

Article

The Heat Transfer Coefficient During Pool Boiling of Refrigerants in a Compact Heat Exchanger

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Abstract

The results of experimental data on the heat transfer coefficient during the boiling of pro-ecological refrigerants in a compact tube-shell heat exchanger are presented. The boiling process occurred in the micro-space of the exchanger shell on the surface of horizontal tubes, which were heated from the inside with warm water. The flow of the refrigerant was gravity-based. The heat exchanger was practically flooded with liquid refrigerant at a saturation temperature (t_s), which flowed out after evaporation in a gaseous form. The tests were conducted for four refrigerants: R1234ze, R1234yf, R134a (a high-pressure refrigerant), and HFE7100 (a low-pressure refrigerant). Thermal characteristics describing the heat transfer process throughout the entire compact heat exchanger, specifically for the boiling process itself, were developed. It was found that in the case of micro-space boiling, there is an exponential dependence of the heat transfer coefficient on the heat flux density on the heated surface. Experimental data were compared to experimental and empirical data presented in other studies. Our own empirical models were proposed to determine the heat transfer coefficient for boiling in a mini-space for individual refrigerants. The proposed calculation models were also generalized for various refrigerants by introducing the value of reduced pressure into the calculation relationship. The developed relationship enables the determination of heat transfer coefficient values during boiling in a micro-space on the surface of horizontal tubes for various refrigerants with an accuracy of $\pm 25\%$.

Keywords: pool boiling; heat transfer coefficient; refrigerants; heat exchanger



Academic Editors: Robert Černý, Mahmoud Bourouis and Jiangshui Luo

Received: 10 June 2025

Revised: 14 July 2025

Accepted: 24 July 2025

Published: 29 July 2025

Citation: Kruzel, M.; Bohdal, T.; Dutkowski, K.; Wołosz, K.J.; Robakowski, G. The Heat Transfer Coefficient During Pool Boiling of Refrigerants in a Compact Heat Exchanger. *Energies* **2025**, *18*, 4030. <https://doi.org/10.3390/en18154030>

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1. Introduction

During the boiling process, the liquid phase of the fluid changes into the vapor phase, which takes place inside the liquid. In this process, energy is supplied utilizing heat from the heating surface to the liquid, and two conditions must be met: the existence of a temperature gradient on the heated surface of the wall (the existence of the so-called superheated liquid) and the existence of nuclei of a new phase (i.e., vapor nuclei). Vapor nuclei are formed on surface irregularities and may also be facilitated by fluctuations in liquid density and contamination. The temperature of the liquid T from which the new vapor phase is to be produced must be higher than the saturation temperature T_s , corresponding to the absolute pressure at the interface. The temperature on the heated wall should be slightly higher than outside the boundary layer. In this situation, the boundary

layer contains a lost liquid in conditions of a lack of stable equilibrium. This causes the system to strive for a stable equilibrium state, and the boiling process begins. On the heated heat exchange surface, vapor bubbles are formed, which grow and detach from the wall. The resulting gas phase is in a thermodynamic equilibrium under these conditions. During boiling, the liquid turns into a gas, and the existing structure of the two-phase system is a mixture of liquid and vapor bubbles. For a vapor bubble to exist in a liquid, the vapor pressure within it must be greater than the pressure of the liquid due to the action of surface tension forces. During the phase transition, heat, momentum, and mass are exchanged. The vapor bubbles rise, move, turbulate, and mix with the surrounding liquid and extract the heat of vaporization from it. There is a continuous decrease in the mass of the liquid and an increase in the mass of the vapor. This type of boiling is called vesicular boiling. A further increase in heat flux on the heated wall can lead to a so-called first boiling crisis, where steam bubbles can merge and create a layer of steam on the heated surface, which will constitute high resistance to heat transfer. Then, there will be a rapid increase in wall temperature and a decrease in the heat transfer coefficient value. Excess produced steam will break away from the wall in the form of large bubbles. This type is called film boiling. An intermediate form of boiling is also possible, called transitional boiling. It is characterized by a lack of stable equilibrium. Vesicular boiling is characterized by much higher heat exchange intensity. It is much more often used in practice than film boiling, which is undesirable due to the possibility of destroying the heated surface with an excessively high heat flux density. The dependence of heat flux density on the heated wall, as a function of the degree of wall overheating, is called the boiling curve.

The presented nucleation mechanism occurs when the shape of the space in which the process takes place does not affect it, and when such an interaction occurs. If the shape of the space in which the boiling phenomenon occurs does not significantly affect heat exchange, then we are dealing with boiling in volume. If the shape of this space has a significant effect, we are talking about boiling in flow. Boiling processes are widely used in technology and industrial practice. This results from the fact that a large amount of heat is exchanged during the phase change, and the heat transfer coefficients reach significant values, several times higher than in single-phase systems. Since steam production is associated with heat extraction from the environment, the boiling phenomenon is commonly used in cooling processes. This applies to both high-temperature (heating) and low-temperature (refrigeration and air conditioning) objects. The practical application of boiling processes brings tangible benefits in obtaining high heat exchange intensity and a constant process temperature, which depend on the value of the prevailing pressure. This reduces the size of heat exchangers and facilitates the control of the heat exchange process. Boiling processes are used in large industrial thermal power plants and small installations operating for local cooling. This applies to mini-power installations for cooling electronic components, computer processors, medical and space equipment, etc.

In the second half of the 20th century, intensive work was carried out on the study of the phenomena of boiling in volume and boiling in flow in conventional systems. The results obtained show wide application in large and medium-sized power plants. Several computational models were created based on the results of experimental studies and theoretical analyzes [1,2]. Current work on boiling processes mainly concerns research in the mini-space. This applies to pool boiling and flow boiling (boiling in minichannels) [2–5].

Despite the large number of published works on bubbly flow boiling (including for refrigerants), there are indications of the need for further research, both experimental and theoretical [6]. The study results so far concern the vast majority of steady states of the bubble boiling process [7–10]. Further attempts have been made to explain the mechanism

of energy transport and to precisely describe the qualitative and quantitative “physics of the phenomenon”.

Many correlations and procedures have been developed to determine the heat transfer coefficient. However, they are related mainly to conventional channels. Ref. [11] offers a computational model for determining the heat transfer coefficient under pool boiling conditions:

$$\text{Nu} = 0.44 \cdot \text{Re}^{0.7} \cdot \text{Pr}^{0.35} \cdot K_p^{0.7}, \quad (1)$$

where

$$K_p = \frac{p}{\sqrt{g \cdot (\rho' - \rho'') \cdot \sigma}} = \frac{p \cdot l}{\sigma}, \quad (2)$$

l is the characteristic dimension:

$$l = \frac{c'_p \cdot \rho' \cdot \sigma \cdot T_s}{r^2 \cdot \rho''^2}, \quad (3)$$

$$\text{Re} = \frac{q \cdot l}{r \cdot \rho'' \cdot \nu}, \quad (4)$$

$$\text{Nu} = \frac{h \cdot l}{\lambda'}. \quad (5)$$

when $\text{Re} < 0.01$, then (1) stands for the following:

$$\text{Nu} = 0.0625 \cdot \text{Re}^{0.5} \cdot \text{Pr}^{1/3}, \quad (6)$$

and when $\text{Re} \geq 0.01$, then

$$\text{Nu} = 0.125 \cdot \text{Re}^{0.65} \cdot \text{Pr}^{1/3}. \quad (7)$$

The author of the paper developed a correlation for nucleate boiling in volume, which takes into account the pressure and molecular mass of the working medium, as well as surface roughness. It can be helpful in practical calculations and in analyzing the work of heat exchangers.

$$h = C \cdot q^{0.67} \cdot M^{-0.5} \cdot p_r^m \cdot (-\ln p_r)^{-0.55}, \quad (8)$$

where

$$m = 0.12 - 0.2 \cdot \lg(Ra)_{\mu\text{m}},$$

C is the constant ($=90$),

q is the heat flux, $\text{J} \cdot \text{m}^{-2}$,

M is the relative molecular mass,

p_r is the reduced pressure, representing the ratio of working fluid pressure to critical pressure, and

Ra is the average surface roughness [μm].

