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## PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

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exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machines

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PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

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# Heat transfer during boiling of environmentally-friendly refrigerant R134a in tube coils

Introduction of new environmentally-friendly refrigerants in refrigerating and air-conditioning systems requires theoretical and experimental research. The paper describes experimental investigations into heat exchange and flow resistance during boiling flow of freon R134a in tube coils – the process that occurs in typical fan air coolers. The obtained results allow the determination of the heat transfer characteristics, including mean values of the refrigerant-side heat transfer coefficient $\alpha$ for the boiling zone in the form of an experiment-based correlation. The correlation lends itself to engineering calculations of air-fin cooller supplied through thermostatic expansion valves where the dryness fraction $x$ of the refrigerant changes during the boiling process from 0.15 to 1.

## Nomenclature

$A$ – coefficient – Eq. (8),
$B$ – coefficient – Eq. (9),
$C$ – coefficient – Eq. (10),
$d$ – diameter, m
$h$ – enthalpy, J/kg
$k$ – overall heat transfer coefficient, W/m$^2$K
Ku – Kutateledze number – Eq. (2)
$l$ – length, m
$L$ – coil length (with elbows), m
Nu – Nusselt number – Eq. (5),
$p$ – pressure, Pa
$q$ – heat flux density, W/m$^2$
$Q$ – heat flux, W
$r$ – heat of vaporisation, J/kg
Re – Reynolds number – Eq. (4),
$T$ – temperature, K
$w\rho$ – mass flux density, kg/m$^2$s
$w$ – velocity, m/s
$X$ – dryness fraction,
$\alpha$ – heat transfer coefficient, W/m$^2$K
$\Delta$ – symbol of difference,
$\delta$ – boundary sub-layer, m
$\lambda$ – thermal conductivity, W/mK
$\mu$ – dynamic viscosity coefficient, Ns/m$^2$
$\rho$ – density, kg/m$^3$.

### Subscripts

$a$ – air,
$exp$ – experimental,
$o$ – boiling,
$s$ – superheat,
$T$ – thermal boundary sub-layer,
$th$ – theoretical,
$i$ – inner diameter,
$o$ – outer diameter,
$'$ – liquid,
$''$ – vapour.

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## 1. Introduction

Small and medium capacity evaporators for cooling liquids and gases (mainly air) are constructed as straight segments of tube coils – parallel finned tubes joined with elbows. The tube coils are fed with a refrigerant from a multi-way distributor or a collecting tube. Distributors are applied in dry evaporators where the refrigerant passes through a thermostatic expansion valve. Collecting tubes are used in flooded evaporators where a refrigerant is recirculated with the aid of a pumping system. The process of boiling that takes place in the tube coils is still a matter of scientific interest [1-3]. This interest is encouraged straight from the market as a rapid development of small and middle-sized enterprises increases the demand for refrigerating systems, and old freons (mainly R12) are superseded by new ecological refrigerants (e.g. R134a, R50a, R404a) to meet new stricter environmental requirements. Moreover, strong competition among producers of refrigerating systems forces them to design and manufacture products of high efficiency and low energy consumption – a goal that makes necessary further fundamental and applied research, including both theoretical and experimental investigations [4-7].

## 2. Physics of the phenomenon

Figure 1 shows a diagram of the boiling process that occurs in a tube coil of a dry evaporator and the interpretation of the changes that the refrigerant undergoes during boiling in the form of a $\ln p - h$ graph. The subcooled refrigerant (point 1), as compared to the saturation temperature, is supplied to the coil through a thermostatic expansion valve. The throttling process ($h = const$, line 1-2) takes place while the refrigerant passes through the valve. During this process, part of the liquid evaporates ($x_2 > 0$) and the temperature of the refrigerant decreases to the evaporating temperature. The remaining part of the liquid evaporates while passing through the coil. Vapour bubbles become bigger, the fraction of the liquid becomes smaller, and finally the flow turns single-phase. In places occupied by the vapour, the inner surface of the evaporator walls is dry, hence the name – dry evaporator. If the amount of the refrigerant is small, the liquid boils out before it reaches the outlet from the evaporator and, consequently, the saturated dry vapour becomes superheated. The extension of the superheated zone (one-phase) can be controlled by changing, for instance, the setting of the expansion valve. On the one hand, the superheated vapour at the outlet from the evaporator favourably prevents flooding of the compressor. On the other hand, with the increasing zone of superheated vapour, the refrigerating capacity decreases due to the decreasing intensity of heat transfer in the convection zone. The average values of heat transfer coefficients referred to the total surface of the evaporator decrease, too. Let us now consider the process of boiling of a refrigerant on its way from the expansion valve, where the dryness fraction ranges between

