/s	0.88	0.81	0.77	0.74	0.73
2.5 m³/s	0.90	0.83	0.79	0.76	0.74
3 m³/s	0.92	0.84	0.80	0.77	0.75
3.5 m³/s	0.93	0.86	0.82	0.79	0.77
4 m³/s	0.94	0.87	0.83	0.80	0.78

operating points are those corresponding to high refrigerant mass flow rates and low air volume flow rates (Table 6-red). For these operating parameters, the capacity of the reference heat exchanger supplied with R410a is higher than that with a reduced heat transfer surface area fed with the environmentally friendly refrigerant R290. In these instances, the finned heat exchanger behaves like the plate heat exchanger presented in Chapter 2 of this paper.

Comparison of the performance of a reference finned heat exchanger supplied with environmentally friendly refrigerant R290 with a finned heat exchanger with reduced heat transfer surface fed with environmentally friendly refrigerant R290

The analyses show that the developed heat transfer surface area of the exchanger is an important design parameter. Therefore, the value of the coefficient C was determined for the reference heat exchanger and the heat exchanger with a reduced heat transfer area fed with R290 (Table 7). The coefficient C is the ratio of the cooling capacity of the heat exchanger with reduced heat transfer surface area supplied with R290 to the cooling capacity of the reference heat exchanger fed with R290. Table 7 shows that if the heat transfer surface area of a heat exchanger supplied with R290 is reduced, its efficiency will fall within the range of 6 ÷ 30%, depending on the operating parameters.

CONCLUSIONS

This paper presents a comparative analysis of a finned heat exchanger fed with the synthetic refrigerant R410a currently commonly used in compressor heat pumps and the environmentally friendly refrigerant R290. The origins of undertaking the research are the timetable for withdrawal of synthetic refrigerants introduced by Regulation (EU) 2024/573 of the European Parliament and of the Council, according to which they are to be replaced by environmentally friendly refrigerants, and the presented operational problems associated with the selection of heat exchangers fed with natural refrigerants. The analyses show that the fin material is not an important design parameter affecting the cooling capacity of a finned heat exchanger. However, the heat transfer surface area is an important design parameter affecting the performance of the heat exchanger. By replacing the synthetic refrigerant R410a with the environmentally friendly refrigerant R290, the heat transfer surface area of a finned heat exchanger can be reduced by up to 50%, which can be achieved by reducing the number of fins and increasing the spacing between them, or by reducing the size of the heat exchanger. The first solution will have a beneficial effect on reducing the degree of frosting of the heat exchanger and, consequently, on reducing the electricity consumption necessary for its defrosting. The second solution will make it possible to reduce the weight of the flammable (Class A3) refrigerant in the system, ensuring that the refrigerant is less likely to ignite. In conclusion, our research demonstrates that transitioning to R290 as a refrigerant in finned heat exchangers is a viable and beneficial direction for future development. The findings indicate that significant reductions in heat transfer surface area are achievable without compromising the performance of the heat exchanger. This not only aligns with regulatory requirements but also presents practical advantages, such as reduced energy consumption and increased safety through lower refrigerant weight. Therefore, this study underscores the potential for further exploration and optimization of natural refrigerants in refrigeration systems, advocating for continued research to refine these systems and fully harness the environmental and operational benefits offered by natural refrigerants like R290.

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