For the vesicular boiling range, simplified formulas in the form of dependencies are often used in engineering calculations, such as $h = f(q)$ or $h = f(\Delta T)$, where $\Delta T = T_w - T_s$. Simplifications of this type were shown above in the case of the Kruzhylin and Kutateladze relations. These are most often formulas of the following type [12]:

$$h = C \cdot q^n, \quad (9)$$

and

$$h = C_1 \cdot q^m \cdot \Delta T^k. \quad (10)$$

The above-simplified calculation formulas are dedicated to specific working media. The authors provide exact values of the coefficients C , C_1 , m , and n for a given refrigerant.

There are also numerous works on the effect of exchanger geometry on the heat transfer intensity during boiling. The authors of [13] determined the effect of the spacing between the tubes and the pressure on the heat transfer coefficient value on the surface of an 18-tube (3×6) bundle of polished copper tubes with an external diameter of $d_e = 18$ mm, heated from the inside electrically. The arrangement of the tubes was staggered. The researchers found the best thermal performance for a 0.3 mm pitch. For a pitch greater than 1 mm, the pressure effect on the transferred heat flux was negligible. The authors of [14] studied water boiling at atmospheric pressure on smooth stainless steel tubes with an external diameter of $d_e = 19.05$ mm, heated electrically. Two or three tubes were arranged vertically above each other with a 1.5 to 6.0 pitch-to-tube diameter ratio. The density of the water mass flow in the saturated state flowing into the tank varied within the range of $G = 0 \div 10 \text{ kg m}^{-2} \text{ s}^{-1}$. The studies showed that the heat transfer coefficient on the lowest tube did not depend on the presence of the tubes above. The highest heat transfer coefficient was obtained on the highest tube in the three-tube system for boiling in a large volume. The authors of [15] conducted studies on boiling distilled water under negative pressure ($35 \div 97.5$ kPa) on two smooth copper tubes with a diameter of 32 mm, placed one above the other, and did not notice a significant effect of the tube material on the values of the heat transfer coefficient. Ref. [16] presents the results of a study on the boiling process of the R113 refrigerant on a bundle consisting of 241 tubes. The studies were carried out at atmospheric pressure for tubes with an external diameter of 19.05 mm and a pitch (in a square system) of 25.4 mm. The heat transfer coefficients at the bottom of the bundle were similar to those achieved for a single tube, while at the top rows, they were much higher. The authors of ref. [17] investigated the boiling in volume of the following refrigerants: R134a, R1336mzz (E), and R1336mzz (Z) on the outer surface of horizontal circular tubes. Smooth tubes and those with modified surfaces were used for the tests. Three different structures were investigated: copper oxide and etched aluminum micro-nanostructures and boehmite nanostructures. In the case of individual refrigerants, different relationships between structure and heat transfer coefficient (HTC) increases were demonstrated. In the case of boiling refrigerant R134a on the surface with etched aluminum, an increase in HTC of up to 250% was obtained, while in the case of boehmite structures, a decrease of 15% occurred. In the case of CuO, at higher heat fluxes, there was a 25% increase in HTC, whereas at lower heat fluxes, the effects were negligible compared to those of smooth copper tubes. Based on the available literature, it can be concluded that the heat transfer intensity during boiling in a bundle of tubes depends on many factors, including heat flux density, liquid properties, pressure, type of tube surface, and the way the tubes are arranged and pitched. Therefore, various types of empirical or semi-empirical correlations are used for calculations in which constant values require experimental determination. In paper [18], the results of experimental studies of volume boiling on the surface of a smooth tube with an external diameter of 19.05 mm and a developed surface (GEWA-B5H, Wieland-Werke AG, Ulm, Germany) of R134a and R1234ze refrigerants and their mixtures with the addition of a lubricant are presented. The tests were conducted in a heat flux range from 10 kW/m^2 to 90 kW/m^2 , showing a decrease. The authors noted that the pure R1234ze heat transfer coefficient on the GEWA-B5H tube was almost 7%, 10%, and 18% lower than that of R134a at saturation temperatures of -6°C , 0°C , and 10°C , respectively. For smooth pipes, the R1234ze heat transfer coefficient was approximately 27% lower than that of R134a in the given range of saturation temperatures. The authors proved that even a small amount of lubricating oil lowers the heat transfer coefficient on the improved surface, which mostly depends on the transport properties, reduced pressure, and the saturation temperature of the mixture. Ref. [19] contains the results of the analysis of heat transfer during boiling in volume at a high saturation temperature of the working medium R245fa.

Boiling took place on the surface of a smooth pipe and a modified surface in the shape of fish scales. The results showed that OHTC for the smooth pipe was $1.9\text{--}3.9\text{ kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$, and that of the pipe with fish scales was $3.8\text{--}4.3\text{ kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ at a saturation temperature of $80.5\text{ }^{\circ}\text{C}$. The HTC for the pipe with fish scales was 1.45–3.13 times higher than that of the smooth pipe. The external HTC of the smooth pipe and the pipe with fish scales increased along with the heat flux, and the rate of enhancement decreased successively. In ref. [20], the influence of contamination on the external surface of a 25 mm diameter pipe on the heat transfer coefficient HTC during boiling in a volume of R245fa medium was presented. The data on HTC between a smooth pipe and a compromised pipe were compared in the temperature range of $85\text{--}120\text{ }^{\circ}\text{C}$. The HTC for the smooth pipe was $1.3\text{--}3.4\text{ kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$, and for the contaminated pipe, $1.2\text{--}3.0\text{ kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$. The decrease in the HTC value for the smooth pipe was 9.5–13.2%, and for the contaminated pipe, 7.1–10.7%. A notable trend in this decrease was observed with increasing temperature. A significant part of the research study concerns various methods of heat transfer intensification during boiling in a volume on a heated surface. Most often, these are passive methods consisting of developing the heat transfer surface. Ref. [21] aimed to experimentally investigate the heat transfer properties during boiling in a volume of R134a medium on the surface of a smooth pipe and three pipes with microfins at saturation temperatures in the range between $3\text{ and }40\text{ }^{\circ}\text{C}$ and heat fluxes of $5.5\text{--}76.5\text{ kW}\cdot\text{m}^{-2}$. The pipes were heated from the inside with warm water. The experimental data showed that the HTC initially increases and then decreases with an increase in the heat flux. Still, the saturation temperature and water velocity have a negligible effect on the value of this coefficient. Microfins facilitate the nucleation of bubbles, and the enhancement of HTC varied between 2.08–4.59. A new relationship was developed based on the ratio of the fin height to the external diameter. The proposed correlation enables the prediction of the HTC with less than 10% average deviation. The authors of ref. [22] proposed a new method for enhancing heat transfer during volume boiling on a horizontal tube. A metal-based foam-immersed tube was proposed to enhance the heat transfer of flooded evaporators. A horizontal tube bundle was used in a staggered arrangement. Furthermore, the performance of eco-friendly refrigerants (R1234yf and R1234ze (E)) was compared with that of R134a. Compared to R134a, the HTC coefficients of R1234yf were almost 10% higher and 5% lower for R1234ze (E). In ref. [23], the experimental data of boiling on surfaces with improved microchannels were presented, where cylindrical spaces were closed using several empty cylinders of heights 1, 3, 5, and 7 mm, and a 1 mm gap was maintained between the wall and the hot surface. The influence of different heights of cylindrical spaces on the heat transfer characteristics of volume boiling was analyzed. This solution allowed for significant intensification of heat transfer compared to a flat surface made of copper, especially with an increase in the confined space height. A predictive model was established, showing very good agreement according to experimental data. The authors of ref. [24] analyzed the usefulness of volume boiling for cooling electronic components. For this purpose, the cooled components were placed in a liquid chamber, where the process of developing supercooled boiling of water occurred on the heated surface. Three configurations of the heat exchange surface in contact with water were tested: a flat surface with a small diameter, a thin plate, and a ribbed surface. It was shown that the highest heat exchange intensity occurred in the case of using a flat plate with a 9 mm diameter connected to a ribbed copper surface. Then, the highest heat flux was obtained at the highest temperature. It was found that supercooled boiling is a good method of increasing the CHF. In addition, the emission of microbubbles, which is observed at high supercooling of the liquid, is a promising technology for achieving high heat flux cooling, exceeding the ordinary CHF. The authors of [25,26] conducted studies on boiling heat transfer during the flow of working fluids