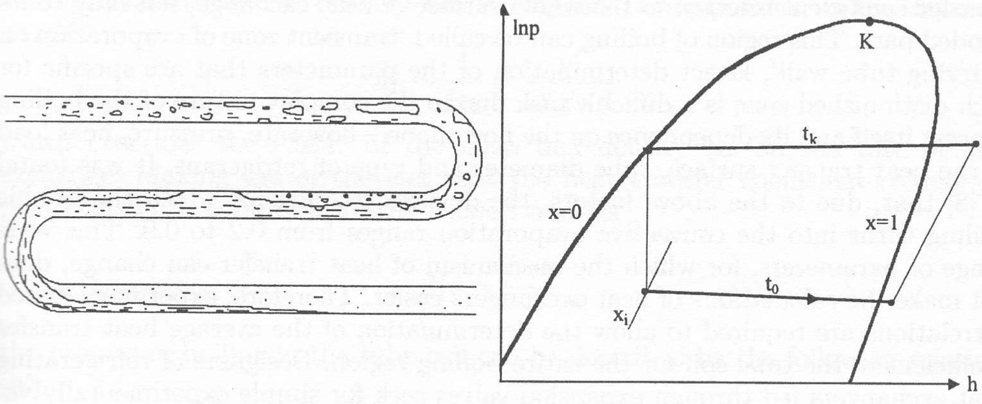


Fig. 1. A diagram of boiling in a tube coil and simplified interpretation of changes of the refrigerant state in an $\ln p - h$ graph.

$x = 0.15 \div 0.3$, to the point where the entire liquid has evaporated, that is $x = 1$. For the time being, let us leave out the process of heat exchange by convection in the superheated vapour at the ultimate part of the tube coil.

The area of boiling is divided into three following zones [8]:

- zone of bubble boiling,
- zone of convective evaporation,
- transient zone of evaporation on a drying tube wall.

The first zone is situated right after the expansion valve where the entire volume of the refrigerant, after expansion and partial evaporation during the throttling process, reaches the saturation temperature corresponding to the saturation pressure in the tube coil. At the entrance to the tube coil, the amount of vapour phase in the two-phase mixture averages from $x = 0.15$ to $x = 0.30$, depending on the temperature and pressure before the expansion valve. Vapour bubbles are generated right at the inside walls of the tube coil, break away from the walls, grow bigger and merge in the flow core. With the growing dryness fraction, the two-phase structure changes from plug (slug) to stratified wavy and then to annular flow. In the case of annular flow the vapour flows at a high velocity through the central part of the tube while the liquid flows along the tube, also at a considerably high velocity, forming a liquid film there. The nature of heat transfer radically changes. The thin liquid film has a small heat resistance, which consequently increases the local heat transfer coefficients. An intensive convection within the liquid film makes it difficult and sometimes prevents the production of vapour bubbles. Thus, the phase transition takes place at the interfacial surface only – this phenomenon is called 'convective evaporation'. Due to evaporation and entrainment of liquid droplets by the vapour phase, the liquid film becomes thinner and first "dry spots" are formed at the tube wall. This takes place at the mass fraction of the vapour phase of about $x = 0.8 \div 0.9$ and manifests itself in a rapid decrease of the heat transfer coefficient referred to the whole surface of heat exchange, not only to its flooded part. This region of boiling can be called 'transient zone of evaporation on a drying tube wall'. Exact determination of the parameters that are specific for each distinuished zone is a difficult task due to the complex nature of the boiling process itself and its dependence on the flow mode – flow rate, pressure, heat load of the heat transfer surface, tube diameter and type of refrigerant. It was found in [8] that, due to the above factors, the dryness fraction for which the bubble boiling turns into the convective evaporation ranges from 0.2 to 0.9. This wide range of parameters, for which the mechanism of heat transfer can change, does not make the calculations of heat exchangers easier. Therefore, experiment-based correlations are required to allow the determination of the average heat transfer coefficient in the tube coil for the entire boiling region. Designers of refrigerating heat exchangers fed through expansion valves seek for simple experimentally validated correlations for new environment-friendly refrigerants as the correlations applied so far usually refer to halogen-derivative refrigerants like R11, R12, R22, R502 during boiling in straight smooth tubes. Correlations given by Bo Pierre, Chawla, Slipcevic, Bogdanow [9-18] are used there.

The author carried out the analysis of the above correlations that describe boiling in tubes for the operational parameters characteristic to dry evaporators in order to determine the effects of particular parameters on the heat transfer coefficient $\alpha$ at the side of the boiling refrigerant. It follows from the analysis, see also [19-20], that the intensity of heat exchange is affected by:

- mass flux density due to forced convection in flow,
- heat flux density connected with the boiling process,
- physical properties of the refrigerant.

During boiling in flow, changes of the fluid temperature are practically negligible, same for the thermal boundary sub-layer $\delta_T$ at the tube wall. Assuming a linear temperature distribution in the boundary sub-layer, it can be written that

$$\text{grad}\ T = \frac{\Delta T}{\delta_T} \ . \tag{1}$$

The temperature difference $\Delta T$ is determined by the density of heat flux $q$ from the channel wall to the boiling liquid. It follows from the dimensional analysis, carried out for instance by Mikielewicz [15], that the quantity $\Delta T$ is a function of the Kutateladze number

$$\text{Ku} = \frac{q}{r \cdot \rho'' \cdot w'} \ , \tag{2}$$

which in a physical sense is a rate of formation of the gas phase referred to the velocity of the liquid

$$w' = \frac{(w\rho)}{\rho'} \ . \tag{3}$$

The thermal sub-layer at the tube wall $\delta_T$ depends, among others, on the Reynolds number

$$\delta_T \sim \frac{1}{\mathrm{Re}} = \frac{\mu'}{w'\rho' \cdot d} \ , \tag{4}$$

which describes the effects of the mass flux density $w\rho$ on the rate of heat exchange. Making use of the fact that the heat transfer coefficient $\alpha$ that depends on $\Delta T$ is comprised in the Nusselt number

$$\mathrm{Nu} = \frac{\alpha \cdot d}{\lambda'} \ , \tag{5}$$

the process of boiling in the tube coil can be described by the following relationship

$$\mathrm{Nu} = f(\mathrm{Re}, \mathrm{Ku}) \ . \tag{6}$$

To sum up, the heat exchange during boiling in flow can be described with the aid of the dimensionless Reynolds and Kutateladze numbers as they include the fundamental quantities characteristic to this process. The explicit form of Eq. (6) will be sought from the experimental investigations.

