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Analysis and Evaluation of Heat Pipe Efficiency to Reduce Low Emission with the Use of Working Agents R134A, R404A and R407C, R410A

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Citation: Adrian, Ł.; Szufa, S.; Piersa, P.; Kuryło, P.; Mikołajczyk, F.; Kurowski, K.; Pochwała, S.; Obraniak, A.; Stelmach, J.; Wielgosiński, G.; et al. Analysis and Evaluation of Heat Pipe Efficiency to Reduce Low Emission with the Use of Working Agents R134A, R404A and R407C, R410A. *Energies* **2021**, *14*, 1926. https://doi.org/10.3390/en14071926

Academic Editor: Manabendra Saha

Received: 18 February 2021
Accepted: 26 March 2021
Published: 31 March 2021

Abstract: This paper presents an analysis of methods to increase the efficiency of heat transfer in heat exchangers. The scope of the research included analysis of efficiency optimization using the example of two tubular heat exchanger structures most often used in industry. The obtained efficiency of heat recovery from the ground of the examined exchangers was over 90%, enabling the reduction of emissions of the heating systems of buildings. The paper presents the results of tests of two types of heat pipes using R134A, R404A, and R407C working agents. The paper also presents the results of experimental tests using the R410A working medium. The results included in the study will also enable the effective use of land as a heat store.

Keywords: heat pipe; heat transfer; heat exchanger; phase change; evaporation; condensation; low emission

1. Introduction

It is widely acknowledged that global demand for energy is constantly increasing. Industries are finding it increasingly difficult to stay competitive because companies in every market are looking for new ways to maximize efficiency while lowering energy costs and improving their environmental image. These challenges are complex and multi-faceted [1].

Heat exchangers play one of the most important roles in the pursuit of optimization while also saving energy. Installations of high-efficiency heat exchanger systems provide new opportunities to reduce energy costs and CO_2 emissions without compromising the performance and quality of the final product.

Many manufacturers of thermal equipment design their cooling and heating equipment to be compact. The compact design of the heat exchanger can significantly reduce installation costs in the replacement of older technology and increase plant efficiency. It is easier to install a heat exchanger in an area in which space for facilities is a critical factor.

The spectacular increase in sales of heat pumps and ground heat exchangers in the last 3 years, and the likely trend for coming years, indicates that the target market is interested in the solutions examined in this study. According to the report of the Polish Organization for the Development of Heat Pump Technology (PORT PC), the heat pump market, and thus that of ground heat exchangers, in Poland in 2020 (one-fifth of heat pumps installed in Poland have a lower source of heat in the form of soil) recorded large sales increases. Since 2017, the heat pump market has grown more than five-fold and, according to the same organization, this trend will continue for the next decade. Renewable energy support programs have contributed to a significant increase in sales of heat pumps in recent years. It should be emphasized that the Polish heat pump market is developing practically without the direct support of investments in this technology from the state. Subsidies and subsidy programs exist that include investments in heat pumps, but there is currently no direct subsidy program for the pumps themselves, as is the case with other renewable energy systems. The results obtained during research into the high efficiency of heat pipe heat exchangers (92%) show that it is possible to use heat pipes as heat exchangers that intensify the heat exchange between lower heat sources from renewable sources (e.g., ground) and heat receivers of central heating devices for buildings (e.g., the heat pump).

Contamination downtime is also an important issue in industrial plants. This is not only an issue regarding productivity—it also has an impact on efficiency. As contamination accumulates, thermal efficiency and pressure decrease, requiring more energy. Pumping fluid through a dirty heat exchanger, for example, also requires more power and considerably greater energy consumption.

This study clearly indicates that the installation of more efficient heat exchangers is often the best means of removing limitations caused by insufficient thermal or cooling capacity. More effective heat flow means that more energy can be used by the technological process, which allows an increase in the available production capacity while reducing costs [2,3].

Thus, understanding relatively simple means of increasing the efficiency of a heat exchanger can be useful in evaluating new technologies and comparing suppliers. There has been significant development in the use of heat pipes over the last half-century, initiated by the use of heat pipes in space science. Today, they are used in many industries. One of the most interesting applications of heat pipes is their use for drying and cooling air, in addition to heat recovery and passive cooling of rooms. Other important applications are pipes as solar collectors and as passive heat exchangers for regulating ground temperature and heating the surface of bridges and viaducts. Due to its structure, the heat pipe is an economical heat exchanger compared to other commonly used approaches. For this reason, and in an era of striving to save [4] and limit the unnecessary dissipation of energy, interest in this type of heat exchangers is still growing [5]. To facilitate this technology, a series of tests has been found to be necessary to analyze the operation of heat pipes and to identify the most effective heat exchangers of the heat pipe type depending on their operating conditions [6]. The current research included an analysis of the efficiency optimization problem based on the example of two types of tubular heat exchanger structures most commonly used in industry.

The research on effective methods of increasing heat transfer in cooling systems was also associated with the need to analyze the most important factors influencing the efficiency of thermal systems using R134A, R404A, R407C, and R410A working agents [7].

These factors were selected due to their thermodynamic properties in the temperature range studied, their availability, and their wide use in civil engineering and air conditioning. As part of this work, the Department of Thermal Technology of the Institute of Fluid-Flow Machinery of the Lodz University of Technology also conducted experimental research on the working factor R410A, the results and conclusions of which are included in the comparative analysis of the current paper. R410A is used in refrigeration equipment, air-cooled chillers, and heat pumps. This factor is also used in sea and land transport, and cold stores [8].

Experimental studies were carried out in conditions corresponding to natural work, i.e., in temperature conditions of heat recovery, both in air-conditioning and ventilation installations, and in construction engineering [9,10]. They consisted of testing a heat exchanger of the heat pipe type in the temperature limits of the lower source between 15 and 50 °C. The effects of phase transformations for the aforementioned working factors and their quantity were investigated, in addition to their influence on the operation and efficiency of the heat pipe. The main focus of this research was to analyze and evaluate the efficiency of the heat pipe for different working factors for different temperatures of the heat supply to the heat pipe [11].

2. Materials and Methods

2.1. Used Materials and Their Properties

In this study, our team conducted research using 4 different refrigerants: R134A, R404A, R407C and R410A. R134A refrigerant is used in medium-temperature, small and medium-sized refrigeration and air-conditioning installations. R134A has a faint ethereal odor and is colorless. Its chemical formula is CH2F-CF3. It is a replacement for the withdrawn R12 refrigerant.

R404A refrigerant is a synthetic refrigerant of near azeotropic mixtures. This refrigerant is a replacement for R502, R22, and R507. R404A refrigerant is used in refrigeration and air-conditioning units.

R407C is a synthetic azeotropic mixture. It is used as a replacement for R22. Heat pipe tests were carried out using working fluids such as R134A, R404A, R407C, and R410A. These factors were selected due to their thermodynamic properties in the studied temperature range, their availability, and the affordability of their use in construction engineering and air conditioning.

R410A refrigerant is a near azeotropic mixture of R32 (50%) and R125 (50%), used in refrigeration and air conditioning devices with an evaporation temperature from −50 to 20 °C and, in some applications, from −70 to 40 °C, e.g., as a replacement for R13B1.

2.2. Assembly of the Test Bench

The experimental tests were conducted on a heat tube exchanger made of copper, with a length of 1769 mm, an 18 mm outside diameter, and a 1 mm wall thickness (Figure 1) [12,13].

Heat pipe data:

- internal diameter: dw = 0.016 (m)
- outer diameter: dz = 0.018 (m)
- tube height: h = 1.769 (m)

In the Department of Heat and Refrigeration Technology of the Institute of Fluid-Flow Machinery of the Technical University of Lodz, a stand for testing heat exchangers of the heat tube type was designed and built. The setup was equipped with a heat-flow apparatus, which included, among other things, ultrathermostats for supplying pipe-in-pipe exchangers serving the heat pipe, circulation pumps, and liquid coolers. The exchanger was made of copper due to the high heat transfer coefficient and the availability and low price of copper pipes. The length was selected so that the part receiving and giving off heat is in the middle of the 0.5–1 m range, which corresponds to a typical series of heights of ventilation and air conditioning ducts and units. As part of the subject of this work, experimental tests were also carried out on a heat pipe made of brass, with a length of 550 mm, an external diameter of 22 mm, and a wall thickness of 1 mm (Figure 2), the results and conclusions of which are included in the comparative analysis in this paper (see work in [14,15]).

The first tests were carried out on a tube with vacuum inside and then filled with air to confirm the thesis that heat conduction through the wall of the tube along its length was negligible. The research consisted of placing both heat supply (i.e., the evaporator) and heat removal (i.e., the condenser) in the flow exchanger, which supplies and receives heat from the heat pipe on the principle of a pipe-in-pipe heat exchanger (Figure 3).

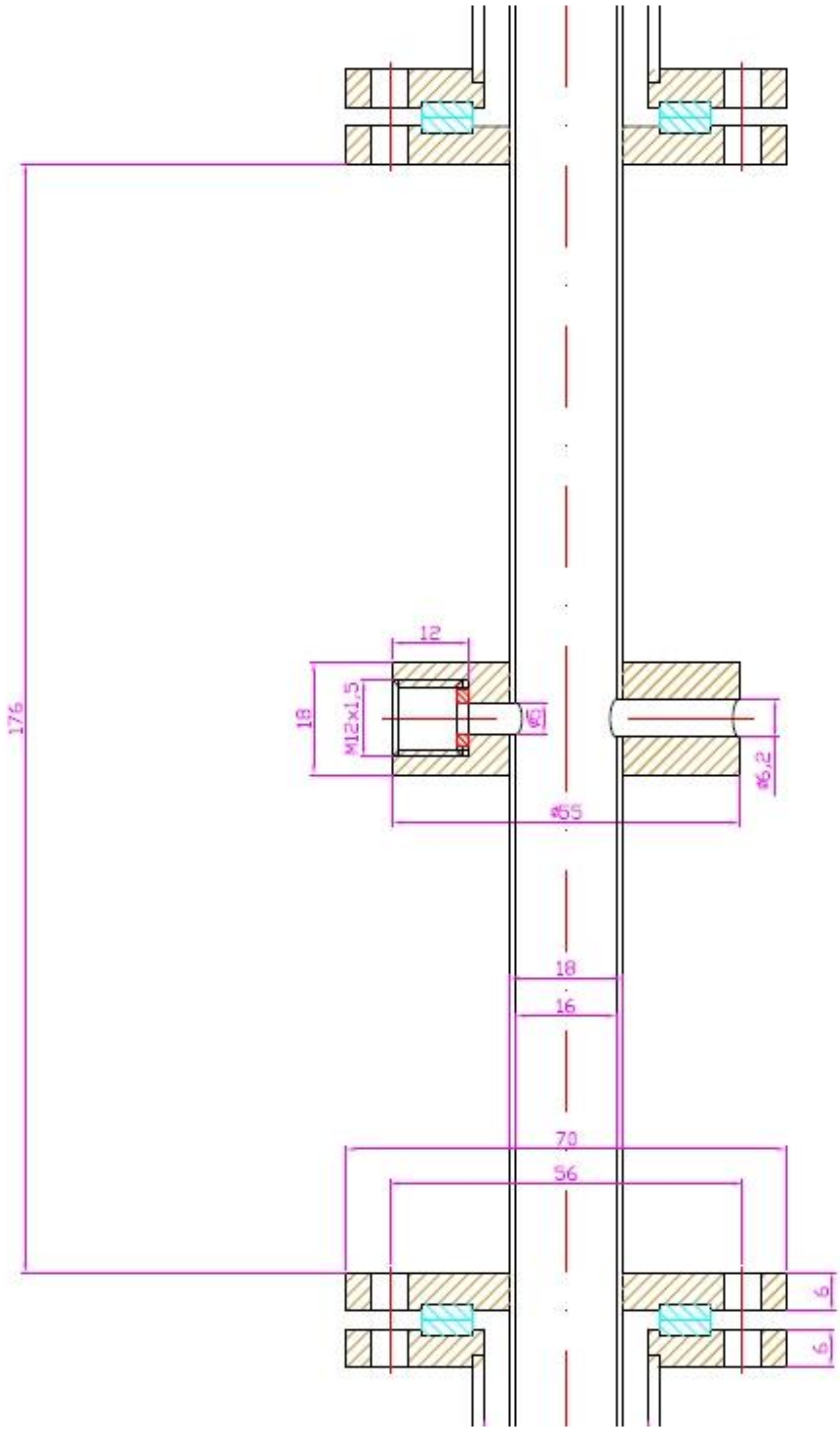


Figure 1. The tested copper heat pipe, 1769 mm long.