(FC72, HFE649, HFE7000, and HFE7100) in a single rectangular minichannel with a depth of 1.7 mm, oriented vertically and horizontally. The channel was heated asymmetrically on one side. The heated wall surface was smooth or improved on the fluid side. Improved surfaces were obtained by vibration-assisted laser surface texturing or electromachining. Porous surfaces produced by soldering or sintering iron powder on the heated surface were also studied. The relationships between the heat transfer coefficient and the spatial variable on the flow direction at supercooled and saturated boiling, as well as boiling curves, were shown. The influence of surface roughness on the heat transfer intensity was found. In the field of boiling, numerous studies were also conducted on the determination of boiling curves. In ref. [27], a review of experimental studies conducted on boiling in volume and flow on engineering nano- and microstructures in order to improve heat transfer during boiling, in particular, the HTC and CHF, was presented. The nano/micro-porous featured surface heat transfer was investigated for different coating conditions. Surfaces with micro- and nanostructures offer new possibilities for heat transfer enhancement during boiling. The HTC and CHF increased. Ref. [28] presents experimental data on the complete boiling curve of tetrafluoromethane (R14) on a horizontal copper surface. During the tests, the following phenomena were observed: bubble boiling, CHF, transition boiling, Leidenfrost point, and film boiling. The CHF point was 220.39 kW/m^2 , and the corresponding surface superheating temperature was 16.1 K. The measured heat flux was 126.83 kW/m^2 , and the surface superheating temperature of the Leidenfrost point was 87.1 K. Numerous researchers have also conducted theoretical work in the field of numerical modeling of the boiling process. The results of numerical simulations of the bulk boiling process based on the improved pseudopotential Boltzmann model of the liquid–vapor phase change were presented in ref. [29]. The saturated boiling curves from the start of nucleate boiling to the CHF, the transition boiling regime, and the stable layer boiling regime were developed. The simulation data showed a significant non-uniformity in the temperature distribution near the top of the hot surface in the nucleate boiling regime, at the critical heat flux point and in the transition boiling regime. The two-dimensional heat conduction cannot be ignored when assessing the heat flux just below the top of the heater surface. The study of the boiling curve was also devoted to ref. [30]. The volume boiling curve on smooth and rough surfaces, including the nucleation region, CHF, and transition boiling, was predicted using a deep learning (DL) model that is based on physical and correlation features. High accuracy was found for a wide range of process conditions (pressure, type of fluid, etc.). In ref. [31], a semi-analytical correlation for bubble growth dynamics was proposed based on experimental data on boiling on porous surfaces. The model accounts for the temporal variation of the evaporation rate within the mini-tunnels to determine the latent heat flux resulting from internal evaporation and the frequency of bubble formation. Vapor nucleation density was predicted using an empirical formula and then used for the estimation of the total heat flux from the porous developed surface. The agreement of the model with the experimental results was determined to be within ± 30 .

An analysis of literature data suggests that minispaces, or small, confined spaces, can alter the boiling point of a liquid by affecting pressure and surface tension. In minispaces, the pressure can be higher than under normal conditions, leading to a higher boiling point. In addition, small spaces can affect the surface tension of a liquid, which can also change the boiling point.

Although the number of studies, especially experimental ones, describing the phenomena of volume boiling in conventional systems is large [32–37], it is insufficient in the case of small-sized systems (minispace, minichannels). The authors of [38–44] conducted experimental studies and theoretical analyses of nucleate boiling phenomena in the flow of low-pressure refrigerants. The results of the studies confirm the high energy poten-

tial of these refrigerants. Volume boiling of hydrofluoroolefins was conducted and also analyzed in [3,45–49]. The aforementioned studies did not concern heat exchangers but only individual minichannels. Another significant problem is using new, environmentally friendly refrigerants as substitutes for freons withdrawn from refrigeration technology. Experimental and theoretical studies should be conducted to use these refrigerants, both for boiling in volume and flow. They will fill the information gap, which will be useful for cognitive and application purposes. With the above in mind, the authors conducted experimental studies and theoretical analyzes of the boiling of new pro-ecological refrigerants in volume boiling conditions. The boiling process was carried out on the surface of horizontal smooth tubes placed inside the shell of a compact exchanger. Due to the small volume of the compact exchanger, the heat exchange intensity on individual exchanger tubes was not tested separately. Still, they were treated together, determining the average values of the heat exchange coefficients. This study contains the results of experimental data and a proposal for a generalized computational correlation that allows for the determination of the value of the heat transfer coefficient. It can be used to design compact heat exchangers for direct cooling of fluids and indirect systems for cooling mini-elements in electronic systems, medical systems, etc.

2. The Experimental Facility

Experimental research on pool boiling was carried out on a specially designed and built experimental set-up, which consisted of the following essential elements:

1. A test system, containing measurement sections with instrumentation;
2. The installation's refrigerant supply system;
3. Power installation;
4. Heating water installation;
5. Control and measurement equipment that cooperate with a computer data recording and processing system.

The refrigerant loop consisted of a refrigerant tank, a Coriolis mass flow meter, a pressure sensor, three temperature sensors at the tank's outlet, and a cooler at the mini heat exchanger's inlet and outlet.

The heating fluid loop consisted of a cryostat regulating the liquid temperature, an adjustable pump, and a mass flow meter. For an even mass flow rate distribution of water through the individual heating channels, the pressure drop values were compared along the channels' length. The differences between the values for the individual channels were within 2%. The temperature of both media was diagnosed using K-type thermocouples. The measurement data were saved using a data logger. The main object of the research is a shell-and-tube mini heat exchanger. The heat exchanger is presented in Figure 1.

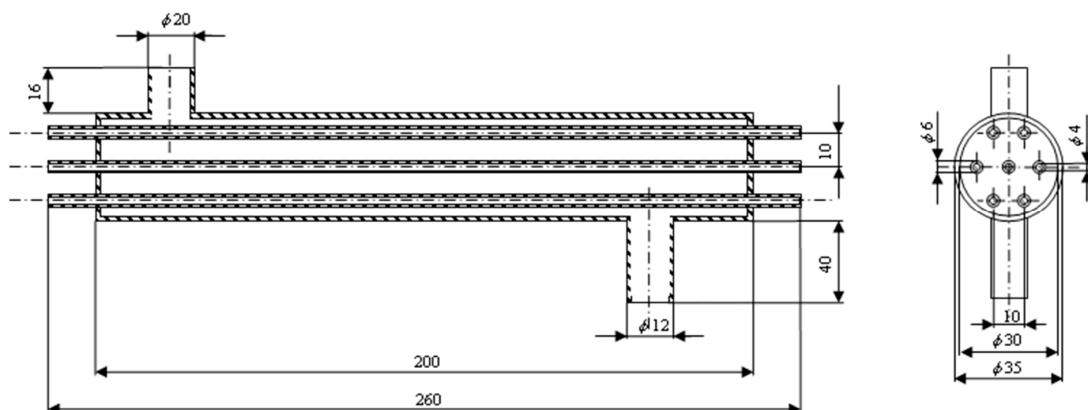


Figure 1. Overall view of the heat exchanger.