## 3. Subject of investigations

The heat exchanger contains the finned tube pipes. Two tube coils, being the main elements of an air-fin cooler were used in experimental investigations. The cooler was also equipped with:

- two axial fans,
- fan casing that formed the aerodynamic channel,
- two-way distributor of the refrigerating liquid.

The heat exchanger under investigations consisted of a typical finned block made of 44 copper tubes of outer/inner diameter $d_o/d_i = 0.016/0.014$ m, and length $l = 0.64$ m each, arranged in a bundle of tubes in a chessboard array. Aluminium fins of thickness $\delta = 0.00025$ m were fitted to the outer surface of the tubes. The fin density of the heat exchange surface was 164 fins/m. Horizontal tube sections were connected with elbows to form two equal tube coils of length $L = 17.85$ m each, (active length without elbows 14.20 m), fed with the refrigerant through a thermostatic expansion valve of type TEN2 produced by Danffos and a distributor of the liquid (the outer diameter of supply tubes was $d_o = 0.006$ m). A collecting tube at the outlet from the tube coils was $d_o = 0.018$ m in diameter. The finned block was situated on the suction side of the axial-flow fans at a significant distance from their blading. Owing to that, a uniform stream of air was secured to flow through the finned block. The velocity of air flowing through the finned block was adjusted by throttling flaps at the forcing side of the fans. Changing the air velocity, and also the temperature difference between the air and the boiling refrigerant allowed us to carry out investigations at various rates of heat transfer. The characteristics of the air cooler are given in Tab. 1 and its schematic diagram is presented in Fig. 2.

Table 1. A list of dimensions characterising the evaporator under investigation

| No. | Parameter | Denotation | Dimension | Value |
|---|---|---|---|---|
| 1 | Finned block | | | |
| | – length, | $L$ | m | 0.64 |
| | – width, | $a$ | m | 0.25 |
| | – height. | $b$ | m | 0.36 |
| 2 | Fins | | | |
| | – thickness, | $\delta \cdot 10^3$ | m | 0.25 |
| | – total number, | $n_f$ | – | 105 |
| | – material. | – | – | Al |
| 3 | Tubes | | | |
| | – outer diameter, | $d_o$ | m | 0.016 |
| | – inner diameter, | $d_i$ | m | 0.014 |
| | – number of tubes, | $n_t$ | – | 44 |
| | – coil length without elbows, | $L_r$ | m | 14.20 |
| | – coil length with elbows, | $L_w$ | m | 17.85 |
| | – material. | – | – | Cu |
| 4 | Total size of the finned block | $V_{bl}$ | $m^3$ | 0.06 |

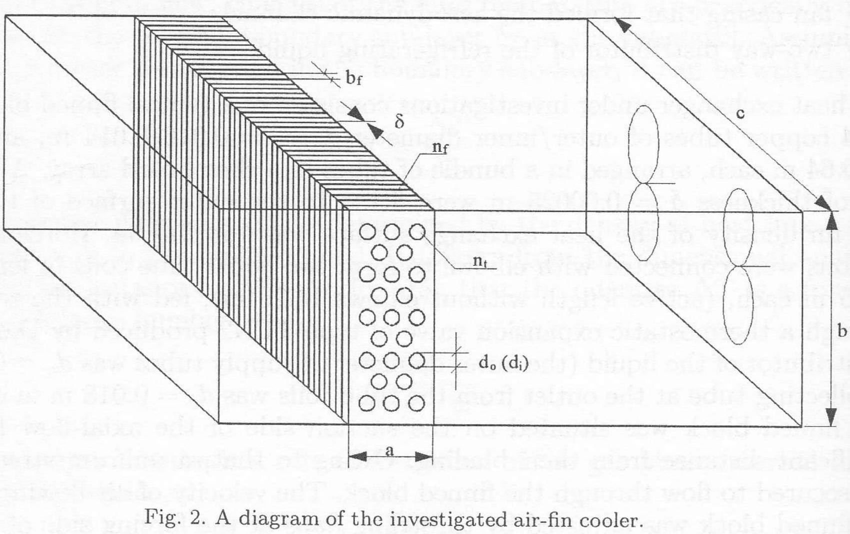


Fig. 2. A diagram of the investigated air-fin cooler.

## 4. Experimental facility

Figure 3 shows a diagram of the experimental facility. The air-fin cooler with tube coils under investigation was placed in a rectangular aerodynamical channel. The air flow was secured by two axial fans. The channel with the cooler inside was placed in an insulated refrigerating chamber. The chamber was equipped with the system stabilising the air conditions – an auxiliary refrigerating unit and heaters controlled by a chamber thermostat.

