Experimental and theoretical analysis of the local condensation heat transfer in a plate heat exchanger

V Grabenstein¹ and S Kabelac²,³

¹Institute for Thermodynamics, Helmut Schmidt University, University of the Federal Armed Forces Hamburg, Germany

²Institute for Thermodynamics, Leibniz University Hannover, Germany

kabelac@ift.uni-hannover.de

Abstract. Plate heat exchanger (PHE) are today widely used in industrial heat transfer applications due to their good thermal performance, modest space requirement, easy accessibility to all areas and their lower capital and operating costs as compared to shell-and-tube heat exchangers. Although authoritative models for the design of PHE used as condensers are missing, the number of applications where a PHE is operating as a condenser increases. On the way to a reliable model based on physical approaches for the prediction of heat transfer and pressure drop during the condensation process inside a PHE, the flow and heat interactions as well as their dependence on the geometrical parameters of the corrugated plates and the operating conditions must be studied in detail. In this work the stepwise procedure for the fundamental construction of such a model is described. An experimental setup was built to analyze the characteristics of the two-phase-flow in PHE. A single gap, consisting of two transparent corrugated plates, was tested with a two-phase flow of air/water and also with boiling refrigerant R365mfc. Flow pattern maps were constructed for plates with corrugation angles of 27 and 63 degrees relative to the direction of flow. Investigations of the local heat transfer coefficients and the pressure drop were done with the same plates. The measurement of the local heat transfer coefficients was carried out by the use of the “Temperature Oscillation InfraRed Thermography” (TOIRT) method. Based on these results three main flow patterns are defined: film flow, bubbly flow and slug flow. For each of the three flow patterns an own model for the heat transfer and pressure drop mechanism are developed and the heat transfer coefficient and the friction factor is calculated with different equations depending on the actual steam quality, mass flow and geometrical parameters by means of a flow pattern map. The theory of the flow pattern based prediction models is proved with own experimental data. The measurements were carried out with an experimental setup in a technical scale. The refrigerant cycle works with R134a as refrigerant and involves two PHEs, used as condenser and evaporator, and a 55 kWel compressor for the compression of the vapor phase. The setup allows the measurement of quasi-local heat transfer coefficients inside the PHEs. Additional heat exchangers assure saturated vapor at the inlet and saturated liquid at the outlet of the condenser.

1. Introduction
Plate heat exchangers (PHE) were originally developed for single-phase applications in the food industry, like the pasteurization of milk. Today this kind of heat exchanger is widely used for any heat transfer problems, not only in single-phase but also in two-phase applications. The considerable growth of installed PHE as single-phase-apparatus, evaporator and condenser in the last decade is pushed by the many advantages of PHEs compared to shell-and-tube heat exchanger, for example the
high heat transfer rate at a compact size, the low content of working fluid, easy access to the heat exchanging plates for cleaning and substituting and the low costs for standard applications.

The high heat transfer rates in PHE are achieved by the enlarged surface, dominating at low Reynolds-numbers, and the very high turbulence due to the small flow passages and the complex flow pattern at high Reynolds-numbers. The strong eddies also causes a high pressure drop and due to the related high local shear fouling is minimized. The operating conditions of the gasket type of PHE is restricted of applications with temperatures less than 180°C and a pressure below 2.5 MPa for the apparatus with polymer gaskets, brazed plate heat exchanger (BPHE) are able to work at higher temperature and pressure.

The general problem concerning PHE used as condenser or evaporator is the sizing and rating of such apparatus, because yet no reliable correlation for the calculation of the heat transfer and the friction pressure drop during the phase changing in PHE are published. Although some correlations are presented in the literature, these equations are often simple and fitted to empirical data of a single experimental setup, so the correlations are only able to generate reliable results for one type of plate heat exchanger (geometrical parameter) or working fluid. In Figure 1 six correlations published in the literature for the heat transfer in a PHE during the condensation process are compared with own experimental results. It can be seen that three correlations calculate the Nußelt number with an error of up to 70%. The correlations from Yan et al. [1] and Shi et al. [5] fits the experimental data best, but the experimental database for the correlation of Shi et al. was done at the same experimental setup as the data for the experimental Nußelt number and the geometrical parameter from the test setup by Yan et al. are quite similar to the own setup.