It consists of seven minichannels with a of $d_i = 4$ mm, a body with an internal diameter of $d_i = 30$ mm, and a length of $L = 200$ mm. The mini exchanger uses pipe channels oriented horizontally with a of $d_e = 6$ mm. When the pipe diameters are compared to those used in practice in typical heat exchangers, they become mini-channels. According to the Mehendale classification, channels with this diameter are included in compact heat exchangers ($d_h = 1\text{--}6$ mm). Based on this classification, the diameter $d_e = 6$ mm is classified as a mini-channel (compact heat exchanger). The channels are arranged in a parallel configuration; the mini-exchanger is equipped with an inlet and an outlet, which constitute the inflow to the mini-exchanger and the outflow of the working medium, respectively. This design of the exchanger ensured the boiling of the refrigerant in a volume defined by the dimensions of the shell. The compact heat exchanger was practically flooded with refrigerant liquid at its saturation temperature, which evaporated freely on the surface of the tubes, with the source of heat necessary for the phase change to occur. The loss of liquid due to its evaporation was gravitationally replenished on an ongoing basis. After connecting all its elements, the mini-exchanger's body was subjected to a pressure test for tightness. After 24 h, no leaks in the system were noticed, so tests were started. The mini-exchanger and the remaining installation were thermally insulated with a thermal insulation layer to reduce heat losses. The flow of the working medium was forced by gravity thanks to the flow of heated water through the mini-exchanger pipes. The working medium flowed into the mini-exchanger, where it was heated by seven mini-channels to a temperature exceeding its boiling point, and it changed from the liquid phase to the gas phase. As a consequence of heating, the fluid evaporated in proportion to the level of thermal energy supplied to the tank. This solution regulated the medium's flow rate. After leaving the mini-exchanger, the working medium reached the condenser through the inlet nozzle and then the tank. Then, the homogeneous liquid of the working medium flowed again through the Coriolis flowmeter to measure the mass flow rate of the medium. The movement was driven by a pump that forced the heating water to flow through the mini-channels. The rotation of the pump rotor and the main valve, located before the measuring section, regulated the flow intensity. The pump speed and shut-off valve control ensured a constant water supply with a specified intensity. The experimental tests were carried out with care and accuracy. Before the basic tests, the experimental stand was subjected to verification tests, and the accuracy of the measuring instruments and equipment was checked. The test assumed that the working medium supplied to the exchanger was sufficient and had filled its minimal volume. The water in the mini-channels flowed so steadily that its temperature changed only slightly over the entire flow length (less than 1 K). The cryostat's role was to stabilize the heated water temperature at the inlet and cool it with the working medium during its flow through the measuring section. The tested shell-and-tube mini-exchanger contained seven parallel channels with a straight axis on which the working medium was boiling. The thermal resistance of copper pipes was included in the heat exchange calculations, i.e., Formulas (15) and (16). In the case of heat losses to the environment, it should be stated that the exchanger was very well insulated with a 5 cm thick layer of polyurethane insulation. The estimated heat losses to the environment amounted to about 2% of the exchanger's thermal power. Before the basic tests began, the test stand underwent repeat tests, and the accuracy of the instruments used in it was checked several times. The schematic diagram of the research stand is presented in Figure 2.

The range of parameter changes during the experimental studies is provided in Table 1 (data refer to the refrigerant).

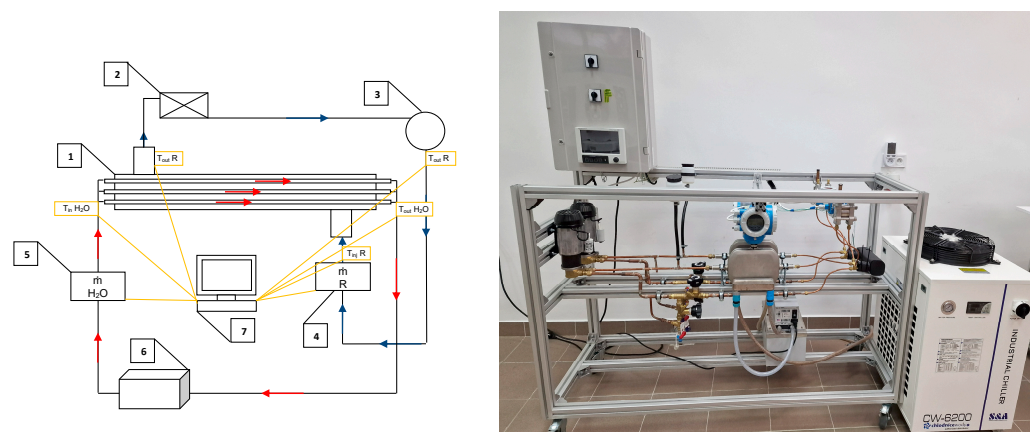


Figure 2. Schematic diagram of the research stand: (1) Compact heat exchanger; (2) refrigerant condenser; (3) liquid refrigerant tank; (4) Coriolis-type flow meter for measuring the flow rate of the refrigerant; (5) Coriolis-type flow meter for measuring the flow rate of the heating medium-water; (6) ultra thermostat for the heating medium-water; (7) data recording device.

Table 1. The scope of the research.

Parameter	Value/Type
Channel's diameter d_i/d_e (mm)	4/6
Mass flow rate G ($\text{kg}\cdot\text{h}^{-1}$)	1–30
Pressure (bar)	1–8
Temperature ($^{\circ}\text{C}$)	25–35
Heat flux density q ($\text{W}\cdot\text{m}^{-2}$)	500–30,000
Refrigerant	R134a; R1234yf; R1234ze; HFE7100

3. Experimental Data

Experimental studies of the boiling process in volume were conducted using four refrigerants: high-pressure R1234yf, R1234ze, and R134a, and low-pressure HFE7100 (3M Poland, Wrocław, Poland). The studies were conducted on a dedicated stand equipped with precise research equipment. The design of the measurement stand enabled the measurement of the thermal and flow parameters of the refrigerant and heating water. The following parameters were measured:

- refrigeration pressure in the system;
- refrigerant temperature at the inlet and outlet of the tested exchanger;
- mass flow rate of the refrigerant in the tested heat exchanger;
- refrigerant temperature in the condenser and liquid tank;
- heating water temperature at the inlet and outlet of the tested exchanger;
- mass flow rate of water in the tested heat exchanger;
- water temperature in the ultra-thermostat.

The studies were conducted in steady-state conditions.

A list of measuring equipment, along with the measuring range and measurement accuracy is presented in Table 2.

Table 2. A list of measuring equipment, along with the measuring range and measurement accuracy.

Measured Value	Device	Measuring Range	Max. Uncertainty
Mass flow	Mass flow meters	0–450 [$\text{kg}\cdot\text{h}^{-1}$]	$\pm 0.15\%$
Absolute pressure	Pressure sensors	0–2500 [kPa]	$\pm 0.05\%$
Temperature	Thermocouples TP-201K-1B-100 (CZAKI THERMO-PRODUCT Sp. z o. o., Raszyn-Rybie, Poland)	−40–+475 [$^{\circ}\text{C}$]	± 0.2 K

Direct measurements of the thermal and flow parameters allowed us to determine the values describing the heat exchange process during refrigerant boiling in the tested heat exchanger. The following values were calculated:

- The heat exchanger's thermal power output:

$$\dot{Q} = \dot{m} \cdot r, \quad (11)$$

where

$\dot{m}$ is the mass flow rate of the refrigerant;

r is the heat of evaporation.

- The heat flux on the heated wall of a channel:

$$q = \frac{\dot{Q}}{A_e}, \quad (12)$$

where

A_e is the outer surface of the seven exchanger tubes. $A_e = 1.58256 \times 10^{-4} \text{ m}^2$.

- Logarithmic mean temperature difference (LMTD):

$$LMTD = \frac{\Delta T' - \Delta T''}{\ln \frac{\Delta T'}{\Delta T''}}, \quad (13)$$

where

$\Delta T'$ is the temperature difference between the water and refrigerant on one side of the exchanger;

$\Delta T''$ is the temperature difference between the water and refrigerant on the other side of the exchanger.