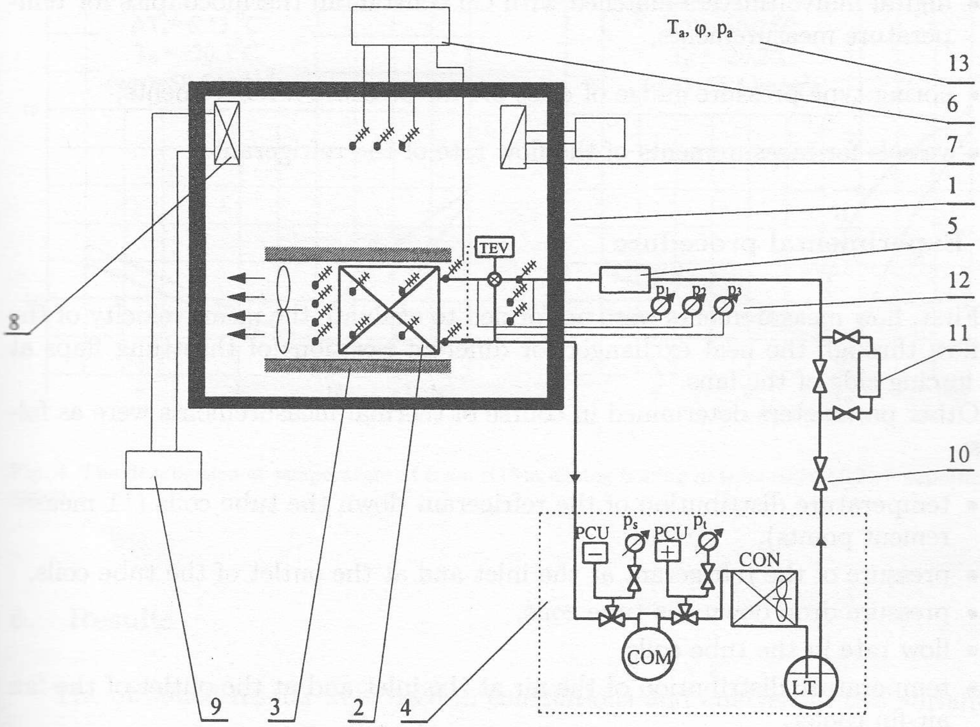


Fig. 3. A schematic diagram of the experimental facility. 1 – insulated refrigeration chamber, 2 – air-fin cooler, 3 – experimental channel, 4 – refrigerating unit for R134a with additional equipment (COM-compressor, CON – air cooled condenser, LT – liquid tank, PCU – pressure control units, $p_t, p_s$ – suction and forcing pressure), 5 – flow meter system, 6 – temperature measuring system, 7 – air heater, 8 – auxiliary evaporator, 9 – auxiliary refrigerating unit, 10 – control valve, 11 - dehydrating filter, 12 – pressure transducers, 13 – ambient parameters system.

The flow of the refrigerant R134a was secured by a compressor-condenser unit. Auxiliary measuring tanks were also used for calibration of the electronic flow meter SONOFLO by Danfos to measure the volume flow rate of the refrigerant. The measurements of quantities that describe the operation of the considered refrigeration unit were computerised, or could be performed by means of a traditional measuring technique.

A computer-aided measurement system comprised the following elements:

- interface for temperature measurements of type LC-015-1612 matched with thermocouples of Cu-constantan,
- interface for voltage measurements of type LC-011-0812 matched with pressure and pressure difference transducers, and the flow meter.

The traditional measuring equipment included:

- digital milivoltmeters matched with Cu-constantan thermocouples for temperature measurements,
- spring-type pressure gauge of class 0.4 for pressure measurements,
- vessels for measurements of the flow rate of the refrigerant.

## 5. Experimental procedure

First, flow measurements were performed to evaluate the mean velocity of the air flow through the heat exchanger for different positions of throttling flaps at the forcing side of the fans.

Other parameters determined in course of thermal measurements were as follows:

- temperature distribution of the refrigerant down the tube coils (11 measurement points),
- pressure of the refrigerant at the inlet and at the outlet of the tube coils,
- pressure drop down the tube coils,
- flow rate in the tube coils,
- temperature distribution of the air at the inlet and at the outlet of the fan air-fin cooler,
- parameters of the refrigeration cycle such as the pressure at the suction and forcing side of the compressor as well as pressures and temperatures at chosen points of the cycle.

Temperature of the medium was measured in the taps located at the bends of coiled tube. Temperature was measured with the accuracy of $\pm 0.05^o$C.
The obtained results enabled the determination of:

- thermal power of the finned coil heat exchanger,
- local and mean values of the overall heat transfer coefficient $k$ for the heat exchanger,
- heat transfer coefficient $\alpha$ at the side of the boiling refrigerant.

Examples of temperature distributions along the tube coils are presented in Fig. 4. The analysis of these distributions allows us to distinguish the zone of boiling from the zone of forced convection where the vapour becomes superheated. In the boiling zone, the temperature of the refrigerant slightly decreases due to the pressure drop along the tube coils. After the evaporation is completed the temperature substantially increases and heating up of the vapour phase takes place.