The aim of the current project is to evaluate the phenomenological connection between the heat transfer, the friction pressure drop and the two-phase flow pattern inside the corrugated gap of a plate heat exchanger. For this part an experimental setup with a single corrugated gap was designed and the adiabatic two-phase flow inside the gap explored. The experimental results from the single gap are compared with those from a PHE working as a condenser in a compression refrigeration cycle in a technical scale. With the knowledge from both setups the understanding of the two-phase characteristic in PHE should be improved and together with a comprehensive literature database about the condensation in PHE and other compact heat exchanger a flow pattern based model for the heat transfer and pressure drop in PHE will be derived. The basic procedure from the experimental measurements to the complex condensation model is given in Figure 2.

![Figure 1: Comparison of own experimental results of condensation of R134a in technical scale PHE and six in the literature cited correlations of Yan et al. [1], Han et al. [2], Jokar et al. [3], Kuo et al. [4], Shi et al. [5] and Akers et al. [6]](image-url)
Steinhagen - phase flow is homogeneous and typical industrial-air two-phase flow in a PHE phase and the gas assure th

Evenly distributed over the whole inlet gap. The corrugated gap with the transparent plates and the flow distribution. Special adapted inlet and outlet manifolds and eight mixing sections with both for their experiments, Winkelmann [7] published visualization experiments working with a water-air two-phase flow. Because of different ranges of the mass flux it is very difficult to compare the results of these works.

For the determination of the local heat transfer and pressure drop in relation to cycle were built up. Both experimental setups are briefly described below.

For the determination of the local heat transfer and the pressure drop in relation to a specific flow pattern and visualization a single adiabatic corrugated gap of a PHE was built up. Typical industrial plates were used as a casting model and the resulting transparent polyurethane plates were used in the setup. The inlet- and outlet triangular areas of these plates were cut off to avoid possible influence on the flow distribution. Special adapted inlet- and outlet manifolds and eight mixing sections with separate access for the liquid-phase and the gas assure that the two-phase flow is homogeneous and evenly distributed over the whole inlet gap. The corrugated gap with the transparent plates and the manifolds with the scheme of the setup can be seen in Figure 3. Both gas and liquid phase are fed

Figure 2: Procedural method to establish a reliable heat transfer and pressure drop model for the condensation inside PHE based on local and integral experimental data and a literature database

2. Literature

Three publications are known which visualized the two-phase flow in plate heat exchanger. All three published visualization experiments working with a water-air two-phase flow. Vlasogiannis et al. [7] and Tribbe and Müller-Steinhagen [8] used hard ($\phi \sim 60^\circ$), soft plates ($\phi \sim 30^\circ$) and the mixing of both for their experiments, Winkelmann [9] visualized only the water-air two-phase flow in a PHE with soft plates.

3. Experimental setup

To get a reliable understanding of the two-phase interactions during the condensation process in a PHE a lot of experimental data is needed. As explained in Figure 2 a setup with a single adiabatic corrugated gap for the determination of the local two-phase phenomena as well as a setup for quasi-local and integral heat transfer and pressure drop of a PHE as condenser in a compression refrigerant cycle were built up. Both experimental setups are briefly described below.

For the determination of the local heat transfer and the pressure drop in relation to a specific flow pattern and visualization a single adiabatic corrugated gap of a PHE was built up. Typical industrial plates were used as a casting model and the resulting transparent polyurethane plates were used in the setup. The inlet- and outlet triangular areas of these plates were cut off to avoid possible influence on the flow distribution. Special adapted inlet- and outlet manifolds and eight mixing sections with separate access for the liquid-phase and the gas assure that the two-phase flow is homogeneous and evenly distributed over the whole inlet gap. The corrugated gap with the transparent plates and the manifolds with the scheme of the setup can be seen in Figure 3. Both gas and liquid phase are fed