- The overall heat transfer coefficient of the heat exchanger:

$$k = \frac{\dot{Q}}{A_e \cdot LMTD}. \quad (14)$$

- The heat transfer coefficient for boiling the refrigerant was determined from the relationship describing the heat transfer through a cylindrical partition:

$$\dot{Q} = \frac{\pi \cdot l \cdot LMTD}{\frac{1}{h_1 \cdot d_i} + \frac{1}{2 \cdot \lambda} \ln \frac{d_e}{d_i} + \frac{1}{h_2 \cdot d_e}}, \quad (15)$$

where

h_1 is the heat transfer coefficient from the heating water side;

h_2 is the heat transfer coefficient from the boiling refrigerant side;

d_i is the internal diameter of the heat exchanger tube;

d_e is the external diameter of the heat exchanger tube;

λ is the thermal conductivity coefficient of the tube material, i.e., copper, where $\lambda = 370 \text{ W (m}^2 \cdot \text{K)}$:

$$h_2 = \frac{1}{d_e \cdot \left(\frac{\dot{Q}}{\pi \cdot l \cdot LMTD} - \frac{1}{h_1 \cdot d_i} - \frac{1}{2 \lambda} \ln \frac{d_e}{d_i} \right)}. \quad (16)$$

- The value of the heat transfer coefficient from the water side, h_1 , was determined from the Żaworonkow criterion dependence for the transitional movement [50]:

$$Nu = 0.00069 \cdot Re^{1.24} \cdot Pr^{0.5}, \quad (17)$$

which can be used for channels with a circular cross-section in the range of $Re = 2000\text{--}10^4$,

where

$$Nu = \frac{h_1 \cdot d_i}{\lambda}. \quad (18)$$

The heat output of the exchanger $\dot{Q}$, determined from Formula (11), was compared with the heat output value determined from the heating water side. It was found that the differences between these values were not large and fell within the error range of $\pm 2\%$, which was considered satisfactory accuracy.

The calculation results for the key heat and flow quantities of the boiling process in a compact shell-and-tube exchanger are presented below. The values of the heat output of the exchanger, the total heat transfer coefficient, the heat flux density, and the heat transfer coefficient for the boiling of the refrigerant on the heated surface were determined.

Experimental research data concern pool boiling of refrigerants R1234yf, R1234ze, R134a, and HFE7100 in a compact heat exchanger. The research was conducted on a dedicated stand equipped with precise research equipment. The presented results concern key parameters in heat-flow relationships, i.e., the heat transfer coefficient, overall heat transfer coefficient, heat flux density, saturation temperature, saturation, and critical pressure.

Figure 3 shows the dependence of the heat exchanger's thermal power ($\dot{Q}$) on the value of the logarithmic mean temperature difference (LMTD). As can be seen, the value of the heat exchanger thermal power ($\dot{Q}$) strongly depends on the value of the logarithmic mean temperature difference (LMTD). An increase in the LMTD value causes an increase in the heat exchanger thermal power ($\dot{Q}$). The size of the increase in thermal power depends on the type of refrigerant used and its properties. The highest heat exchanger power occurs during the boiling of refrigerant R1234ze at a similar level of LMTD. The lowest value of $\dot{Q}$ was recorded during the flow of the HFE7100 refrigerant. The difference in these values is about 40% concerning the thermal power when using the R1234ze refrigerant.

Figure 4 shows the dependence of the overall heat transfer coefficient k on the refrigerant mass flow rate $\dot{m}$.

The total heat transfer coefficient increases with the increase in the refrigerant mass flow rate $\dot{m}$. The size of the increase depends on the type of refrigerant used and its properties. The highest values of the total heat transfer coefficient k were observed at a mass flow rate of $\dot{m} = 30$ kg/h. A similar trend applies to all refrigerants. The highest k values occurred during the boiling of the hydrofluoroolefin (R1234ze/ yf). The lowest k values occurred during boiling in the volume of the HFE7100 refrigerant.

Based on the calculation relationship (16), the values of the heat transfer coefficient for the boiling of the refrigerant h_2 were determined. The experimental results show that for the same values of heat flux density q , the values differ for the tested refrigerants. In qualitative terms, the nature of the dependence of the heat transfer coefficient value on the heat flux density is similar. This is an exponential relationship of the type commonly used for boiling in volume in conventional systems, as described by Formula (19). Taking the above into account, the relationship was adopted to describe the boiling process in a minivolume, and the values of the coefficients C and n were determined for each of the factors. The calculation results are presented in Table 3.

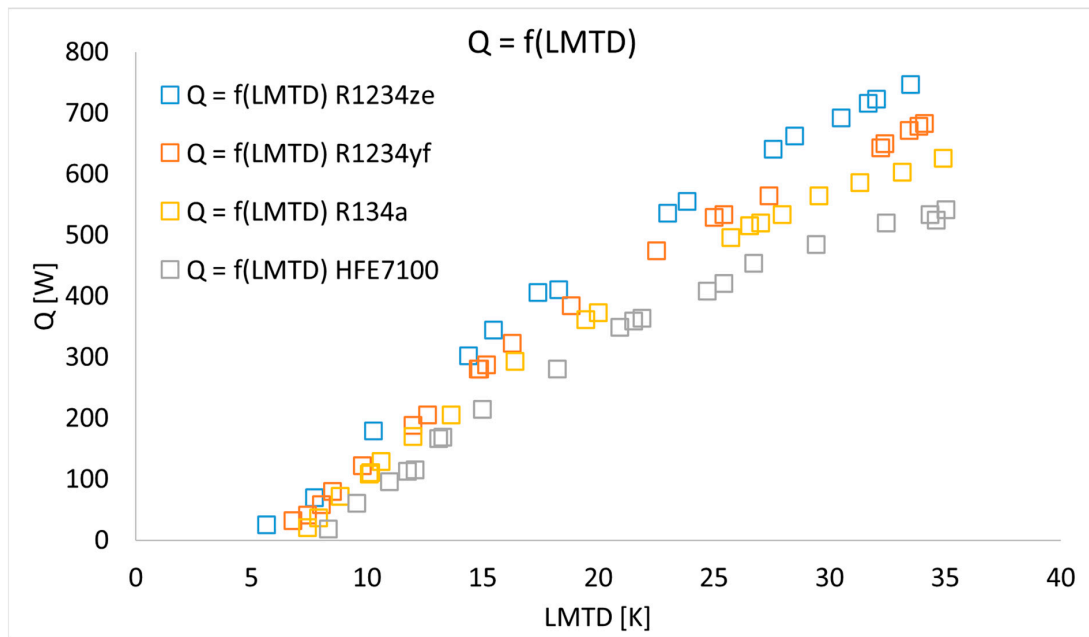


Figure 3. The influence of the logarithmic mean temperature difference (LMTD) on the thermal power of the exchanger ($\dot{Q}$).