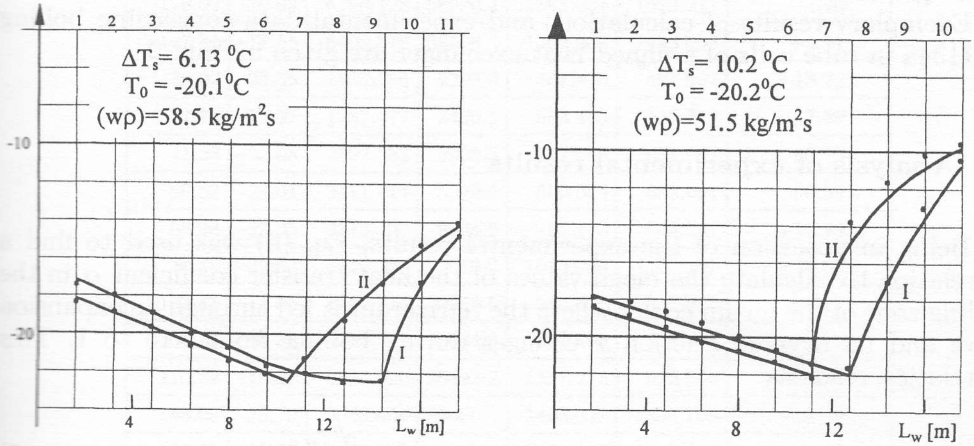


Fig. 4. The distribution of temperature of freon R134a during boiling in tube coils. ($\Delta T_S$ – superheat of vapour).

## 6. Results

The obtained results were used in calculations and analysis in two variants:

a) applied investigations – the calculation results were presented with reference to the whole heat exchanger. As a result, the average quantities describing the heat transfer process, in the boiling and superheating zone together, were obtained;

b) fundamental investigations – the calculation results referred to the boiling zone (of two-phase flow of the refrigerant).

The results of applied investigations of the refrigerating unit in a range of operational conditions, similar to usual service conditions, are reported in a separate work [21], and will not be discussed in the present paper.

The chapter to follow comprises results of basic investigations that allow the determination of the heat transfer coefficient $\alpha$ in the boiling zone of the tube coil.

Values of the heat transfer coefficient $\alpha$ at the refrigerant side were determined based on experimental results and calculations of the overall heat transfer coefficient $k$ for the boiling zone. The coefficient $k$, or its reverse, comprises the heat resistance on both sides of the wall. At the air side, the heat resistance was calculated from the relationship of Gogolin [22] valid for finned surfaces of heat exchange. This formula was extensively tested to achieve good effects and had been applied in many practical cases by the author of the present paper, showing good agreement with the experimental results, see [23].

Exemplary results of calculations and experimental data concerning boiling of R134a in tube coils of a finned heat exchanger are given in Tab. 2.

## 7. Analysis of experimental results

Being in posession of the experimental results, Eq. (6) was used to find a correlation to calculate the mean values of the heat transfer coefficient $\alpha$ in the boiling zone of the air-fin cooler where the refrigerant is fed through an expansion valve and its dryness fraction $x$ changes during boiling from 0.15 to 1. This correlation reads as

$$\mathrm{Nu} = 19.2 \cdot (\mathrm{Re})^{0.8} \cdot (\mathrm{Ku})^{0.6} \cdot \left(\frac{\rho''}{\rho'}\right)^{0.5}, \tag{7}$$

The last term $(\frac{\rho''}{\rho'})^{0.5}$ in the above equation includes the direct effect of material properties of the refrigerant (whereas the indirect effects are comprised in the dimensionless numbers). Exponents of dimensionless numbers Re and Ku are assumed to be equal to the corresponding ones determined for boiling phenomena and recommended in the literature [6,12,18]. Coefficient 0.5 for the density ratio and constant 19.2 have been approximated by means of the least square method.

The correlation (7) was developed for the following range of dimensionless numbers:

$$\mathrm{Re} = 600 \div 12000$$

$$\mathrm{Ku} = 0.002 \div 0.0900.$$

Substituting the dimensionless numbers with physical quantities, the correlation (7) can be rewritten in a simpler form

$$\alpha = A \cdot (w\rho)^{0,2} \cdot q^{0,6} \cdot d^{-0.2} \quad [\mathrm{W/m^2K}] \tag{8}$$

where:

Table 2. An example of experimental and calculation results for the boiling process of R134a in a finned coil heat exchanger