During the research, a thermographic analysis of the operation of a heat pipe type heat exchanger was also carried out [16]. For the purposes of thermographic analysis of the processes taking place inside the tested exchanger, a polycarbonate heat pipe was also built (Figure 4) [17].

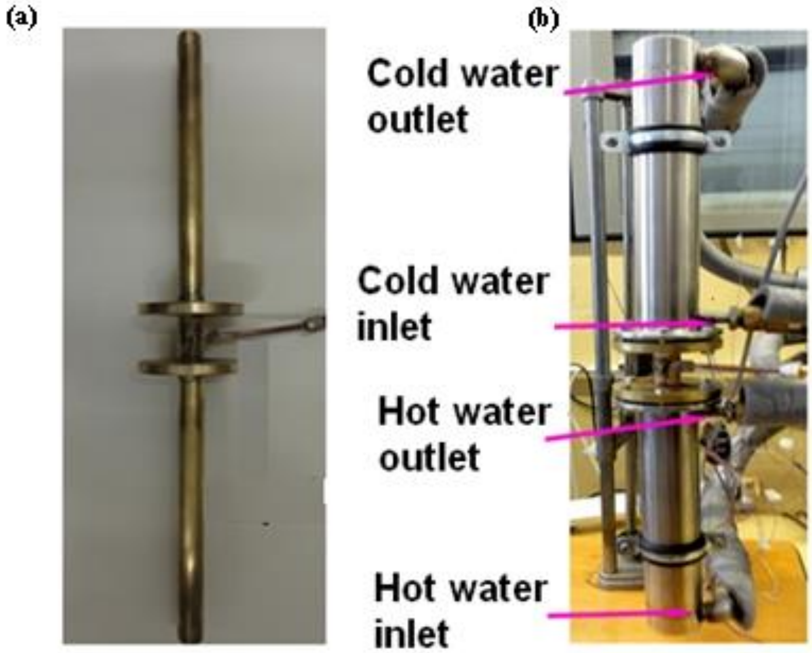


Figure 2. A brass heat pipe, 550 mm long and 22 mm in diameter [16]: (**a**) a brass heat pipe, (**b**) a heat pipe with heat exchangers and marked inlets and outlets for hot and cold water.

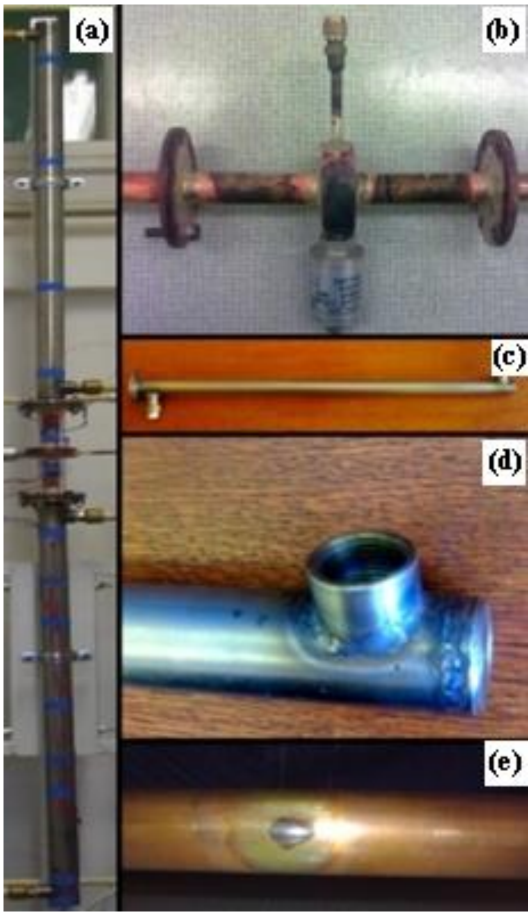


Figure 3. Tested heat pipe 1769 mm long: (**a**) complete heat pipe, (**b**) isothermal part of the heat pipe, (**c**) flow heat exchanger, (**d**) flow heat exchanger, (**e**) temperature measurement point.

Figure 4. A 1769 mm long heat pipe made of polycarbonate: (**a**) complete heat pipe, (**b**,**c**) isothermal part of the heat pipe, (**d**) heat pipe without flow exchangers.

A thermographic analysis was conducted on an actual heat pipe exchanger produced from polycarbonate, 1769 mm long, 18 mm outside diameter, and 1 mm wall thickness, i.e., with dimensions correlated to the copper heat pipe. Both the heat pipe and the flow exchangers were built from polycarbonate due to the transparency that allows observation of the processes taking place inside the heat pipe. The thermographic analysis was conducted using a thermovision camera [18,19]. The installation was also equipped with basic measuring equipment, which included thermocouples with meters, and digital flow meters with meters and rotameters [20]. A photo of the constructed test stand is shown in Figure 5.

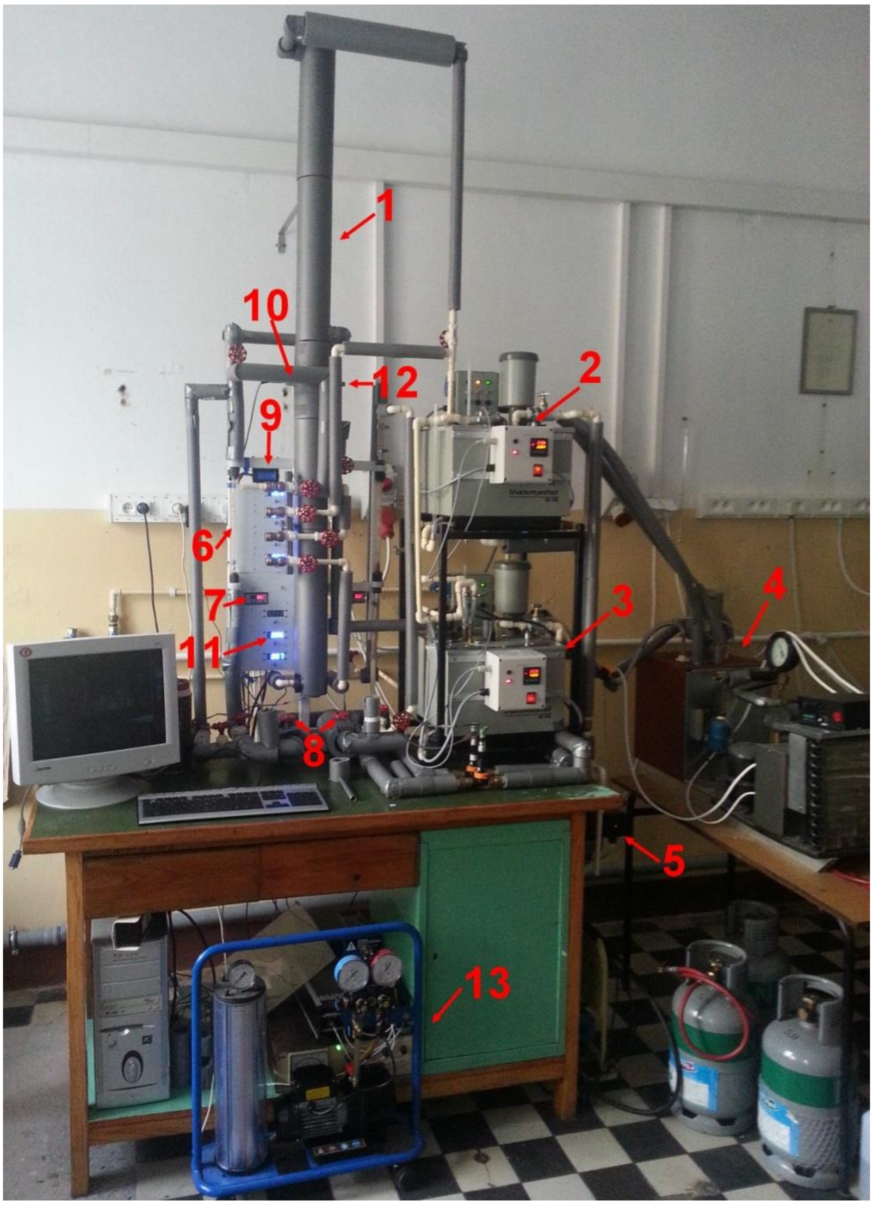


Figure 5. Measuring station where: 1—heat pipe with insulated heat exchangers 2—ultrathermostat–cooling water, 3—ultrathermostat–heating water, 4—chilled water aggregate, 5—circulation pumps, 6—rotameter, 7—gauge flow, 8—control valves, 9—pressure gauge, 10—pressure sensor, 11—millivoltmeter, 12—heat pipe filling valve, 13—filling station.

The heat pipe (1) was enclosed in the evaporating and condensing parts with heat exchangers for its heating and cooling. The evaporator was heated in the temperature range of 15–50 °C, and the condenser was cooled with water at 10 °C. To maintain constant temperature parameters in the heat exchangers, the station was equipped with two ultrathermostats equipped with a microprocessor-based temperature control system (2,3) cooperating with an ice water unit (4). Photos of the heat pipe with exchangers are shown in Figure 6.

To ensure the possibility of regulating and measuring the water flow through the heat exchangers, circulation pumps (5), rotameters (6), electronic flow meters (7), and shut-off and flow regulating valves (8) were installed. The possibility of measuring the pressure in the central part of the tube was provided by a pressure gauge (9) with a sensor (10). Installing thermocouples in conjunction with milli-voltmeters (11) made it possible to measure the thermoelectric force in selected places on the heat pipe, and thus to determine the temperatures in these places. Additionally, this system allowed us to determine the temperature of water flowing in and out of the heat exchangers. In the central part of the heat pipe there was a valve (12) for filling it.

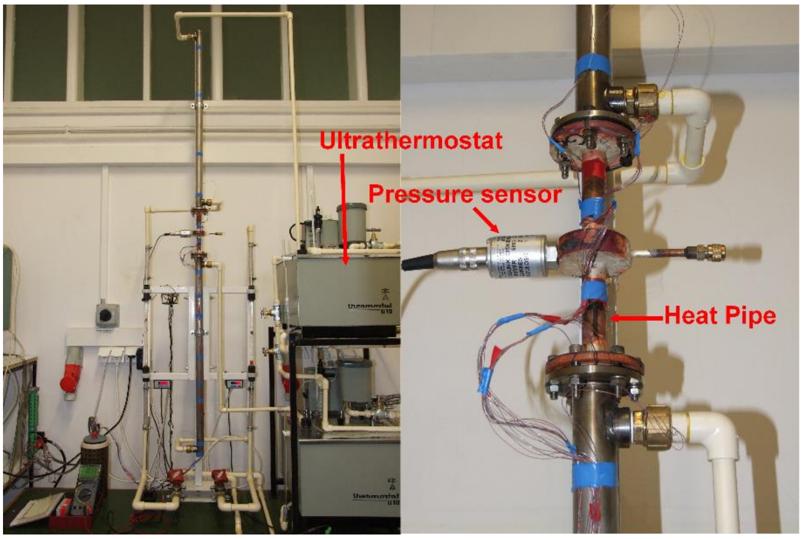


Figure 6. Test stand before insulation.

The main tests were carried out on a real heat exchanger made of copper, 1769 mm long, 18 mm diameter, and 1 mm thick. The heat pipe was filled with the refrigerant filling station (13), which first generated a vacuum in the pipe, and then filled the pipe with the measured amount of the working medium.

2.3. Measuring Points and Quantities of Interest

The heat transfer tests in the heat pipe type heat exchanger were carried out for various temperature ranges in terms of phase changes taking place inside the heat pipes. An analysis and evaluation of the efficiency of the heat pipe work was carried out for various working factors and the related phase changes, and for different temperatures on the side of the heat supply and constant temperature on the side of the heat receipt from the heat pipe [21]. The analysis and development of the research results consisted of assessing the effectiveness of this type of exchanger. The research methods concerning this heat exchanger consisted of examining a real exchanger and computer modeling of this exchanger to obtain a correlation between the simulation results and experimental studies. Equipment for thermal-flow measurements and computer equipment for numerical modeling of this heat exchanger was used for the tests, and for the preparation of the test results. Proper functioning of the heat pipe depends highly on the temperature inside it, the pressure of the working medium, and the temperature of the heating water at the heat exchanger inlet. Once the pressure is known, the temperature of evaporation of the working medium is read from thermophysical tables or with the use of a program, e.g., Solkane. The heat pipe can only operate when the evaporating temperature of the working medium inside is lower than the temperature of the water washing the evaporator part of the tube. In order for the tube to function, the working medium inside the heat tube should be heated by the thermal energy supplied. Conversely, when the temperature inside the heat tube is higher than the temperature of the water washing the vaporizer part of the tube, it cannot function properly.