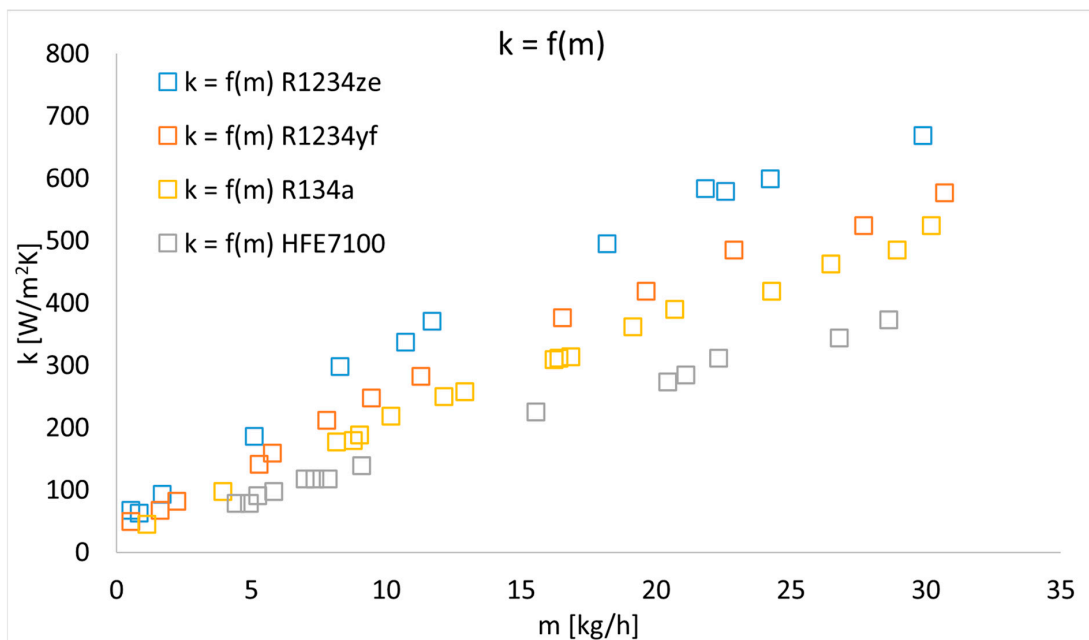


Figure 4. The influence of the refrigerant mass flow rate and the overall heat transfer coefficient.

Table 3. Parameters of prediction equations.

Equation: $h = C \cdot q^n$ (19)				
No.	Refrigerant	C	N	MAPE
1	R1234ze	0.968	0.79	10%
2	R1234yf	0.985	0.74	12%
3	R134a	0.980	0.74	13%
4	HFE7100	0.969	0.72	7%

While the boiling characteristics of refrigerants in a micro-space are similar in quality, they differ in quantity (Figure 5). This results from the different physical properties of these

refrigerants, particularly the different values of working pressure during boiling at similar saturation temperatures. The reference level of working pressure in relation to the critical pressure for a given refrigerant is also essential. This is important when comparing thermal tests for low- and high-pressure refrigerants.

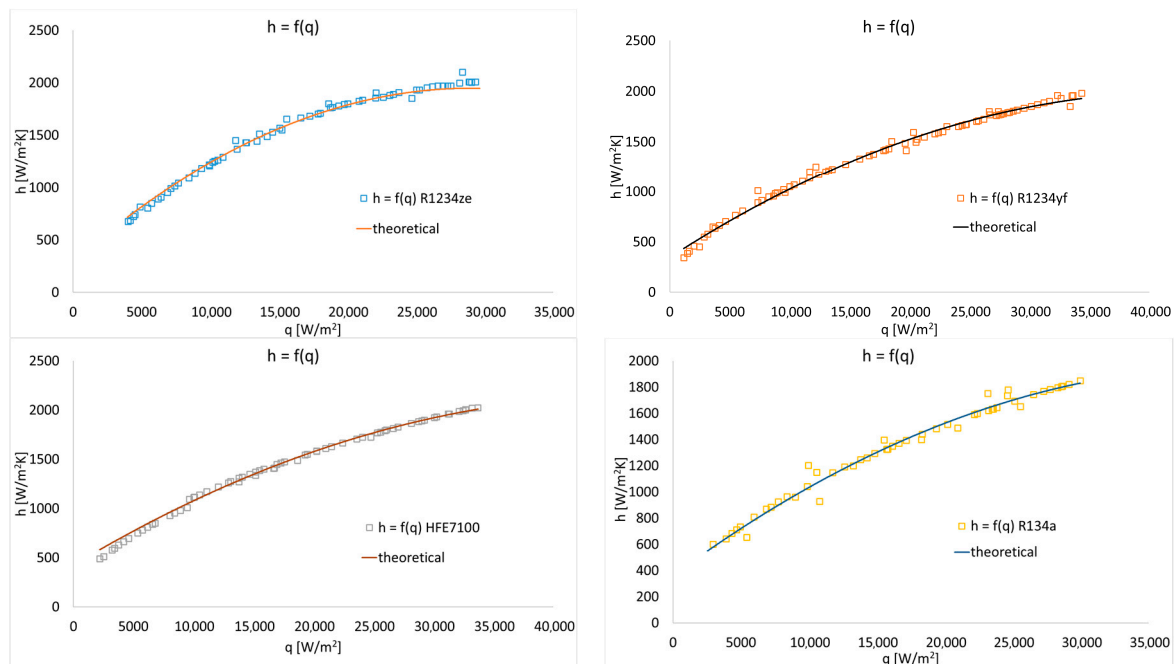


Figure 5. The dependency of the average heat transfer coefficient on the heat flux density q .

Additionally, the experimental data were compared with the data from studies by other authors [17,18] and the commonly used Cooper correlation [32]. The results of the comparison are presented in Figure 6.

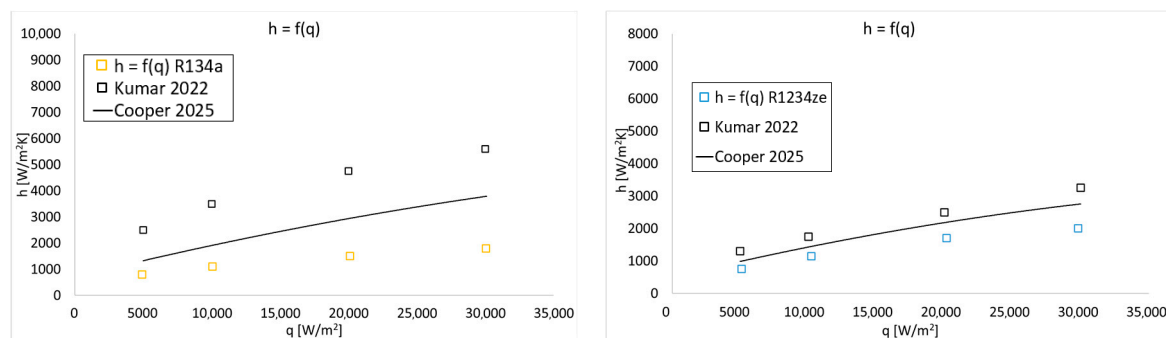


Figure 6. The dependency of the average heat transfer coefficient on the heat flux density q at $t = 30$ °C [18,32].

The comparison shows that the results of the study authors' research differ slightly from the experimental data [17,18] and modeling results [32]. The reason is probably the discrepancy in the number of tested channels. The data presented in the literature concern similar process parameters, but only consider one channel. In the case of the work presented in this study, a tube bundle was tested. Therefore, it is necessary to consider the phenomena occurring between the tube bundle channels, which influence the change in heat exchange intensity and, consequently, other values of the heat transfer coefficient at the same heat flux values.

This paper presents the results of tests for high-pressure (R1234ze, R1234yf, and R134a) and low-pressure (HFE7100) fluids. This means that the physical properties of these fluids differ significantly due to varying pressure values at similar temperatures during the tests. This results from the different relationship of the pressure in the exchanger to the critical pressure for a given fluid. Only the introduction of the reduced pressure to the model, which is the ratio of the current pressure to the critical pressure, allows for relatively unified modeling and comparisons for low- and high-pressure fluids. This can be observed in the graphs drawn in the log p-h coordinate system, which clearly shows the location of the test point in relation to the critical point.

Therefore, it was decided to consider introducing reduced pressure in the form of the ratio of the saturation (working) pressure value to the critical pressure for individual refrigerants, according to the dependence (19). After using the Statistica program, a generalized calculation dependence (20) was obtained that is valid for all four refrigerants (dependence).

$$h = 7.69 \left(\frac{p_s}{p_{cr}} \right)^{-0.11} \cdot q^{0.52}, \quad (20)$$

where

- p_{cr} is the critical pressure;
- p_s is the saturation pressure;
- $C = 7.69$;
- $m = -0.11$;
- $n = 0.52$.