| $w\rho$ [kg/m$^2$s] | $T_0$ °C | $q_i$ [W/m$^2$] | $\Delta p_{exp}$ [Pa] | $\alpha_{exp}$ [W/m$^2$K] | $\alpha_{th}$ [W/m$^2$K] | $\frac{\alpha_{th}-\alpha_{exp}}{\alpha_{exp}} \cdot 100$ |
|---|---|---|---|---|---|---|
| 178.02 | -10.02 | 6106.128 | 58990.4 | 1588.631 | 1519.211 | -4.40 |
| 147.81 | -12.55 | 6402.579 | 38500.2 | 1402.568 | 1493.003 | +6.41 |
| 36.32 | -26.56 | 1731.786 | 2208.8 | 567.470 | 491.962 | -13.22 |
| 37.78 | -27.07 | 1987.477 | 2406.5 | 583.463 | 537.759 | -7.89 |
| 41.33 | -23.05 | 2652.583 | 2908.3 | 737.655 | 658.997 | -10.70 |
| 66.08 | -23.01 | 2411.414 | 8322.5 | 680.064 | 683.691 | +0.59 |
| 68.08 | -22.05 | 3373.197 | 8204.5 | 755.113 | 843.747 | +11.79 |
| 71.30 | -22.05 | 5819.223 | 9055.2 | 1040.289 | 1181.191 | +13.56 |
| 102.43 | -13.06 | 3162.313 | 20965.5 | 946.605 | 907.073 | -4.12 |
| 106.65 | -18.33 | 3280.339 | 20004.6 | 890.403 | 918.523 | +3.15 |
| 110.32 | -18.88 | 3723.020 | 20493.2 | 1197.234 | 995.956 | -16.79 |
| 158.08 | -25.79 | 5093.464 | 50660.5 | 1455.527 | 1264.109 | -13.79 |
| 160.31 | -20.38 | 6055.942 | 47809.6 | 1367.787 | 1430.143 | +4.53 |
| 177.60 | -15.33 | 6344.307 | 54003.2 | 1713.772 | 1526.009 | -10.97 |
| 178.09 | -14.98 | 6718.500 | 55670.3 | 1477.771 | 1582.110 | +7.04 |
| 184.15 | -15.36 | 7872.045 | 56709.1 | 1666.190 | 1749.369 | +4.98 |
| 193.76 | -10.45 | 10143.103 | 62051.7 | 1761.153 | 2092.154 | +18.80 |
| 194.14 | -10.31 | 11607.279 | 62239.0 | 2737.110 | 2270.440 | -17.06 |
| 198.37 | -12.36 | 7437.510 | 65504.3 | 1840.504 | 1733.564 | -5.81 |
| 198.04 | -13.40 | 6026.615 | 66093.2 | 1447.571 | 1522.118 | +5.11 |
| 28.06 | -26.76 | 1251.005 | 10043.6 | 346.537 | 384.169 | +10.66 |
| 32.43 | -27.01 | 1529.744 | 1670.8 | 437.436 | 445.852 | +2.04 |
| 45.76 | -21.15 | 1323.802 | 3655.8 | 459.532 | 445.868 | -2.83 |
| 45.02 | -20.06 | 2213.901 | 3688.5 | 667.172 | 607.153 | -9.00 |
| 59.47 | -18.67 | 2361.641 | 6588.1 | 766.017 | 670.278 | -12.53 |
| 84.83 | -17.49 | 2462.824 | 12498.4 | 758.178 | 740.811 | -2.37 |
| 88.31 | -16.15 | 3175.581 | 13440.6 | 962.237 | 873.676 | -9.15 |
| 122.02 | -10.46 | 3557.055 | 25004.2 | 856.924 | 1017.100 | +18.67 |
| 124.53 | -10.02 | 4128.528 | 25078.8 | 1292.819 | 1118.453 | -13.47 |
| 135.24 | -12.45 | 4269.707 | 30633.6 | 1082.130 | 1150.575 | +6.38 |
| 137.00 | -11.46 | 4145.653 | 33032.2 | 1045.998 | 1137.171 | +8.70 |

$A$ – a coefficient comprising the properties of the refrigerant as a function of the evaporation temperature $T_0$ [°C],
$q$ – heat flux density, referred to the inside surface of the tube in the boiling zone, [W/m$^2$],
$w\rho$ – mass flux density, [kg/m$^2$s],
$d$ – inner diameter of the tube coil, [m].

Equation (8) is suggested for use in engineering calculations of the heat exchangers for refrigerating systems and allows the determination of mean values of the heat transfer coefficient $\alpha$ inside tube coils during evaporation of an ecological refrigerant R134a within $x = 0.15 \div 1$. The form of Eq. (8) is similar to other correlations for boiling-in-flow calculations [9,16-18]. However, unlike those correlations it comprises:

- the effects of boiling conditions in tube coils on the process of heat exchange in typical air-fin cooler fed with the refrigerant through an expansion valve,
- the effects of the boiling temperature $T_0$ (boiling pressure $p_o$) on the heat exchange rate (via a coefficient $A$).

Equation (8) obtained for the new ecological refrigerant was compared with the correlation of Bogdanow [18] used in calculations of the heat transfer coefficients during boiling of halogen-derivative refrigerants (R11, R12, R22, R113, R142) in smooth horizontal tubes

$$\alpha = B \cdot \frac{\dot{m} \cdot \dot{q}^{0.6}}{d^{0.6}} \ [\mathrm{W/m^2K}] \ , \tag{9}$$

where:
$\dot{m}$ – mass flux of the refrigerant, [kg/s],
$B$ – dimensional coefficient dependent on the refrigerant properties and evaporation temperature $T_0$.

Substituting the mass flux of the refrigerant $m$ [kg/s] with the mass flux density $(w\rho)$ [kg/m$^2$s], a new form of Eq. (9) can be obtained

$$\alpha = C \cdot (w\rho)^{0,2} \cdot q^{0.6} \cdot d^{-0.2} \ \ \mathrm{W/m^2K} \tag{10}$$

where $C$ is a dimensional coefficient in Bogdanow's formula.