The research plan included the following activities:

1. Emptying the heat pipe (creating a vacuum);
2. Filling the heat pipe with an appropriate amount of the working medium;
3. Determining the assumed temperatures in the ultrathermostats;
4. Pumping heating and cooling factors for the heat pipe;

5. Setting the uniform flow of heating and cooling factors in the heat pipe at 15 Lh;
6. Determining the resultant temperatures tz2, trg, trz, tg2;
7. Reading of measurement data.

The schematic diagram of the test stand is shown in Figure 7.

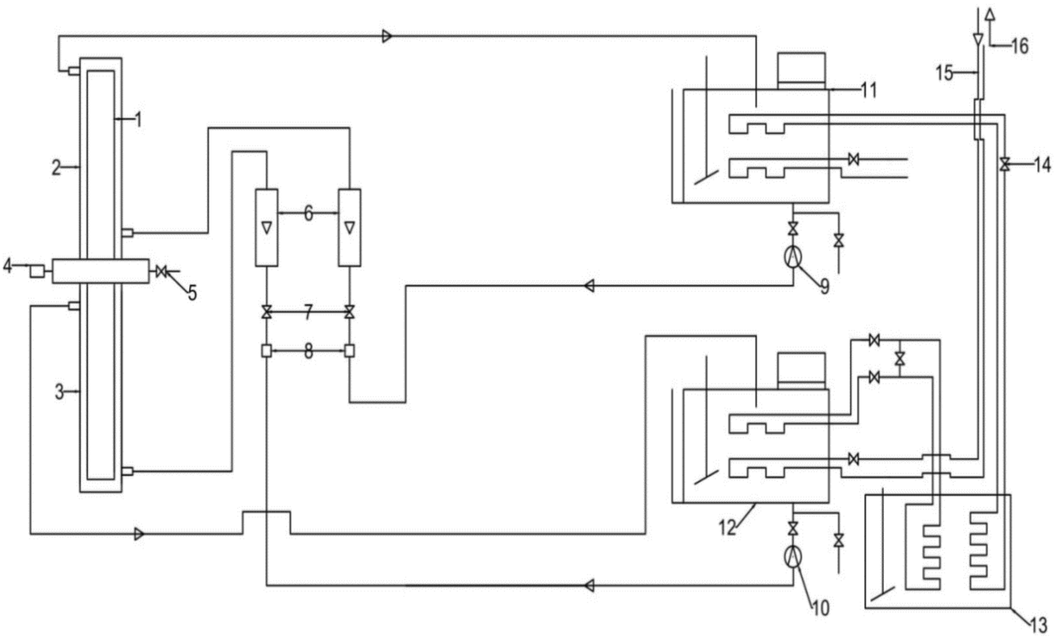


Figure 7. Schematic diagram of the test stand.

Table 1 shows fillings of the heat exchanger used to heat (3) and cool (2) the heat pipe. The diagram also shows the installation location of the pressure sensor (4) and the valve (5) through which the heat pipe is filled. The factor that heats the heat pipe is filled with an ultrathermostat (12), in which electric heaters and coils supplied from the ice water generator (13) and the water supply network (15) are responsible for maintaining the appropriate temperature. The heating medium from the ultrathermostat (12) is forced by the pump (10) through the flow meter (8) and the rotameter (6) to the flow heat exchanger pipe in the pipe (3), where it gives its heat to the heat pipe (1) and returns to the ultrathermostat (12). Valve (8) is responsible for regulating the amount of flowing medium. In the ultrathermostat (11) there is a cooling agent, the temperature of which is kept at a constant level by electric heaters and coils fed with chilled water from the chilled water generator (13) and water from the mains (15). Chilled water flow is regulated by a valve (14). The coolant from the ultrathermostat (11) is forced by the pump (9) through the flow meter (8) and the rotameter (6) to the flow heat exchanger pipe in the pipe (2) where it receives heat from the heat pipe (1) and then, when heated, it returns to the ultrathermostat (11).

Table 1. Comparison of the most advantageous fillings of all tested working fluids, vacuum, and air.

Tg1 [°C]	R134A 10%			R404A 10%			R407C 5%			Vacuum			Air		
	Qg [W]	Qz [W]	η [%]	Qg [W]	Qz [W]	η [%]	Qg [W]	Qz [W]	η [%]	Qg [W]	Qz [W]	η [%]	Qg [W]	Qz [W]	η [%]
15.00	30.76	27.73	90.00	24.17	20.03	83.00	0.44	0.00	0.00	0.00	0.00	0.00	0.22	0.00	0.00
20.00	39.48	35.65	90.00	37.29	31.70	85.00	0.88	0.00	0.00	0.00	0.00	0.00	0.66	0.00	0.00
25.00	63.48	57.88	91.00	59.11	51.72	88.00	13.13	1.76	13.00	0.00	0.00	0.00	1.09	0.00	0.00
30.00	83.05	78.33	94.00	78.68	71.85	91.00	34.93	11.23	32.00	0.00	0.00	0.00	2.18	0.00	0.00
35.00	104.60	99.04	95.00	109.00	100.70	92.00	54.47	34.12	63.00	1.09	0.00	0.00	4.36	0.44	10.00
40.00	134.90	127.60	95.00	135.10	124.20	92.00	84.84	61.64	73.00	2.18	0.22	10.00	6.53	0.88	13.00
45.00	160.70	151.80	94.00	154.50	140.70	91.00	128.10	97.97	76.00	3.26	0.44	13.00	7.61	1.32	17.00
50.00	199.30	184.80	93.00	176.00	150.80	86.00	162.50	126.60	78.00	4.34	0.66	15.00	8.69	1.98	23.00

Figure 8 shows a diagram of the installation of the heat pipe on the test stand with the temperature measurement points.

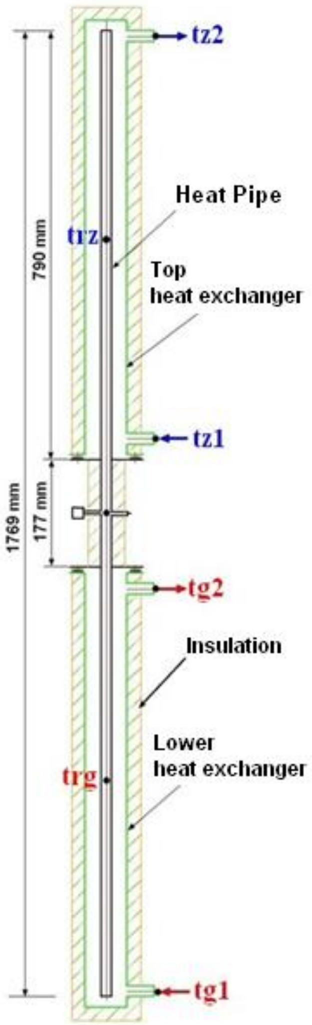


Figure 8. Scheme of the installation of the heat pipe at the measuring stand with temperature measurement points, where tz1—cold water temperature at the inlet to the cooling exchanger, tz2—cold water temperature at the outlet from the cooling exchanger, tg1—hot water temperature at the entrance to the heat exchanger, tg2—hot water temperature at the outlet of the heating exchanger, trz—heat pipe wall temperature in the center of the condensing section, trg—heat pipe wall temperature in the center of the evaporating section.

2.4. Test Procedure (Used for the Repetitiveness and Archivibility of the Document, Number of Tests Performer)

The heat pipe tests consisted of measuring the temperatures at the inlet and outlet of heating and cooling water from pipe-in-pipe heat exchangers, measuring the pressure inside the pipe, determining the water flow in the exchangers, and determining the temperatures at measuring points along the height of the heat pipe. The temperature determined of heating and cooling water in the exchangers was obtained with the use of ultrathermostats. The temperature of the cooling water at the inlet to the exchanger in the condenser part was constant at about 10 °C. The value of the heating water temperature at the inlet to the heat exchanger in the evaporator part for subsequent measurements ranged from 15 to 50 °C and was changed in increments of 5 °C. The flow of heating and cooling water flowing around the heat pipe was about 15 L/h. After stabilizing the heat exchange, the water temperatures at the inlet and outlet of the exchangers and along the height of the heat pipe were read. Measurements were made by means of thermocouples installed on the measuring stand, using a millivoltmeter and a temperature recorder. The values obtained in millivolts were converted to temperature in degrees Celsius using the thermocouple constant. The wall temperatures in the middle of the evaporator and condenser were measured in the heat pipe tested. The heat pipe was tested with the use of three refrigerants, namely, R134A, R404A, and R407C. For each of these factors, the heat tube was filled by volume for subsequent measurements at amounts of 2.5%, 5%, 10%, 20%, 30%, and 40% of the total volume of the heat tube. Knowing the total volume of the heat pipe and the density of a given medium, its amount expressed in grams was determined, which corresponds to its filling in the volumes indicated above. The density of factors at a given temperature was determined using the tables of thermophysical properties.

2.5. Tests of a Copper (Cu) Heat Pipe 1769 mm Long

During the test of a 1769 mm long copper heat pipe, the temperatures at the inlet and outlet of hot and cold water from heat exchangers heating and cooling the heat pipe were measured. The pressures inside the tube and the temperatures of the wall of the heat tube were also measured at measurement points halfway up the condenser and evaporator of the heat tube.

The temperature of cold water at the entrance to the heat exchanger in the condenser part was about 10 °C. In the evaporator section the hot water temperature at the entrance to the heat exchanger was between 15 and 50 °C and it was changed in increments of 5 °C for each measurement. The flow rates of cold and hot water flowing around the heat pipe were about 15 L/h.

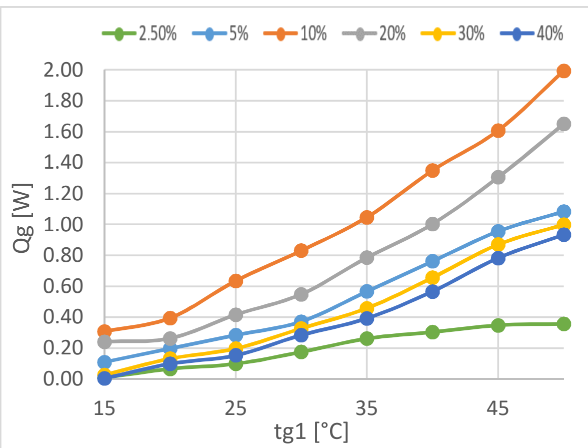
The tests were performed for the heat pipe without any working medium with a vacuum and then with air inside the heat pipe and with the use of refrigerants: R134A, R404A, and R407C. The heat tube was filled by volume for subsequent measurements at amounts of 2.5%, 5%, 10%, 20%, 30%, and 40% of its entire volume.

3. Results

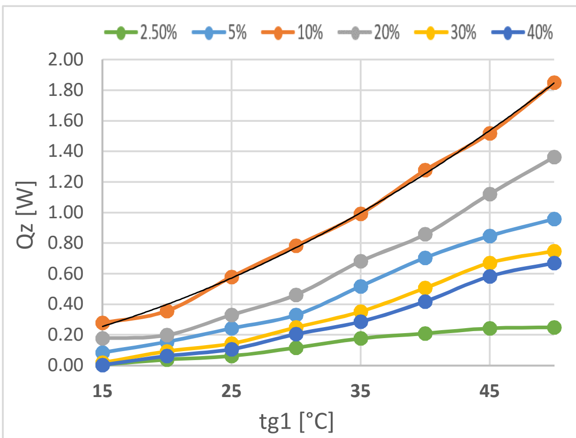
3.1. Heat Copper Tube Exchanger, 1769 mm Long, 18 mm Outside Diameter and 1 mm Wall Thickness

3.1.1. Comparative Analysis of Test Results for Various Charges of the R134A Working Medium

Figures 9 and 10 show diagrams for different amounts of the R134A working medium filling the heat pipe, comparing the heat flux values collected and given off by the heat pipe, and a diagram comparing the heat pipe efficiency.



(**a**)



(**b**)

Figure 9. Comparison diagram of heat fluxes (**a**) absorbed by the heat pipe and (**b**) given off by the heat pipe for all charges with the R134A working medium.