Figure 7 shows the dependence of the average values of the heat transfer coefficient on the heat flux density, as calculated according to the generalized Equation (20).

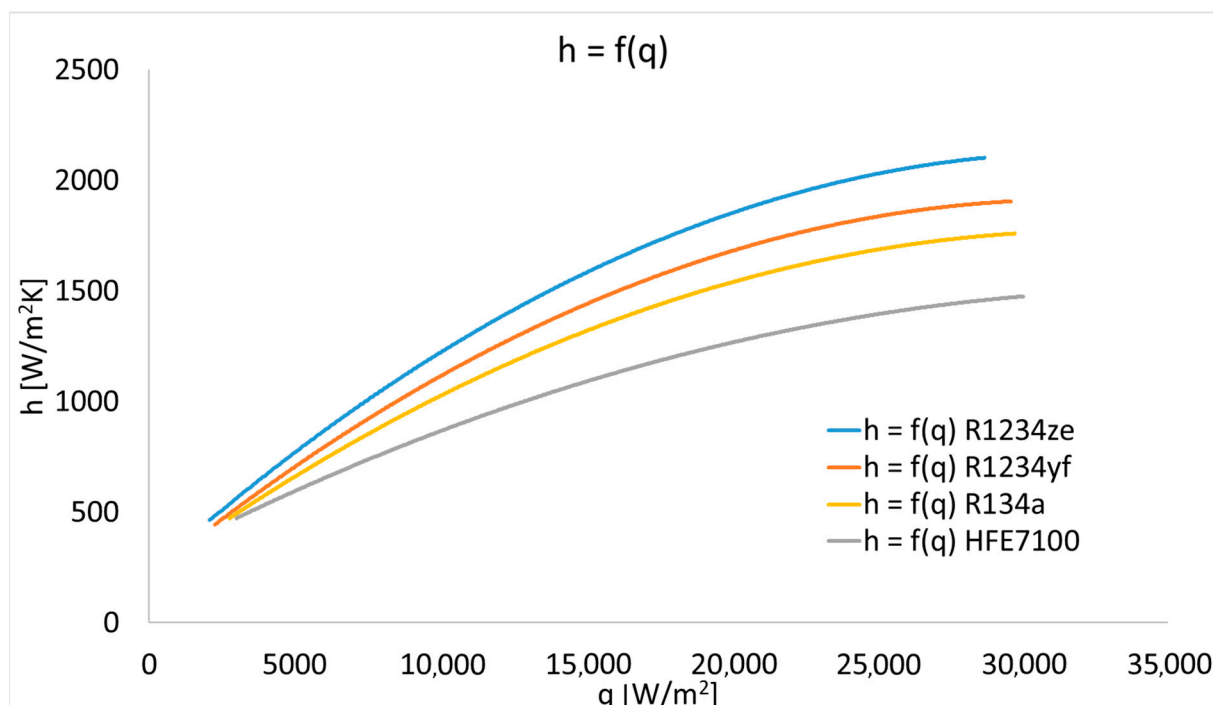


Figure 7. The dependency of the average heat transfer coefficient values on the heat flux density calculated according to Equation (20).

The experimental results were compared with the calculations' results according to Equation (20). As a result of the comparison, a level of high agreement was found between

the modeling results and the experimental results. The mean absolute percentage error is 10.8%, and 99.8% of the discrepancies of the analyzed points are within the range of less than 25% (Figure 8), which is considered satisfactory. The maximum measurement errors of the fundamental thermal and flow parameters are as follows: pressure, $p = \pm 0.05\%$ (max 100 kPa); heat flux, $Q = \pm 2\%$ (max 5 [W]); temperature, $T = \pm 0.2$ K; overall heat transfer coefficient, $k = \pm 3\%$ (max 5.4 [W m⁻² K]); local heat transfer coefficient, $h = \pm 5\%$ (max 50 [W m⁻² K]).

(a)

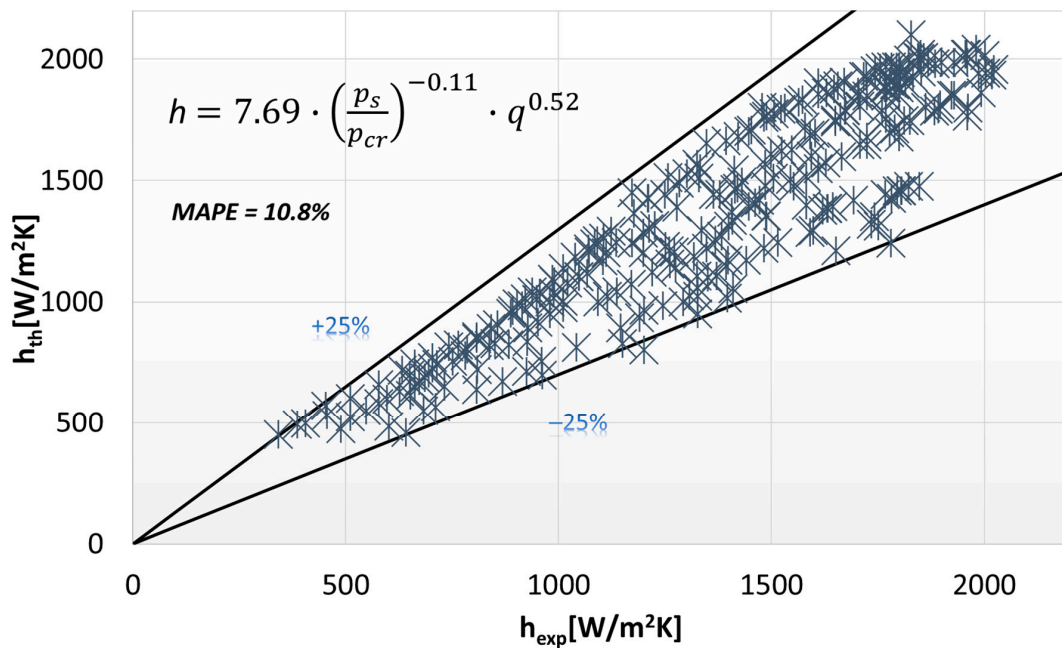
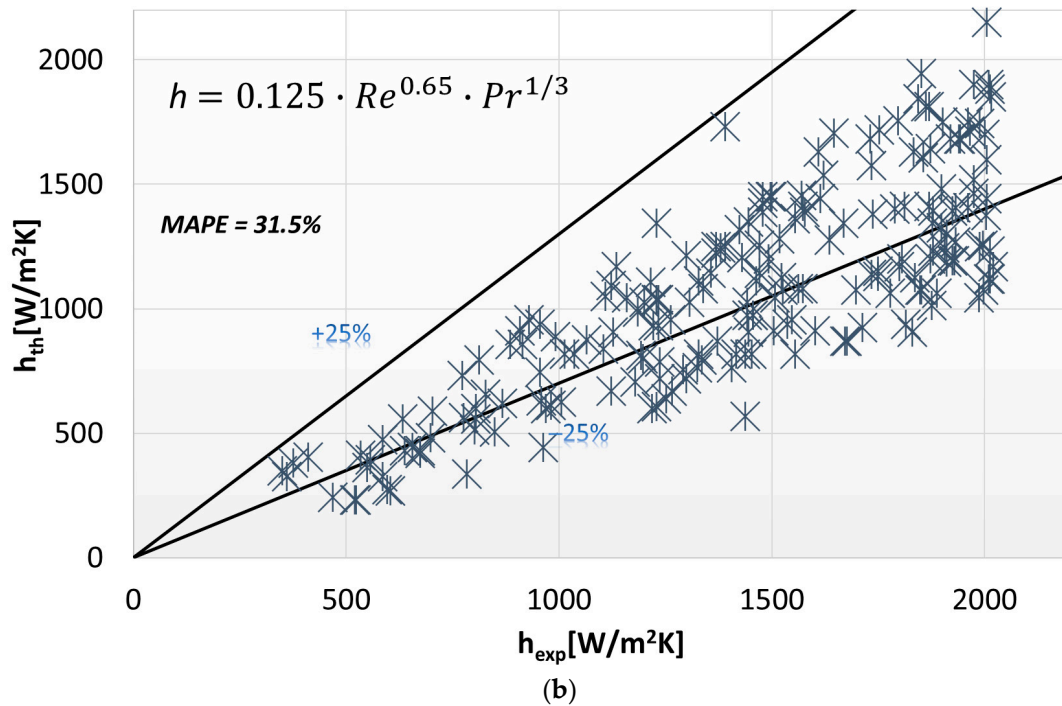


Figure 8. Results of the comparison of the average heat transfer coefficient experimental and calculated values according to (a) (7) and (b) (20).