Figure 3: Experimental setup of the adiabatic corrugated gap, left side: scheme of the experimental setup, right side: photo of the transparent gap and the inlet and outlet manifold
separately from a tank to the mixing sections to get the possibility of measuring pressure, temperature and flow rate of every phase and calculating the resulting vapour quality. The two-phase flow is collected in the outlet-manifold and led back to the tank where it separates into the single phases. The cycle is closed and by adjusting the saturation temperature of the working fluid inside the tank the saturation pressure can be regulated. As a working fluid the refrigerant R365mfc is used because of the relative high normal boiling point (\( T_{\text{NBP}} = 40.1^\circ\text{C} \)), at ambient laboratory temperature (25°C) the saturation pressure is below the ambient pressure. This pressure difference between the ambient and gap is very important for the mechanical stability. The pressure of the two-phase flow and the pressure drop over the corrugated passage are measured. As second two-phase flow the mixture from air and water is used to analyze the flow patterns inside the corrugated gap. The surface tensions and the resulting contact angle between the liquid phase and the solid wall has an influence on the wettability and the flow pattern. Experiments were done to determine the contact angles between all in the experiments used working fluids and plate materials. The change in contact angle between the transparent polyurethane plates and the common stainless steel plates is negligible for the refrigerant R365mfc. The results of the contact angle measurements, the experimental setup and the results of the visualization are explained in detail in [10] and [11].

The experimental refrigerant system with the PHE as a condenser in a technical scale is sketched in Figure 4. The setup encloses one primary and three secondary fluid loops. The primary fluid is the refrigerant R134a, the secondary fluid for the cooling of the condenser, the desuperheater and the subcooler is de-ionized water, and for heating the evaporator a water-glycol mixture is used to prevent freezing. The desuperheater and the subcooler before and after the condenser ensure saturated liquid and gaseous refrigerant at the entrance and exit of the condenser, it can be assumed that no deheating or subcooling takes place inside the condenser. The wall surface and the fluid temperatures are measured at seven locations along the channel of the secondary fluid. These measured temperatures were used to form an energy balance on six control volumes and finally calculate a quasi-local heat transfer coefficient at the refrigerant side as a function of the vapour quality. For the compression of the superheated refrigerant vapour coming from the separator a 55 kW el six-cylinder compressor is used that allows for working with either two, four or all six cylinders, thus achieving different volume flow rates in the refrigerant loop. Two oil separators in row assume that the oil content in the

![Figure 4: Experimental setup of the compression refrigerant cycle in a technical scale](Image)
refrigerant is kept lower than 0.02%, measured before the expansion valve. More information about the experimental setup can be found in [12].

In a first step in both setups, corrugated gap and PHE as condenser in a technical scale, plates with a corrugation angle of 63° are used, later the influence of other geometrical parameters, especially the corrugation angle will be studied.

4. Data evaluation

The identified flow pattern and the local heat transfer and pressure drop inside the corrugated gap are given in terms of the superficial velocity of every phase

\[ u_{kj} = \frac{\dot{V}_j}{A_f} \quad \text{with} \quad A_f = b B_p \quad (j=l: \text{liquid}; j=g: \text{gas}) \tag{1} \]

which is the theoretical velocity if one phase is passing the gap as a single phase only. It can be calculated by dividing the volume flow rate \( \dot{V}_j \) by the cross section area \( A_f \), which is nearly the product of the channel spacing \( b \) and the width of the plate \( B_p \). The plates used for the corrugated gap have a mean channel spacing of 3 mm, a width of 0.388 m and a corrugation angle relative to the direction of flow of 63°.

The determination of the local heat transfer coefficients inside the corrugated was done with the “Temperature oscillation InfraRed Thermography” (TOIRT). This method works with a periodical oscillating heat flux into a wall, introduced by a 15 W laser, and the measuring of the surface temperature. The heat flux and the surface temperature response oscillating with the same frequency but the temperature has a phase delay to the heat flux. With a three-dimensional finite difference model of the wall it is possible to calculate with an inverse iteration the local heat transfer coefficients on the opposite wall. Freund and Kabelac [13] described the method in detail.