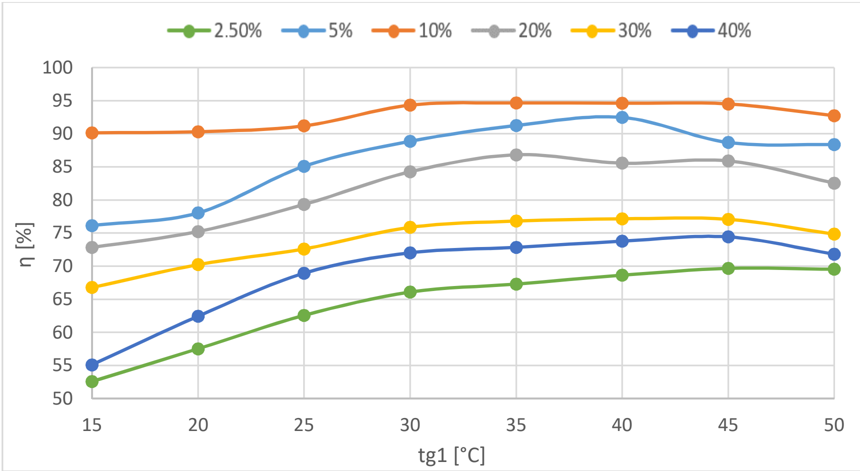


Figure 10. Comparison diagram of the heat pipe efficiency for all charges with the R134A refrigerant.

The values of the heat flows absorbed by the evaporator are always higher than those taken off by the condenser. This indicates that more heat is taken by the refrigerant than given up. This difference results mainly from heat losses to the environment, such as through the isothermal component, in which the pressure sensor and the valve for filling the heat pipe with the working fluid are located.

The heat pipe works with satisfactory efficiency at 5%, 10%, and 20% of its volume filling in the temperature between tg1 15 °C and 50 °C; whereas at 2.5% filling, 30% and 40% of the volume of the heat pipe works with satisfactory efficiency in the temperature range tg1 = 25–50 °C. In all fillings, the pressure in the heat pipe rises linearly with the rise of the temperature at the entrance to the lower heat exchanger. The highest value of the heat flux taken by the evaporator and given off by the condenser occurs when the heat pipe is filled in 10% of its volume with the R134A refrigerant.

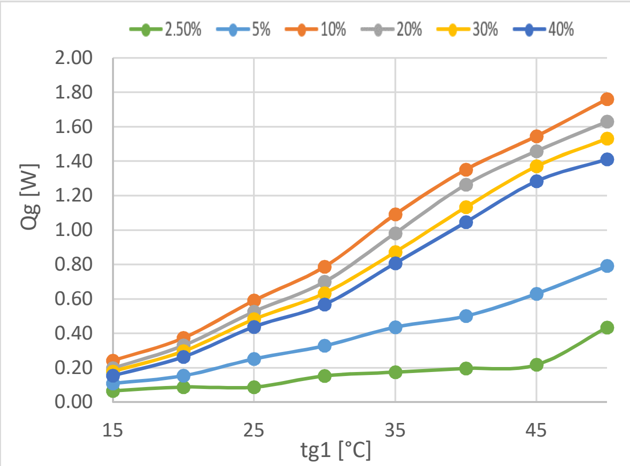
The highest efficiency was achieved for 10% filling in the heat pipe operating range at 25–50 °C, amounting to approx. 95%. Heat transfer efficiency was also high in the other fillings within the scope of the heat pipe′s operation at temperatures of 25–50 °C. However, in the remaining temperature range, the efficiency of the heat pipe decreased. The heat transfer efficiency of the heat pipe was found for the R134A refrigerant at a range of 53–95%. Due to the value of the heat flux exchanged and heat transfer efficiency in the entire temperature range, the highest efficient operation of the heat pipe occurs when it is filled with the R134A working medium at 10%.

3.1.2. Comparative Analysis of Test Results for Various Charges with the R404A Working Medium

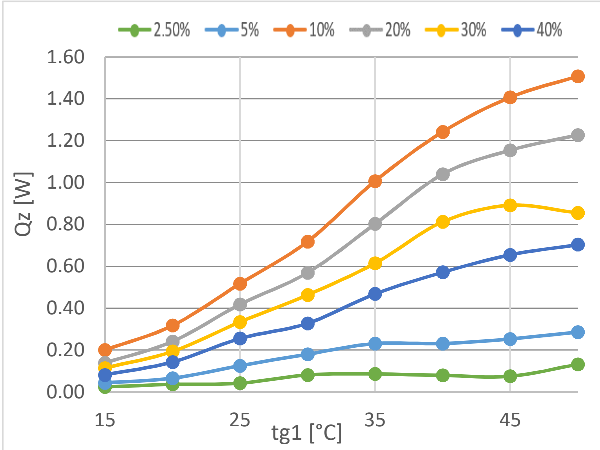
Figures 11 and 12 show diagrams for different amounts of the R404A working medium filling the heat pipe, comparing the values of heat fluxes collected and given off by the heat pipe, and a graph comparing the efficiency of the heat pipe.

The values of the heat flows obtained by the evaporator are in every case higher than those given out by the condenser. This proves that a higher amount of heat is taken by the cooling fluid than given up. These different values can be observed mainly from heat losses to the environment, for example, through the isothermal component, in which the pressure sensor and the valve for filling the heat pipe with the working medium are located.

The heat pipe works with a satisfactory efficiency at 10%, 20%, 30%, and 40% of its capacity filling in the temperature limit between tg1 20 and 50 °C, whereas, when filling 2.5% and 5% of its volume, the heat pipe works with satisfactory efficiency in the temperature range tg1 from 45 to 50 °C. In all fillings, as the temperature at the entrance to the lower heat exchanger rises, the pressure in the heat pipe rises approximately linearly. The highest value of the heat flux taken by the evaporator and given off by the condenser occurs when the heat pipe is filled in 10% of its volume with the R404A refrigerant.



(**a**)



(**b**)

Figure 11. Comparison diagram of heat flux (**a**) absorbed by the heat pipe and (**b**) given off by the heat pipe for all fillings with the R404A working medium.

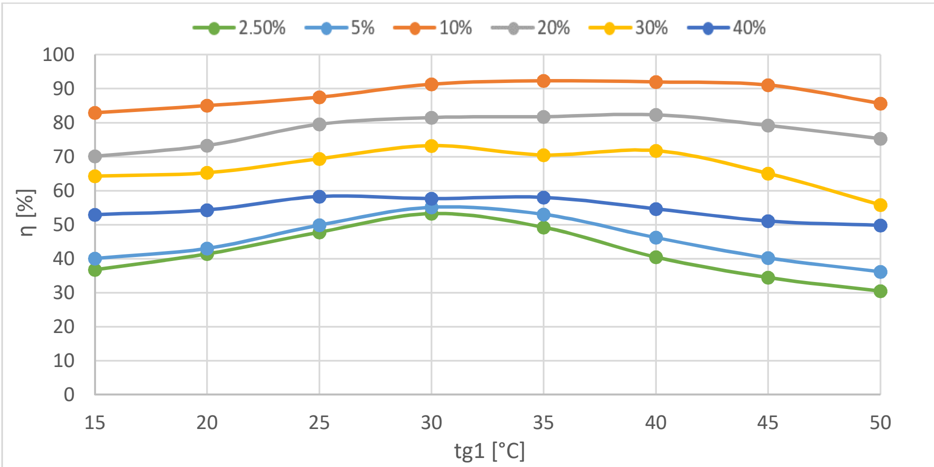


Figure 12. Comparison chart of heat pipe efficiency for all charges with the R404A refrigerant.

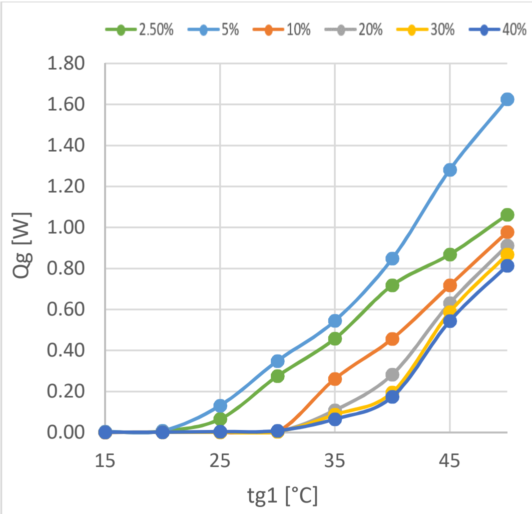
For 10% filling in the heat pipe operating range at 25–45 °C, the highest efficiency was achieved, amounting to approx. 90%. For the remaining fillings in the heat pipe operating range at 25–45 °C, its efficiency was also the highest. However, in the remaining temperature range, the efficiency of the heat pipe decreased. The efficiency of the heat pipe was maintained for the R404A refrigerant in the range of 30–92%.

Due to the amount of transferred heat flux and efficiency in the entire temperature range, the most effective operation of the heat pipe occurs when it is filled with the R404A working medium. Analysis of measurement error and measurement uncertainty wasnplaced in Appendix A.

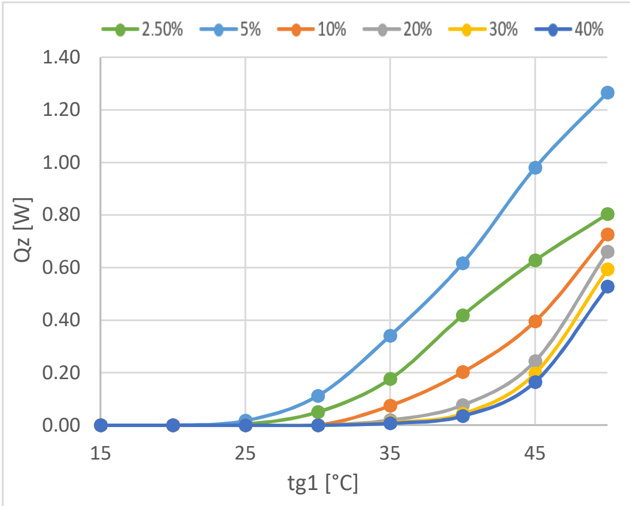
3.1.3. Comparative Analysis of the Test Results for Various Charges with the R407 C Working Medium

Figures 13 and 14 show diagrams for various amounts of the R407C working medium filling the heat pipe, comparing the values of heat fluxes consumed by the heat pipe and a graph comparing the efficiency of the heat pipe.

The values of the heat flows taken by the evaporator are always greater than those given off by the condenser, which proves that more heat is taken by the refrigerant than given up. This difference results mainly from heat losses to the environment, for example, through the isothermal component, in which the pressure sensor and the valve for filling the heat pipe with the working medium are located.



(**a**)



(**b**)

Figure 13. Comparison diagram of heat flux (**a**) absorbed by the heat pipe and (**b**) given off by the heat pipe for all fillings with the R407C working medium.

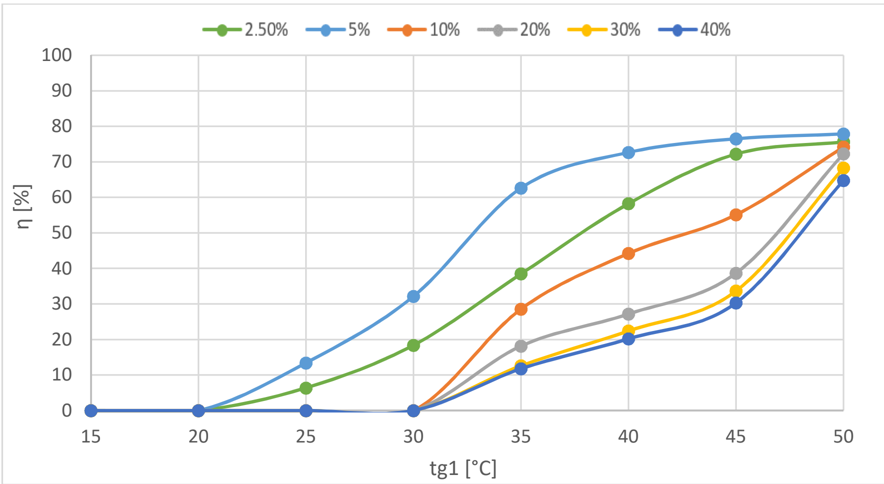


Figure 14. Comparison diagram of the heat pipe efficiency for all charges with the R407C refrigerant.

The heat pipe works with a satisfactory efficiency at 2.5%, 5%, and 10% of its volume filling in the tg1 temperature range from 35 to 50 °C, whereas at 20% filling, 30% and 40% of the volume of the heat pipe works with satisfactory efficiency in the temperature range tg1 from 45 to 50 °C. In all fillings, the pressure in the heat pipe increases approximately linearly as the temperature at the entrance to the lower heat exchanger increases. The highest value of the heat flux, taken by the evaporator and given off by the condenser, occurs when the heat pipe is filled to 5% of its volume with the R407C refrigerant. For all fillings with the working medium R407C, the operation of the heat pipe started at tg = 30–35 °C at the inlet to the heat exchanger heating the heat pipe. Up to this temperature, the tg temperature was lower than the R407C evaporation temperature for a given pressure and the heat pipe did not transfer heat.