Understandably, the coefficients $A$ and $C$ in Eqs. (8) and (10) differ in value. As an example, values of $A$ in Eq. (8) for R134a and $C$ in Eq. (10) for R12 are given in Fig. 5. Values of $A, B, C$ are gathered in Tab. 3.

A comparison of results of experimental investigations and predictions based on the correlation (7) is presented in Fig. 6, showing a good agreement within the range of ±20%. A comparison of calculations based on the correlation (7) and results of experiments concerning the heat transfer coefficient in two variants $\alpha = f(q)$ for $w\rho = const$ and $\alpha = f(w\rho)$ for $q = const$, respectively for $T_0 = -10$°C and $T_0 = -20$°C is shown in Figs. 7 and 8. The computational and experimental results agree well in the considered range of parameters.

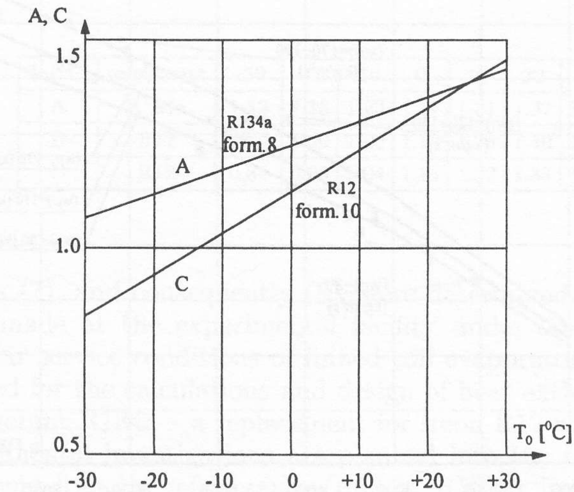


Fig. 5. Dimensional coefficients in Eq. (8) for R134a and in Eq. (10) for R12 as a function of the boiling temperature.

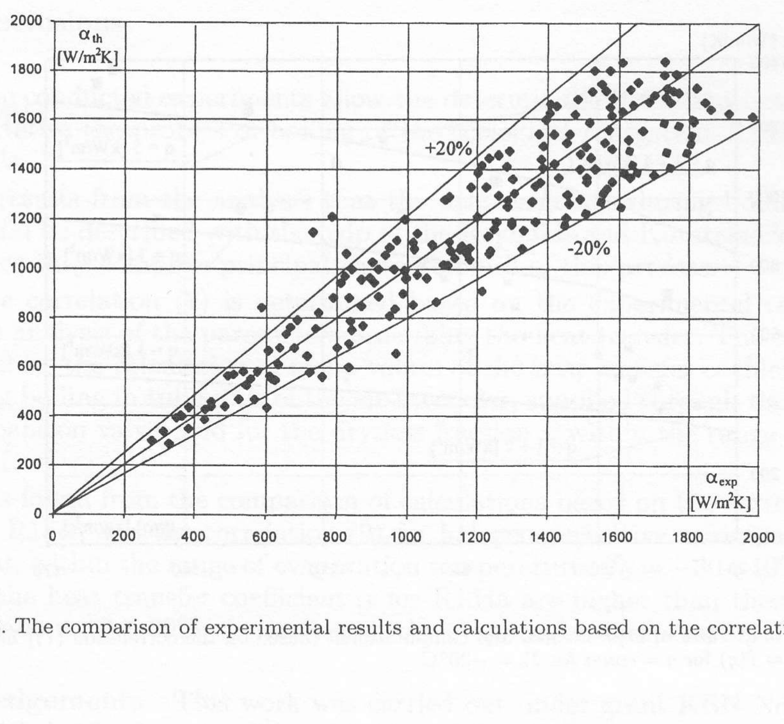


Fig. 6. The comparison of experimental results and calculations based on the correlation (7).

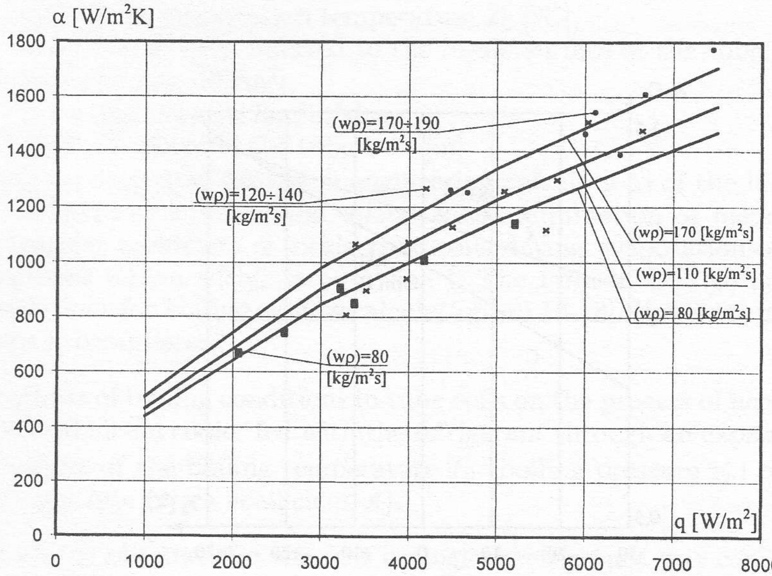


Fig. 7. The comparison of experimental and computational (based on the correlation (7)) heat transfer coefficient $\alpha = f(q)$ for $w\rho = const$ for $T_o = -10^oC$.