The calculation of quasi-local heat transfer coefficients along a plate are based on the measurements of the wall surface and fluid temperature of the secondary fluid, the mass flow rates of the primary and secondary fluid through the PHE and the refrigerant pressure. For every control volume \( i \) the heat flux \( \dot{q}_i \) can be calculated by

\[ \dot{q}_i = \dot{m}_w \ c_w(T) \left( T_{w,i-1} - T_{w,i} \right) / (2 A_i) \tag{2} \]

where \( \dot{m}_w \) is the mass flow rate of the cooling water, \( c_w(T) \) the temperature depending specific heat capacity and \( A_i \) the heat transfer area at one side of the this control volume. For the determination of the heat transfer coefficient on the refrigerant side the refrigerant wall temperature is calculated from the measured wall temperature on the water side

\[ T_{r,\text{wall},i} = T_{w,\text{wall},i} - \frac{\dot{q}_i s_p}{k_p} \tag{3} \]

with the plate thickness \( s_p \) and the thermal conductivity of the plate material \( k_p \). The quasi-local heat transfer coefficient at the refrigerant side then results as

\[ h_{r,i} = \frac{\dot{q}_i}{T_{r,\text{wall},i} - T_{r}^{\text{sat}}(p_{r,i})} \tag{4} \]

where \( T_{r}^{\text{sat}}(p_{r,i}) \) represents the saturation temperature of the refrigerant as a function of the pressure at this location. To substitute the location depending with a vapor depending heat transfer coefficient, the energy balance equation can be used to determine the change of vapor quality inside the control volume

\[ \Delta x_i = \frac{2 \dot{q}_i A_i}{m_r \Delta \bar{h}_r T_r^{\text{sat}}(p_{r,i})} \tag{5} \]

with the specific enthalpy of vaporization of the refrigerant.
The frictional pressure drop is calculated based on the assumption that the total pressure difference over the PHE, which can be measured, is the summarization of four single pressure drop phenomena: the pressure drop inside the manifold $\Delta p_{\text{man}}$, the acceleration pressure difference due to the increasing specific volume during the condensation process $\Delta p_{\text{acc}}$, the gravitational pressure difference $\Delta p_{\text{gr}}$, and the frictional pressure drop $\Delta p_f$. The frictional pressure drop comes than to

$$\Delta p_f = \Delta p_{\text{exp}} - \Delta p_{\text{man}} + \Delta p_{\text{acc}} + \Delta p_{\text{gr}}. \quad (6)$$

Kakac and Liu [14] suggest for the pressure drop in the inlet and outlet manifold

$$\Delta p_{\text{man}} = 1.4 \frac{G_{\text{man}}^2}{\rho_m} \quad (7)$$

with the mass flux in the manifold $G_{\text{man}}$ and the mean density $\rho_m$ of the two-phase flow. The acceleration and gravitational pressure difference is calculated with a homogeneous model of Collier [15]

$$\Delta p_{\text{acc}} = \frac{G^2 \Delta x}{\rho_m} \quad (8)$$

$$\Delta p_g = \rho_m g \Delta z \quad (9)$$

with the mass flux in the plate gap $G$, the change in the vapor quality $\Delta x$, the gravitational acceleration $g$ and the height between inlet and outlet of the PHE $\Delta z$. The mean density is calculated with the saturated density of the liquid and gaseous phase

$$\frac{1}{\rho_m} = \frac{x}{\rho_l} + \frac{1-x}{\rho_g}. \quad (10)$$

5. Measurement uncertainties
Careful calibration gives an estimate on +/- 8% for the local heat transfer coefficients measured with the TOIRT-method. A thorough uncertainty analysis according to Moffat [16] for the determination of the quasi-local and integral heat transfer coefficients was done by Djordjevic et al. [12]. The results shows uncertainties for the heat transfer coefficients of 15% and for the pressure drop of 20%.

6. Experimental results
The experimental results are divided into three parts: the two-phase flow pattern visualization inside the adiabatic corrugated gap, the local heat transfer and pressure drop of the two-phase flow inside the corrugated gap in conjunction with the flow pattern and the quasi-local and integral heat transfer and pressure drop during the condensation process inside the PHE in a technical scale.