For the 5% filling in the heat pipe's operating range at 35–50 °C, the highest efficiency was achieved, amounting to approx. 70%. For the remaining fillings in the heat pipe working range, its efficiency was much lower and it increased only at tg = 50 °C. The efficiency of the heat pipe was maintained for the R407C refrigerant in the range of 0–78%.

Due to the amount of heat flux transferred and efficiency in the entire temperature range, the most effective operation of the heat pipe occurs when it is filled with the

R407C working medium. Analysis of measurement error and measurement uncertainty wasnplaced in Appendix A.

3.1.4. Analysis of the Most Favorable Refrigerants: R134A, R404A, R407C

Table 1 shows a comparison of the most favorable fillings for all media tested, for vacuum and air. Figures 15 and 16 show comparative diagrams of heat flux collected and given off by the heat pipe and the efficiency of the heat pipe for the best fillings of all working media tested, and for vacuum and air. Analysis of measurement error and measurement uncertainty wasnplaced in Appendix A.

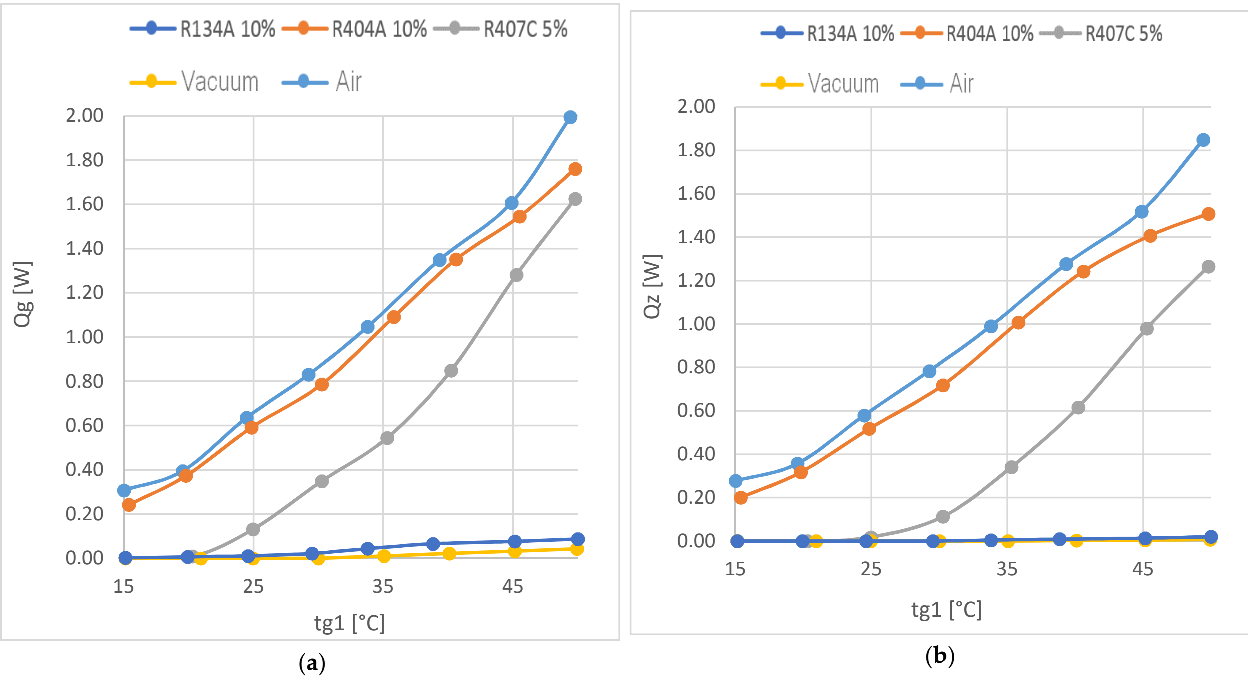


Figure 15. Comparison chart of heat flux (**a**) absorbed by the heat pipe and (**b**) given off by the heat pipe for the most favorable filling of all tested working media, and for vacuum and air.

3.1.5. Thermographic Analysis of the Heat Pipe during Operation

A thermographic analysis of the heat pipe in operation was conducted on a heat pipe exchanger made of polycarbonate, 1769 mm long, 18 mm outside diameter, and 1 mm wall thickness, i.e., dimensions related to the copper heat pipe. The heat pipe and the heat flow exchangers were constructed of polycarbonate due to the transparency that allows observation of the processes taking place in the heat pipe. The thermographic analysis was conducted with a thermal imaging camera. Figure 17 represents the thermographic analysis done with a connected polycarbonate heat pipe, and the following figures show thermal images of the heat pipe in operation. The heat pipe was analyzed with the R134A refrigerant in the amount of 10% of the total volume. The temperature of the hot water inlet feeding the evaporator section was tg1 = 50 °C and the colder water entrance feeding the condenser section was tz1 = 10 °C. The heat pipes consist of three distinct areas: the evaporator, which is the heat supply area; the condenser, which is the heat receiving area; and the adiabatic area between them. When the heat pipe is in operation, there is a closed, two-phase cycle with condensation and vaporization of the working fluid remaining in saturated conditions when the temperature is between the triple point and the critical state. The heat applied to the evaporator area causes the working fluid in the evaporator to evaporate. The high temperature and correspondingly high pressure in this area create a steam flow towards the opposite, cooler end of the vessel, where the steam then condenses, giving up its latent heat of vaporization. They then transport the liquid back to the evaporator by gravity in the gravity heat tubes or by capillary forces in the porous structure.

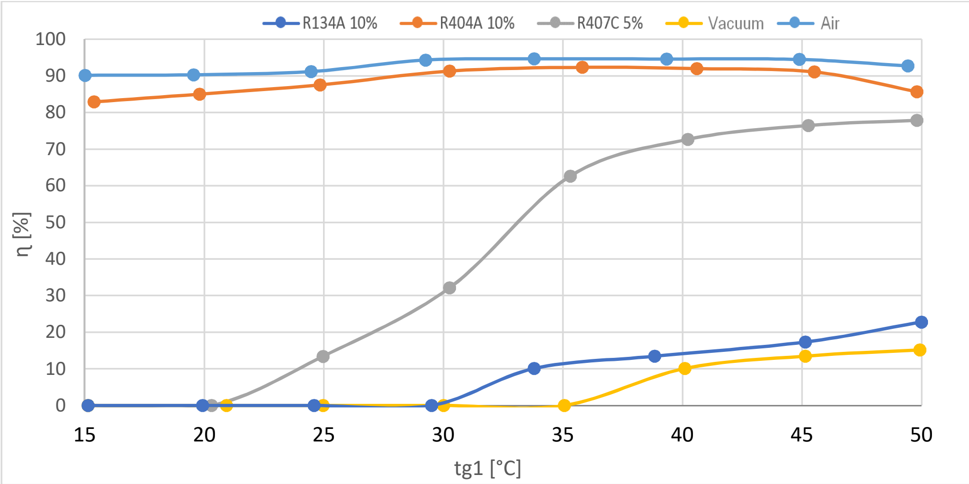


Figure 16. Comparison chart of the heat pipe efficiency for the most favorable fillings of all tested working media, and for vacuum and air.

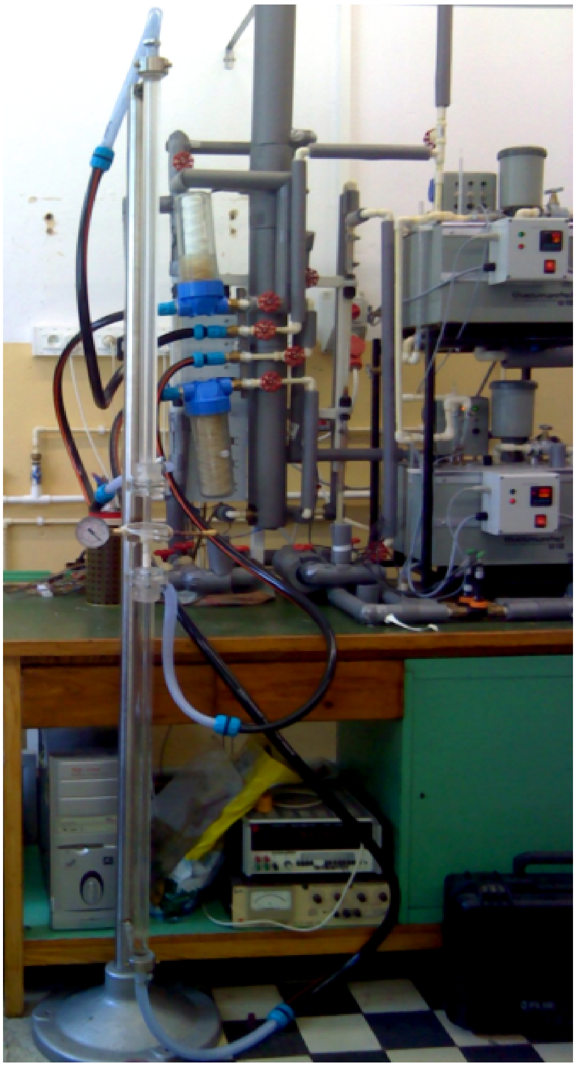
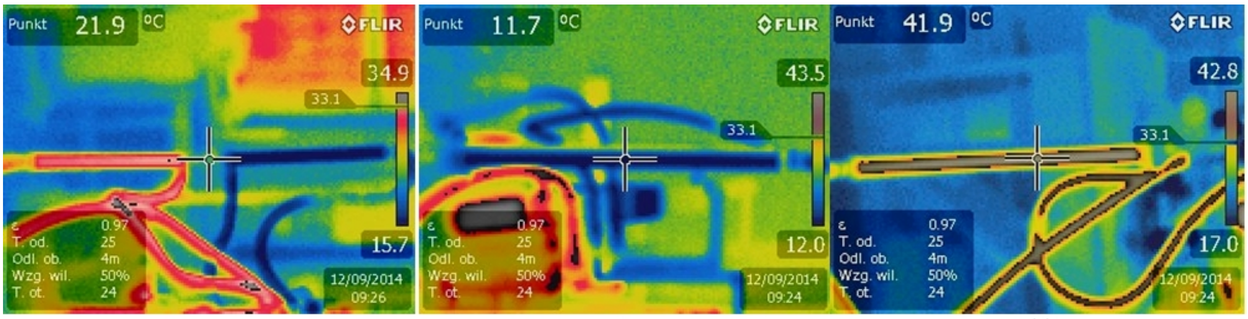


Figure 17. Measuring stand with connected polycarbonate heat pipe.

The mentioned thermographic analysis of the heat pipe during its operation proved the operation of the heat pipe at the analyzed temperature limit (Figure 18). The fact that both the heat pipe and its supply exchangers were made of polycarbonate made it possible to compare the moment of starting the process of evaporation of the working fluid and its correlation with other similar research. The high temperature and correspondingly high pressure in this area create a steam flow towards the opposite, cooler end of the vessel, where the steam then condenses, giving up its latent heat of vaporization. They then transport the liquid back to the evaporator by gravity in the gravity heat tubes or by capillary forces in the porous structure. Author did a photos of the evaporation process inside the connected polycarbonate heat pipe Figures 19 and 20.



(a) (b) (c)

Figure 18. Heat pipe during operation: (**a**) heat pipe, (**b**) evaporator section, (**c**) condenser section.

Figure 19. Photo of heat pipe during operation showing evaporation process.

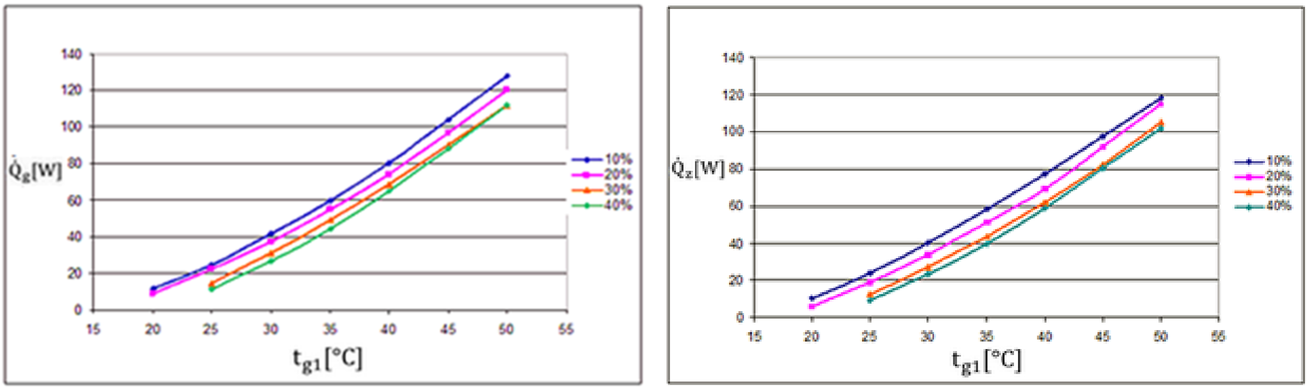
Figure 20. Photo of heat pipe during operation showing evaporation process.