Figure 8a shows a comparison of the calculation results according to the Kutateladze relationship (7) and the results of our own experimental studies. Figure 8b shows a comparison of the calculation results according to Formula (20) with the results of our own experimental studies.

It is clearly visible that the calculation results according to relationship (7) significantly underestimate the experimental results. The differences in the calculated values of the heat transfer coefficient in this case average 20% compared to the correlation proposed by the authors of the study (20). This situation may result in the fact that, in minispace conditions, the heat transfer coefficient assumes higher values compared to conventional channels. Therefore, it is justified to use correlation (20) to determine the heat transfer coefficient in compact heat exchangers during pool boiling of working media.

The distribution of MAPE error for the entire data population is presented below (Figure 9).

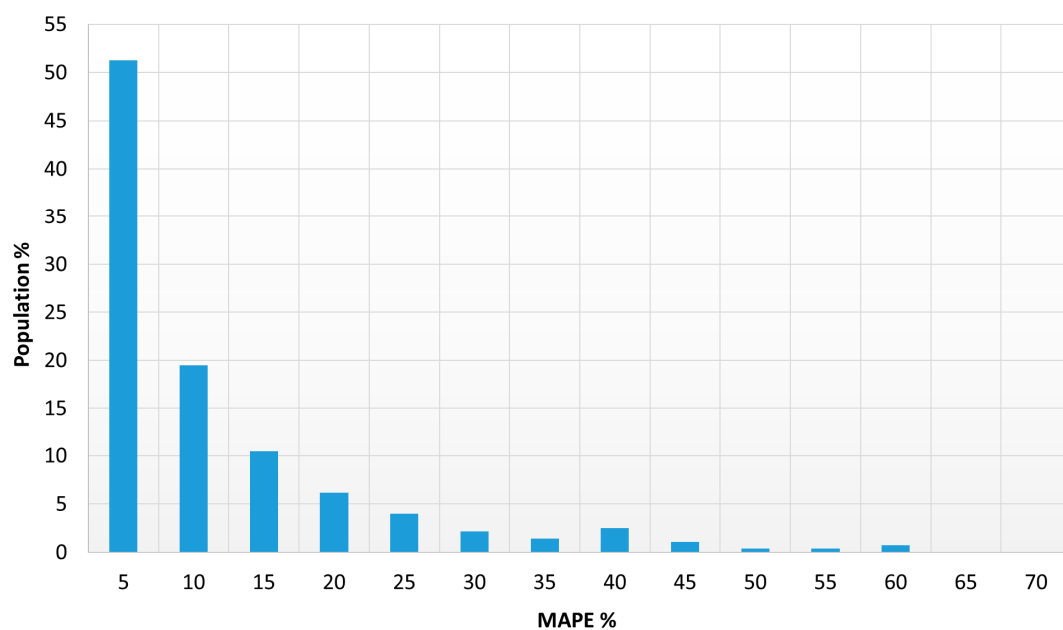


Figure 9. Mean absolute percentage error distribution for the Nusselt number.

4. Conclusions

1. Refrigerants boiling in a compact shell-and-tube heat exchanger were studied. The boiling process occurred in the micro-space of the exchanger shell on the surface of horizontal tubes, which were heated from the inside with warm water. The flow of the refrigerant was gravity-based. The heat exchanger was practically flooded with a liquid refrigerant at saturation temperature, which, after evaporation, flowed out in gaseous form. The studies were conducted for four refrigerants: R1234ze, R1234yf, R134a (a high-pressure refrigerant), and HFE7100 (a low-pressure refrigerant).
2. Based on the obtained results, thermal characteristics were developed describing the heat exchange process in the entire compact heat exchanger and in the case of the boiling process itself.
3. Experimental data were compared to experimental and empirical data presented in other studies. It was found that the results of the study authors' research differ slightly from the experimental data [17,18] and modeling results [32]. This may result in a discrepancy in the number of tested channels. The data presented in the literature concern similar process parameters, but only consider one channel. In the study presented by the authors, a tube bundle was tested. Therefore, it is necessary to take

into account the phenomena occurring between the tube bundle channels, which influence the change in heat exchange intensity and, consequently, other values of the heat transfer coefficient at the same heat flux values.

4. The dependence of the heat exchanger thermal power ($\dot{Q}$) on the logarithmic mean temperature difference (LMTD) was determined. An increase in the LMTD value causes an increase in the heat exchanger thermal power ($\dot{Q}$), and the size of this increase depends on the type of refrigerant used and its properties.
5. The overall heat transfer coefficient increases as the refrigerant mass flow rate increases. The magnitude of this increase is different for each refrigerant. The highest k -values occurred during the boiling of the hydrofluoroolefin (R1234ze/yz), and the lowest were for the HFE7100 refrigerant.
6. The values of the heat transfer coefficient for the boiling of the refrigerant were determined. It was found that for the same values of heat flux density, these values are different for the tested refrigerants. However, in qualitative terms, the nature of the dependence of the heat transfer coefficient value on the heat flux density is similar. This is an exponential dependence, commonly used for boiling in volume in conventional systems, as described by Formula (8). However, the coefficients' C , m , and n values are different for each refrigerant.
7. The proposed calculation models for various refrigerants were generalized by introducing the reduced pressure value in the form of the ratio of the saturation (working) pressure to the critical pressure into the calculation relationship (20). The developed relationship enables the determination of heat transfer coefficient values during boiling in a micro-space on the surface of horizontal tubes for various refrigerants with an accuracy of $\pm 25\%$.

The authors plan to conduct further research using new eco-friendly refrigerants and other compact heat exchanger designs.

Author Contributions: Methodology, T.B.; Software, G.R.; Validation, K.J.W.; Formal analysis, M.K.; Investigation, G.R.; Data curation, K.D. and G.R. All authors have read and agreed to the published version of the manuscript.

Funding: This research received no external funding.

Data Availability Statement: The original contributions presented in this study are included in the article. Further inquiries can be directed to the corresponding author.

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

Nomenclature

A	area (m^2)
d	diameter (m)
G	mass flux density ($\text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$);
h	heat transfer coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)
L	length (m)
$\dot{m}$	mass flow rate ($\text{kg}\cdot\text{h}^{-1}$)
Nu	Nusselt number
q	heat flux density ($\text{W}\cdot\text{m}^{-2}$)
$\dot{Q}$	heat flux (W)
r	heat of condensation/evaporation ($\text{J}\cdot\text{kg}^{-1}$)
Re	Reynolds number
t	temperature ($^{\circ}\text{C}$)
T	temperature (K)

Index

<i>c</i>	condensation, coolant
<i>exp</i>	experiment
<i>e</i>	external
<i>f</i>	fluid
<i>h</i>	hydraulic
<i>i</i>	internal
<i>l</i>	liquid
<i>r</i>	heat of phase change
<i>th</i>	theoretical
<i>s</i>	saturation
<i>w</i>	wall, water

Greek symbols

Δ	difference
λ	thermal conductivity ($\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$)
ν	kinematic viscosity ($\text{m}^2 \text{s}^{-1}$)

Acronyms

HE	heat exchanger
OHTC	overall heat exchange coefficient

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