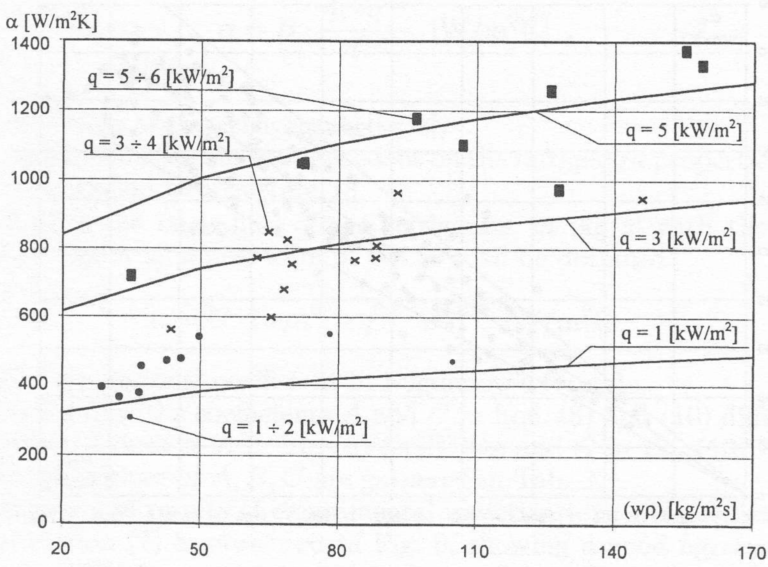


Fig. 8. The comparison of experimental and computational (based on the correlation (7)) heat transfer coefficient $\alpha = f(q)$ for $q = const$ for $T_o = -20^oC$.

Table 3. Values of coefficients $A, B, C$ in Eqs. (8)-(10) as a function of the boiling temperature $T_0$

| $T_0$[°C] | refrigerant | −30 | −20 | -10 | 0 | 10 | 20 | 30 |
|---|---|---|---|---|---|---|---|---|
| A | R134a | 1.12 | 1.18 | 1.23 | 1.27 | 1.31 | 1.37 | 1.43 |
| B | R12 | 0.883 | 1.00 | 1.09 | 1.18 | 1.28 | 1.40 | 1.54 |
| C | R12 | 0.84 | 0.95 | 1.04 | 1.13 | 1.22 | 1.33 | 1.47 |

Correlations (7), and consequently (8), were determined on the basis of 163 measurements made at the experimental facility under operational conditions similar to regular service conditions of finned coil evaporators. Therefore, it can be recommended for the calculations and design of heat exchangers fed with the ecological refrigerant R134a – a replacement for freon R12 widely used so far.

More experimental investigations are planned into the process of boiling of other environment-friendly refrigerants (R404a, R507a) in tube coils in order to obtain correlations similar to Eq. (7) with possible corrections of numerical coefficients.

## 8. Conclusions

1. The conducted experiments allow the determination of quantities that characterise the process of boiling of the ecological refrigerant R134a in tube coils.
2. It results from the analysis that the heat exchange during boiling in flow could be described with the help of the Reynolds and Kutateladze numbers since they comprise principal characteristics of this process.
3. The correlation (7) is determined based on the experimental results and the analysis of the parameters describing the heat transfer. The correlation enables the calculation of mean values of the heat transfer coefficient $\alpha$ during boiling in tube coils of the air-fin cooler, supplied through thermostatic expansion valves and for the dryness fraction $x$ within the range from 0.15 to 1.
4. It is found from the comparison of calculations based on the correlation (7) for R134a and the correlation (9) for halogen-derivative refrigerants (R12) that, within the range of evaporation temperatures $T_0 = -30 \div 10^o$C, values of the heat transfer coefficient $\alpha$ for R134a are higher than those for R12 by about $10 \div 30\%$, depending on the evaporation temperature $T_0$.

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## Wymiana ciepła podczas wrzenia ekologicznego czynnika chłodniczego R134 w wężownicach rurowych

**Streszczenie**

Wprowadzenie nowych ekologicznych czynników w chłodnictwie i klimatyzacji wymaga realizacji badań teoretycznych i eksperymentalnych. W pracy przedstawiono wyniki badań eksperymentalnych wymiany ciepła i oporów przepływu podczas wrzenia czynnika R134a w wężownicach rurowych typowych dla wentylatorowych chłodnic powietrza. Uzyskane wyniki pozwoliły wyznaczyć wielkości opisujące proces wrzenia, w tym współczynniki przejmowania ciepła $\alpha$ od strony wrzenia czynnika chłodniczego. Dla strefy wrzenia opracowano korelację empiryczną, która może być wykorzystywana do obliczeń inżynierskich chłodnic wentylatorowych zasilanych poprzez termostatyczne zawory rozprężne przy zmianie stopnia suchości $x$ czynnika chłodniczego od 0.15 do 1.