3.2. Tests of a Brass (CuZn39Pb3) Heat Pipe 550 mm Long

The subject of the second part of the research was heat pipes made of brass, 550 mm long, 22 mm outside diameter, and 1 mm wall thickness. During the brass heat pipe test, the temperatures at the inlet and outlet of hot and cold water from heat exchangers were measured. The pressures inside the tube were also measured and the temperatures at the measuring points along the height of the heat tube were determined. The temperature of colder water at the entrance to the heat exchanger in the condenser part was about 10 °C, whereas the hot water temperature at the heat exchanger in the evaporator section was within a range of 15–50 °C every 5 °C for each subsequent measurement. The value of the cold and hot water stream flowing around the tube was about 15 L/h. The heat pipe tests were performed with the use of the refrigerants R407C and R410A. The heat tube was filled by volume for subsequent measurements at amounts of 10%, 20%, 30%, and 40% of the entire volume. The same heat pipe was tested for R134A and R404A refrigerants. Analysis of measurement error and measurement uncertainty wasnplaced in Appendix A.

3.2.1. Comparative Analysis of Test Results for Various Charges of the R134A Working Medium

Figures 21 and 22 show diagrams for various amounts of the R134A working medium filling a heat pipe made of 550 mm long brass, looking at the values of heat flux collected and given off by the heat pipe, and a diagram comparing the heat transfer efficiency of the tested heat pipe [22].



(**a**) (**b**)

Figure 21. Diagram of heat flux (**a**) collected by the evaporator and (**b**) given by the condenser inside heat pipe for various charges of the R134a refrigerant depending on the temperature of the water washing the evaporator [15].

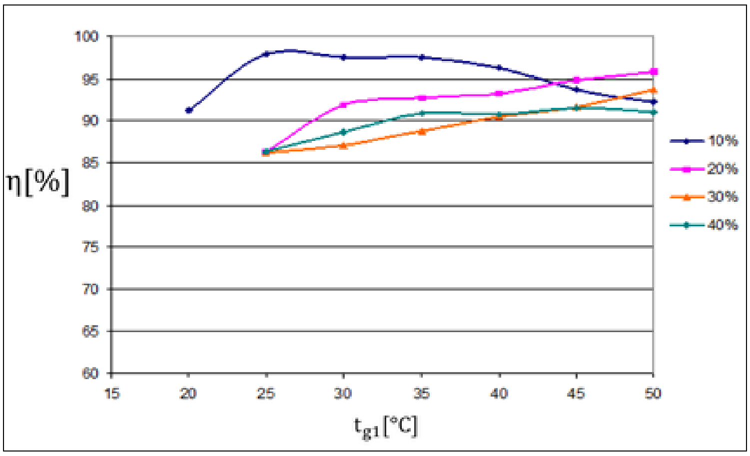
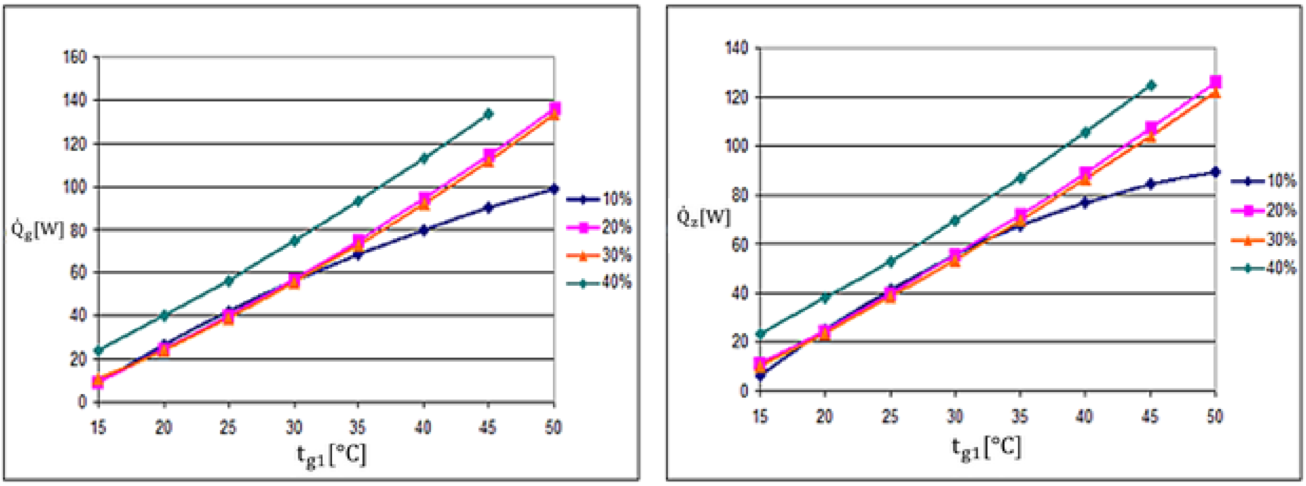


Figure 22. Diagram of the heat pipe efficiency for various charges with R134a depending on the temperature of the water washing the evaporator [15].

When filling the heat pipe with R134A refrigerant, the best choice would be to fill it to 10% of its volume due to the highest values of heat fluxes taken by the evaporator and given off by the condenser (max 130 W). The efficiency values of the heat pipe for this filling are also high and reach a level of 92%–98%, whereas the operation of the pipe begins at a temperature of 20 °C, and in the remaining fillings the heat pipe operates at a temperature of 25 °C.

3.2.2. Comparative Analysis of Test Results for Various Charges with the R404A Working Medium

Figures 23 and 24 show diagrams for different amounts of the R404A working medium filling a heat pipe made of 550 mm long brass, showing the values of heat flux collected and returned by the heat pipe, and a graph comparing the efficiency of the tested heat pipe.



(a) (b)

Figure 23. Diagram of heat flux (**a**) collected by the evaporator of the heat pipe and (**b**) given by the heat pipe condenser for different charges of the R404A refrigerant depending on the temperature of the water washing the evaporator [15].

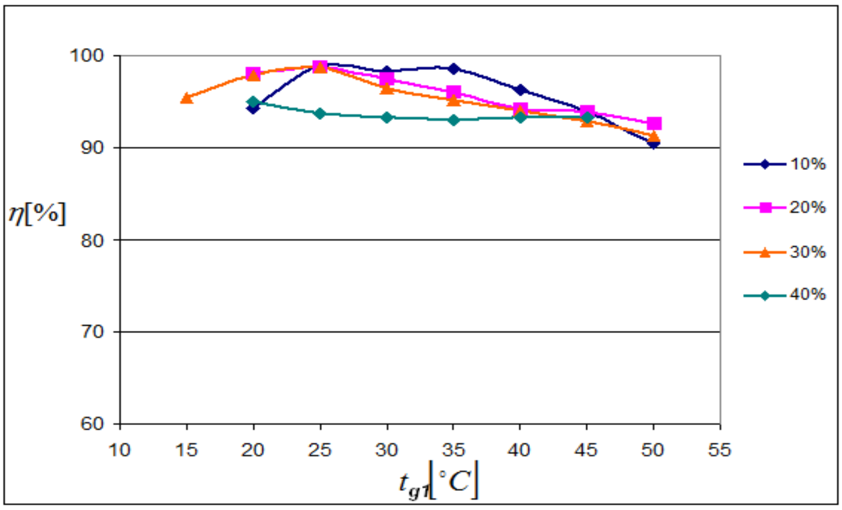


Figure 24. Efficiency diagram of the heat pipe for various charges with R404A depending on the temperature of the water washing the evaporator [15].

A heat pipe filled with R404A refrigerant works best for charges ranging from 20% to 40% of the tube′s volume, because the values of the transferred heat fluxes are the highest, reaching 135 W. The heat pipe for all charges achieves high efficiency values ranging from 90% to 98%.

3.2.3. Comparative Analysis of Test Results for Various Fillings with the R407C Working Medium

Figures 25 and 26 show diagrams for various amounts of the R407C working medium filling a heat pipe made of 550 mm long brass, showing the values of heat flux collected and given off by the heat pipe, and a graph showing the efficiency of the designed heat pipe [22].

With the use of the R407C refrigerant, it was found that it is most advantageous to fill the tube to 20%, 30%, and 40% of its volume because the values of heat flux absorbed by the evaporator and given off by the condenser reach the highest values (Q ġ = ~90 W). By comparison, taking into account economic and efficiency issues, it is more profitable to fill the heat pipe up to 30% of its volume with the R407C factor [22].

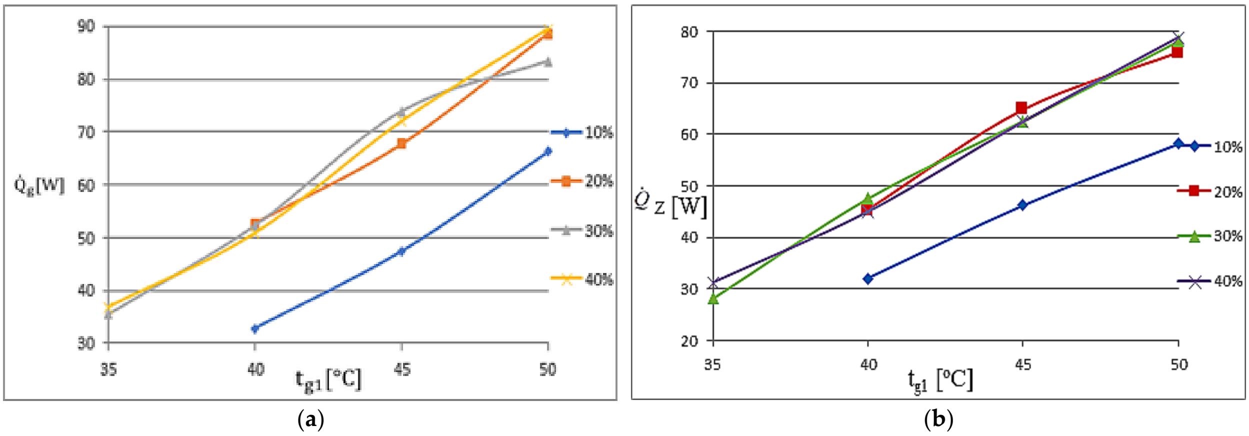


Figure 25. Diagram of heat flux (**a**) collected by the evaporator and (**b**) given by the condenser for various charges with R407C depending on the water temperature at the inlet to the heat exchanger in the evaporator section [22].

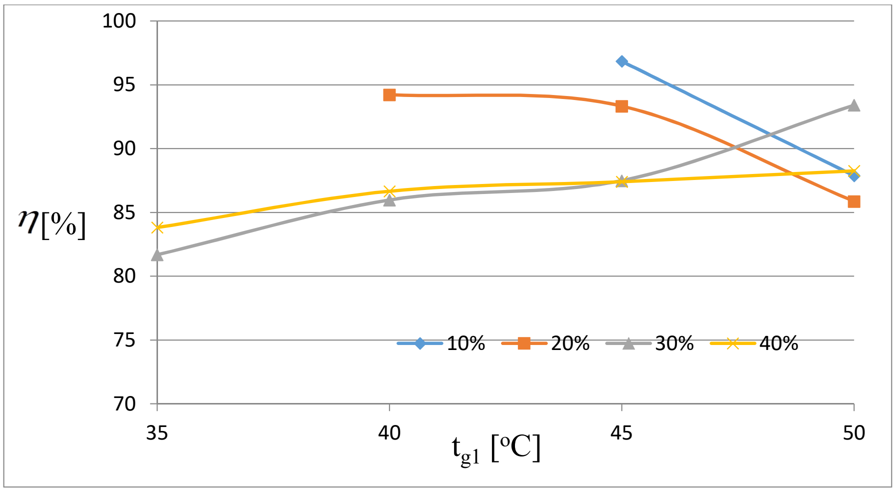


Figure 26. Heat pipe efficiency chart for various R407C charges depending on the water temperature at the inlet to the heat exchanger in the evaporator section [22].

3.2.4. Comparative Analysis of Test Results for Various Charges with the R410A Working Medium

Figures 27 and 28 show diagrams for different amounts of the R410A working medium filling a heat pipe made of 550 mm long brass, showing the values of heat flux collected and given off by the heat pipe, and a diagram showing the efficiency of the tested heat pipe [22].

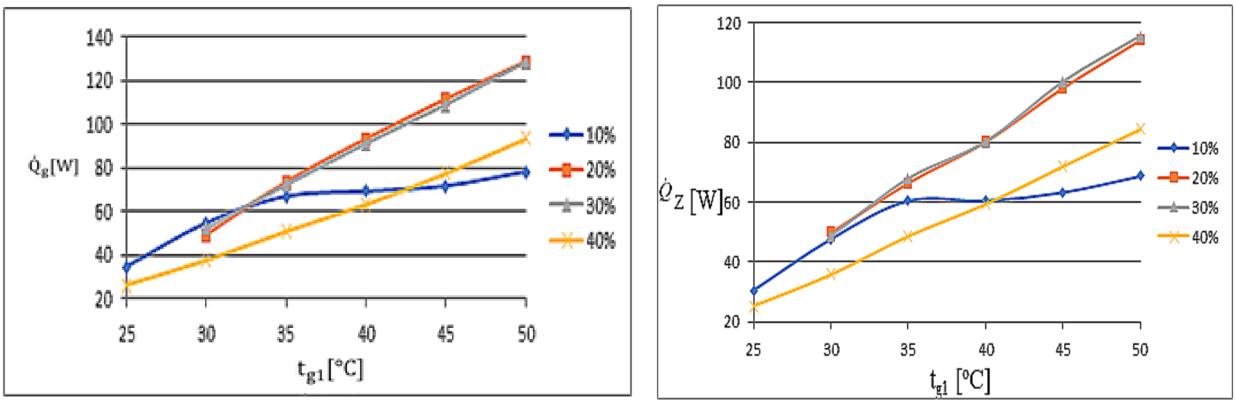
In that case of filling the heat pipe with R410A refrigerant, the values of heat flux collected by the evaporator and given off in the condensation section occurring at 20% and 30% fillings were the highest (Q ċmax = ~128 W), and much lower values were achieved with 10% and 40% fillings of the tube.

The highest value of the heat pipe efficiency (~95%) was obtained during 40% filling, within a temperature range of 25–50 °C, whereas the lowest value in the same temperature range was obtained for 10% filling. On the basis of the above graphs, it was noticed that the heat pipe was found to be the most efficient for 20% and 30% R410A refrigerant. For efficiency reasons, it is best to choose 20% of the total volume of the heat tube with R410A. The least effective solution is to fill the tube to 40% of its volume.

3.3. Comparative Analysis of the Most Advantageous Fillings of the Heat Pipe 550 mm Long Compared to the Heat Pipe 1769 mm Long

Of the four refrigerants tested (R134A, R404A, R407C, and R410A), R404A is the best refrigerant to fill the brass heat pipe with a length of 550 mm and a diameter of 22 mm.

The value of the heat flux taken in the evaporation section and given off by the condenser, in this case for 20% of the volume filled, reaches a value of ~135 W (similar to the R410A charge), but the operating range of the heat pipe is greater for the R404A refrigerant and amounts to 20–50 °C compared to R410A, where it is in a range of 25–50 °C (Table 2).



(**a**) (**b**)

Figure 27. Diagram of heat flux (**a**) collected by the evaporator and (**b**) given by the condenser for various refrigerant charges R410A depending on the inlet water temperature of the heat exchanger in the evaporator part [22].

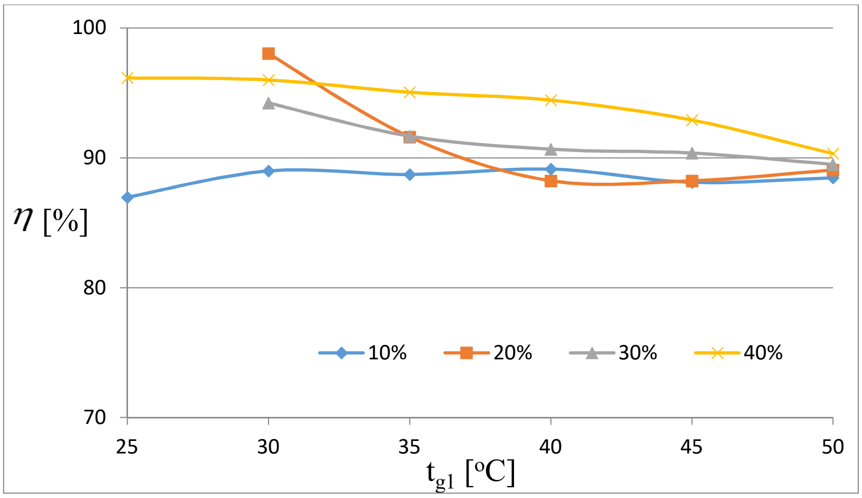


Figure 28. Heat pipe efficiency diagram for various R410A refrigerant charges depending on the water temperature at the heat exchanger inlet in the evaporator section [22].

Table 2. Comparison of the most advantageous fillings of a heat pipe 550 mm long and 22 mm in diameter made of brass, compared to a heat pipe 1769 mm long and 18 mm in diameter made of copper.

T_{g1} [°C]	Copper Tube. 1769 mm Long. 18 mm in Diameter. Filled with 10% of the Total Volume of R134A			A Brass Heat Pipe. 550 mm Long. 22 mm in Diameter. Filled with 20% of the Total Volume of R404A		
	Q_g [W]	Q_z [W]	η [%]	Q_g [W]	Q_z [W]	η [%]
15.00	30.76	27.73	90.15	7.00	6.48	0.00
20.00	39.48	35.65	90.30	21.00	20.23	0.00
25.00	63.48	57.88	91.19	39.28	38.73	98.59
30.00	83.05	78.33	94.32	58.26	55.03	94.23
35.00	104.64	99.04	94.64	75.25	75.02	99.70
40.00	134.92	127.65	94.61	91.84	88.30	96.68
45.00	160.70	151.86	94.50	117.90	106.57	90.39
50.00	199.38	184.87	92.72	134.89	125.73	93.21

Based on the results of this analysis of the of experimental studies, it can be concluded that the most effective working medium of a heat pipe made of copper with a length of 1790 mm and a diameter of 18 mm in the tested temperature range is the working medium R134A with a 10% filling of the heat pipe. The maximum heat flux absorbed by the heat pipe reaches a value of 199.3 W, whereas the maximum heat flux emitted by the heat pipe reaches a value of 184.8 W. In addition to the largest heat fluxes transferred through the heat pipe, this factor in the said filling achieved the best heat exchange efficiency reaching 95%.

4. Discussion

Both numerical and experimental studies of heat pipes have been carried out by numerous scientists. Heat pipes with similar geometry and similar diameters were tested by Chen and co-authors [23]. In their research, they built a theoretical model of a heat pipe, the results of which are similar to the experimental results presented in this publication. Gao et al. [24] investigated the use of heat pipes for the decomposition of sulfuric acid based on heat transfer surfaces and catalytic volume, and obtained a near 7% acceleration of the decomposition process due to the intense heat exchange taking place in the heat pipes. Ji-Qiang Li and others [25] investigated the power and efficiency of heat exchange in a Stirling engine where there are high technological requirements in terms of heat exchangers. Taking into account the range of temperatures of the applicability of heat pipes and the results of the efficiency of heat transfer through them, it is possible to suggest the use of heat pipes for their tests. By comparison, Song et al. [26] investigated heat pipes in the form of concentric annular tubular heat sinks, obtaining even higher efficiency and values of the transmitted heat flux than those tested in this study. The heat exchangers analyzed, however, have many limitations, which do not allow them to be used in such a wide range. In turn, Xiang [27] and co-authors experimentally tested geometrically similar heat pipes, acting in an anti-gravity manner and thus equipped with a conical chamber. The heat pipes tested were characterized by high efficiency, as for anti-gravity heat pipes, but are still less efficient than gravity tubes and have greater limitations resulting from the need to use a wick.

4.1. Discussion on the Choice of the Working Medium and the Selection of Its Quantity

Four refrigerants were used in the current research: R134A, R404A, R407C, and R410A. Based on the analysis of experimental studies, it can be concluded that the most effective working medium of a heat pipe made of copper with a length of 1769 mm in the tested temperature range is the R134A refrigerant with a 10% filling of the heat pipe. With an assumed water flow of 15 L/h for the measured temperature differences, the heat flux collected by the heat pipe reaches a value of 199.3 W, and the heat flux emitted by the heat pipe reaches a value of 184.8 W. In the case mentioned, the evaporator washes water at 50 °C. When operating in the remaining temperature range, the R134A refrigerant, filled with 10%, also performed best. In addition to the largest heat fluxes transmitted through the heat pipe, this factor in the said filling achieved the best heat exchange efficiency, reaching 95%. From the remaining tested working fluids, the R404A refrigerant, also with 10% filling, proved to be effective for the efficiency of the heat pipe tested, although it achieved worse results than the R134A refrigerant. The R407C factor was found to be the worst of the factors tested, both in terms of the transferred heat fluxes and the efficiency of heat transfer through the heat pipe.

4.2. Discussion on the Efficiency of Heat Pipes and the Value of the Heat Flux Transferred by Them

Comparing the heat pipes analyzed on the basis of the analysis of experimental and thermographic studies [28–30], it can be concluded that when comparing the most favorable fillings of a heat pipe with a length of 550 mm compared to a heat pipe with a length of 1769 mm, a 1769 mm long tube is more favorable when made of copper. A copper heat tube transfers higher heat fluxes and has comparable efficiency, which was also used in different branch scale installations for thermo-chemical conversion of biomass [31,32]. The advantage of the copper tube is primarily due to the larger heat transfer surface area of both the evaporating and condensing heat tube sections which can also be used in different applications for heat recovery [33,34]. Of the four tested refrigerants (R134A, R404A, R407C, and R410A), R404A is the best refrigerant to fill the brass heat pipe with a length of 550 mm and a diameter of 22 mm. The value of the heat flux absorbed by the evaporator and given off by the condenser, in this case for 20% of the volume filled, reaches a value of ~135 W (similar to the R410A charge), but the operating range of the heat pipe is greater for the R404A refrigerant and amounts to 20–50 °C compared to R410A where it lies in a range of 25–50 °C.

The heat tube was filled to amounts of 2.5%, 5%, 10%, 20%, 30%, and 40% of its total volume with the specified refrigerants. Each of these factors has different physicochemical properties that affect the operation of the heat pipe. It was not the authors' intention to calculate the heat transfer coefficient in this paper. The aim of the research was to balance the heat pipe heat exchanger in terms of the lower–upper heat transfer efficiency and to determine the efficiency of the entire apparatus, because this is important for practical air conditioning and ventilation applications. The heat transfer coefficient between the liquid flowing around the tube and the boiling medium for industrial applications is not a key parameter in experimental research according to the authors. In the next publication on the numerical model built by the authors and the simulation of heat transfer processes in a heat pipe, a large section will be devoted to the issue of the heat transfer coefficient.

4.3. Discussions on the Advisability of Using Heat Pipe Heat Exchangers for the Needs of Civil Engineering and Air Conditioning to Reduce Low Emissions

During the past half-century, there has been a significant development in the use of heat pipes, pioneered by their use in space science. Today they are used in many industries. One of the most interesting applications of heat pipes is their use for drying and cooling air, and for heat recovery and passive cooling of rooms. Other important applications are uses in solar pipes of solar collectors, and as passive heat exchangers for regulating ground temperature and heating the surfaces of bridges and viaducts. Due to its structure, the heat pipe is an economical heat exchanger compared to those commonly used. For this reason, in an era of striving to save and reduce the unnecessary dissipation of energy, interest in the use of this type of heat exchanger is still growing. A series of tests was thus necessary to analyze the operation of heat pipes and to determine the most effective heat exchangers of the heat pipe type depending on their operating conditions. The results of experimental tests on the heat pipes and thermographic tests fully confirmed the legitimacy of using heat exchangers of the heat pipe type to reduce emissions. The research results prove the possibility of a wide application of heat pipes to reduce the consumption of primary energy by reducing energy dissipation in heat transfer processes and by their intensification. The results of the experimental studies obtained prove the high efficiency of heat transport by means of heat pipes, which can significantly affect the efficiency of thermal devices.

5. Conclusions

The obtained efficiency of heat recovery from the ground of the examined exchangers obtained values over 90%, which will enable the reduction of low emissions in heating systems of buildings. The results included in the study will also enable the effective use of land as a heat store. As part of this work, issues related to the analysis of the efficiency of heat pipes were analyzed to reduce emissions based on experimental studies. In relation to the above, it was proved that the analysis of the operation of the heat pipe type heat exchanger enables the selection of the best working medium among those tested, and its optimal amount used to fill the heat pipe. This allows the maximum efficiency and the maximum value of the transferred heat fluxes in the temperature range of the lower source of 15–50 °C to be obtained.

The experimental studies conducted permitted a number of conclusions to be drawn regarding the solutions used in heat pipes. These conclusions mainly concern:

- selection of the best working medium and determination of its optimal quantity;
- the efficiency of the heat pipes and the value of the heat flux transmitted by them;
- the desirability of using heat pipe heat exchangers for structural engineering and air conditioning to reduce low emissions.

Based on the results obtained from this research regarding the high efficiency of heat pipe heat exchangers (92–98%), it is possible to use heat pipes as heat exchangers that intensify the heat exchange between lower heat sources from renewable sources (e.g., ground) and heat receivers of central heating devices for buildings (e.g., the heat pump). The conclusions from the research conducted prove that the use of heat pipes in construction engineering and air-conditioning is consistent with activities limiting primary energy consumption and increasing energy efficiency by limiting emissions [33]. Conclusions resulting from this study also confirm the possibility of heat pipe operation in a wide temperature range and the ability to work at the smallest temperature differences, which contributes to more efficient use of renewable sources and allows their use in places which, thus far, are limited by profitability constraints. In the next stage of work, the authors plan to carry out research on a ready-made prototype of a ground heat exchanger of the heat tube type in real conditions.

Author Contributions: Conceptualization, Ł.A., S.S., P.P., P.K. (Przemysław Kubiak), P.K. (Piotr Kuryło); methodology, Ł.A., S.S., P.P., P.K. (Piotr Kuryło), P.K. (Przemysław Kubiak), G.W., J.C.; software, Ł.A., S.S., P.P.; validation, Ł.A., S.S., P.P., P.K. (Piotr Kuryło), P.K. (Przemysław Kubiak); formal analysis Ł.A., S.S., P.P., P.K., G.W., J.C.; investigation, Ł.A., S.S., P.P., P.K., F.M., K.K., S.P., A.O., J.S., P.K. (Piotr Kuryło), P.K. (Przemysław Kubiak); resources Ł.A., S.S., P.P., G.W., J.C.; data curation, Ł.A., S.S., P.P.; writing—original draft preparation, Ł.A., S.S., P.P., P.K. (Piotr Kuryło), P.K. (Przemysław Kubiak), K., F.M., K.K., S.P., A.O., J.S., P.K.; writing—review and editing, Ł.A., S.S., P.P., P.K. (Piotr Kuryło), P.K. (Przemysław Kubiak), F.M., K.K., S.P., A.O., J.S., P.K., G.W., J.C.; visualization, Ł.A., S.S., P.P.; supervision, Ł.A., S.S., P.P., P.K. (Piotr Kuryło), P.K. (Przemysław Kubiak), G.W., J.C., F.M., K.K., S.P., A.O., J.S., P.K. (Piotr Kuryło), P.K. (Przemysław Kubiak), G.W., J.C.; project administration, Ł.A., S.S., P.P., P.K. (Piotr Kuryło), P.K. (Przemysław Kubiak); funding acquisition, Ł.A., S.S., P.P., G.W., J.C.; All authors have read and agreed to the published version of the manuscript. Please turn to the CRediT taxonomy for the term explanation. Authorship must be limited to those who have contributed substantially to the work reported.

Funding: Statutory resources of the Lodz University of Technology.

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: Not applicable.

Acknowledgments: NCN GRANT: Preludium II no: DEC-2011/03/N/ST8/05912 pt. "Research on heat exchange processes and distribution of velocity fields by the PIV method and the effects of phase changes and capillary transport occurring in heat exchangers of the heat tube type". The grant was financed by the National Science Center in Krakow.

Conflicts of Interest: The authors declare no conflict of interest.

Appendix A

The tests that were carried out are affected by errors resulting from the measurements, the equipment used, and calculations. Below is an example of calculating errors for 10% of the heat pipe volume with R134A refrigerant and the hot water temperature at the heat exchanger inlet of ~35 °C.

Limit error of hot and cold water flow through the rotameter:

$$\Delta_{lim}\left(\dot{V}_c\right) = 0.5\frac{l}{h}, \tag{A1}$$

Δ_{lim}–limit error.

Total uncertainty of hot and cold water flow through the rotameter:

$$u_c\left(\dot{V}_c\right) = \frac{\Delta_{lim}\left(\dot{V}_c\right)}{\sqrt{3}} = \frac{0.5\frac{l}{h}}{\sqrt{3}} = 0.3\frac{l}{h} \tag{A2}$$

Limit error of the millivoltmeter for hot water flowing into the exchanger is $E_9 = 1.34$ mV:

$$\Delta_{lim}(E_9) = 0.015\%{\cdot}E_9 + 0.004\%{\cdot}Z, \tag{A3}$$

$$\Delta_{lim}(1.340\ \text{mV}) = 0.015\%{\cdot}1.340\ \text{mV} + 0.004\%{\cdot}200\ \text{mV} = 0.008\ \text{mV}, \tag{A4}$$

where 0.015% and 0.004%–the values stated by the manufacturer of the millivoltmeter; Z—measuring range of a millivoltmeter.

Total uncertainty of a millivoltmeter for hot water flowing into the exchanger: E_9 = 1.34 mV.

$$u_c(E_9) = \frac{\Delta_{lim}(E_9)}{\sqrt{3}} = \frac{0.0082\ \text{mV}}{\sqrt{3}} = 0.005\ \text{mV}, \tag{A5}$$

Millivoltmeter limit error for hot water flowing from the exchanger: E_{10} = 1.10 mV.

$$\Delta_{gr}(E_{10}) = 0.015\%{\cdot}E_{10} + 0.004\%{\cdot}Z, \tag{A6}$$

$$\Delta_{gr}(1.1\ \text{mV}) = 0.015\%{\cdot}1.1\ \text{mV} + 0.004\%{\cdot}200\ \text{mV} = 0.008\ \text{mV}, \tag{A7}$$

The total uncertainty of a millivoltmeter for the warm flowing water: E_{10} = 1.10 mV.

$$u_c(E_{10}) = \frac{\Delta_{lim}(E_{10})}{\sqrt{3}} = \frac{0.008\ \text{mV}}{\sqrt{3}} = 0.005\ \text{mV}, \tag{A8}$$

Thermoelectric force limit error:

$$\Delta_{gr}(\Delta E_c) = \Delta_{gr}(E_{10}) + \Delta_{gr}(E_9) = 0.008\ \text{mV} + 0.008\ \text{mV} = 0.016\ \text{mV}, \tag{A9}$$

Total uncertainty of measuring the difference in hot water temperature:

$$u_c(\Delta t_c) = \frac{\Delta gr(\Delta E)}{\sqrt{3}}{\cdot}25\frac{°\text{C}}{\text{mV}}, \tag{A10}$$

The relative error of the hot water heat flux:

$$\frac{u_c\left(\dot{Q}_g\right)}{\dot{Q}_g} = \sqrt{\left(\frac{u_c\left(\dot{V}_c\right)}{\dot{V}_c}\right)^2 + \left(\frac{u_c(\Delta t_c)}{\Delta t_c}\right)^2} \tag{A11}$$

$$\frac{u_c\left(\dot{Q}_g\right)}{\dot{Q}_g} = \sqrt{\left(\frac{0,3\frac{l}{h}}{15\frac{l}{h}}\right)^2 + \left(\frac{0.24\ °\text{C}}{6.05\ °\text{C}}\right)^2} = 0.044 \tag{A12}$$

Uncertainty of the hot water heat flux:

$$u_{\dot{Q}_g} = \frac{u_c\left(\dot{Q}_g\right)}{\dot{Q}_g}{\cdot}\dot{Q}_g = 0.044{\cdot}104.64\ \text{W} = 5\ \text{W} \tag{A13}$$

The value of the hot water heat flux, taking into account the measurement error, is: $\dot{Q}_g = 104.64(5)$ W.

The error of the value of the hot water stream is therefore 5 W. The limit error of the cold water flow and the total uncertainty of its flow are specified above.

Error calculation example for the cold water temperature at the inlet to the heat exchanger of ~10 °C:

Millivoltmeter limit error for cold water flowing into the exchanger for $E_{11} = 0.390$ mV:

$$\Delta_{gr}(E_{11}) = 0.015\% \cdot E_{11} + 0.004\% \cdot Z, \tag{A14}$$

$$\Delta_{gr}(0.390 \text{ mV}) = 0.015\% \cdot 0.390 \text{ mV} + 0.004\% \cdot 200 \text{ mV} = 0.008 \text{ mV}, \tag{A15}$$

Total uncertainty of a millivoltmeter for cold water flowing into the exchanger for $E_{11} = 0.390$ mV:

$$u_c(E_{11}) = \frac{\Delta gr(E_{11})}{\sqrt{3}} = \frac{0.008 \text{ mV}}{\sqrt{3}} = 0.005 \text{ mV}, \tag{A16}$$

Millivoltmeter limit error for cold water flowing from the exchanger for $E_{12} = 0.568$ mV:

$$\Delta_{gr}(E_{12}) = 0.015\% \cdot E_{12} + 0.004\% \cdot Z \tag{A17}$$

$$\Delta_{gr}(0.68 \text{ mV}) = 0.015\% \cdot 0.568 \text{ mV} + 0.004\% \cdot 200 \text{ mV} = 0.008 \text{ mV}, \tag{A18}$$

Total uncertainty of a millivoltmeter for cold water flowing from the exchanger $E_{12} = 0.568$ mV:

$$u_c(E_{12}) = \frac{\Delta gr(E_{12})}{\sqrt{3}} = \frac{0.008 \text{ mV}}{\sqrt{3}} = 0.005 \text{ mV}, \tag{A19}$$

Thermoelectric force limit error:

$$\Delta_{gr}(\Delta E_z) = \Delta_{gr}(E_{11}) + \Delta_{gr}(E_{12}) = 0.008 \text{ mV} + 0.008 \text{ mV} = 0.016 \text{ mV} \tag{A20}$$

Total uncertainty of measuring the cold water temperature difference:

$$u_c(\Delta t_z) = \frac{\Delta gr(\Delta E)}{\sqrt{3}} \cdot 25 \frac{°\text{C}}{\text{mV}}, \tag{A21}$$

$$u_c(\Delta t_z) = \frac{0.016 \text{ mV}}{\sqrt{3}} \cdot 25 \frac{°\text{C}}{\text{mV}} = 0.23 \text{ °C}, \tag{A22}$$

the relative error of the cold water heat flux:

$$\frac{u_C\left(\dot{Q}_z\right)}{\dot{Q}_z} = \sqrt{\left(\frac{u_c\left(\dot{V}_z\right)}{\dot{V}_z}\right)^2 + \left(\frac{u_c(\Delta t_z)}{\Delta t_z}\right)^2} \tag{A23}$$

$$\frac{u_C\left(\dot{Q}_z\right)}{\dot{Q}_z} = \sqrt{\left(\frac{0.3\frac{l}{h}}{15\frac{l}{h}}\right)^2 + \left(\frac{0.23 \text{ °C}}{5.67 \text{ °C}}\right)^2} = 0.045 \tag{A24}$$

cold water heat flux uncertainty:

$$u_{\dot{Q}_z} = \frac{u_C\left(\dot{Q}_z\right)}{\dot{Q}_z} \cdot \dot{Q}_z = 0.045 \cdot 99 \text{ W} = 4 \text{ W}, \tag{A25}$$

The value of the cold water stream, taking into account the measurement error, is: $\dot{Q}_z = 99(4)$ W. The error of the cold water flow value is, therefore 4 W.

Heat pipe efficiency relative error:

$$\frac{u_C(\eta)}{\eta} = \sqrt{\left(\frac{u_{\dot{Q}_g}}{\dot{Q}_g}\right)^2 + \left(\frac{u_{\dot{Q}_z}}{\dot{Q}_z}\right)^2}, \tag{A26}$$

$$\frac{u_C(\eta)}{\eta} = \sqrt{\left(\frac{5\ \mathrm{W}}{104.64\ \mathrm{W}}\right)^2 + \left(\frac{4\ \mathrm{W}}{99\ \mathrm{W}}\right)^2} = 0.06, \tag{A27}$$

uncertainty of heat pipe efficiency:

$$u_\eta = \frac{u_C(\eta)}{\eta} \cdot \eta = 0.06 \cdot 0.95 = 0.06 \tag{A28}$$

The efficiency value of the heat pipe, taking into account the measurement error, is written as: η = 0.95(6). The error of the heat pipe efficiency value is 6%.

The errors in all other cases were determined in a similar way. As a result of the performed error calculations, their values are within the range 3–8%. With the increase in hot water temperature at the entrance to the heat exchanger heating the heat pipe, the errors in the values of heat flux absorbed and returned by the heat pipe increase, and the errors in the efficiency values reach smaller and smaller values.

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