6.1. Two-phase flow pattern
By the visualization of the two phase flow in a corrugated gap three clearly identifiable flow pattern are found, characterized and classified in a flow pattern map: Bubbly flow, film flow and slug flow. The bubbly flow occurs at high fractions of the liquid phase. The liquid phase transports the gaseous phase in small bubbles which follow the corrugation of the plates, when the flow rate of the gas is small the flow direction of the bubbles is more directional downstream. With rising mass fluxes of the gas phase the number of bubbles increases and the size decreases because of higher shear forces and collisions at the contact points of the plates. The film flow develops at low rates of liquid flow and high rates of vapour. The film flow can be characterized by a small liquid film running down on the surface of the walls whereas the vapour is streaming between these films downward. The velocity of the gaseous phase is much higher than the liquid phase so that slip between both phases occurs and shear forces make the interface wavy and turbulent. This effect increases with rising vapour flow rate.
The slug flow appears at high flow rates of the liquid and the vapour phase. The slug flow can be identified by fast moving areas alternately of a churn flow with a lot of bubbles in a liquid phase and a film flow with a very small rate of liquid flow or only vapour flow. The areas extended over the whole plate width and have sharp fronts direct to the direction of flow. Between two flow pattern transition regimes occur which are mixtures of the two adjacent flow pattern. The boundaries of the classification are floating and subjective. In Figure 5 the flow pattern map of the two-phase flows air/water and liquid/gaseous R365mfc in the adiabatic corrugated gap with a corrugation angle of 63° are presented.

6.2. Local heat transfer and pressure drop and the dependency on flow pattern
The local heat transfer coefficients and the pressure drop depending on the flow pattern are measured for an adiabatic two phase flow in the corrugated gap. The local heat transfer coefficients are measured at seven different points distributed over the plate width, then they are averaged. In figure 6 these local heat transfer coefficients are given according to the superficial velocities of the liquid and the vapour of R365mfc in a corrugated gap with a corrugation angle of 63° as a function of the vapour and liquid superficial velocity, the measurements with the same flow pattern are outlined.
phase. The plot of the heat transfer rate shows a tendency to increase with the flow rates of both phases. For a reliable prediction about the interaction of the flow pattern and the two phase heat transfer coefficient more experimental data is needed, work is in progress on this behalf. The frictional pressure drop of the adiabatic two-phase flow inside the corrugated gap in the context of the superficial velocities of the liquid and the vapor phase is plotted in figure 6. The pressure drop is clearly increasing with the flow rate of the liquid phase but only little increasing with a rising flow rate of the gas. With rising superficial velocity of the vapor the frictional pressure drop is reaching a plateau and begins to decreasing after the vapor flow rate is increased further. The change in the two-phase flow pattern can be one reason for this non-linear behavior as can be seen in the rising of the frictional pressure drop in the film flow regime compared with the falling frictional pressure drop in the slug flow regime at constant superficial velocity of the liquid phase.

6.3. Quasi-local and Integral heat transfer and pressure drop

In Figure 7 the quasi-local heat transfer coefficients during the condensations process in a whole technical PHE are plotted as a function of the vapour quality and the mass flux and the vapour quality and the saturation pressure respectively. Both graphs illustrate the decreasing of the heat transfer coefficient with the vapour quality and the progressing condensation process. An increasing of the mass flux can raise the heat transfer whereas higher saturation pressures lowers the heat transfer.

The frictional pressure drop during the condensation process inside a plate heat exchanger is plotted in Figure 8 as a function of the mass flux and the saturation pressure. The frictional pressure drop

![Figure 7: Quasi-local heat transfer coefficients as function of the vapour quality x, the mass flux G and the saturation pressure p\text{sat} during the condensation process of R134a in a PHE at constant saturation pressure p\text{sat} of 510 kPa (left side) and constant heat flux q of 8.5 kW/m² and mass flux G of 30 kg/(m²s) (right side)](image)

![Figure 8: Frictional pressure drop as a function of the mass flux G and the saturation pressure p\text{sat} during the complete condensation process of R134a in a PHE; the refrigerant is saturated at inlet and outlet of the PHE)](image)
drop behaves equivalent to the heat transfer, both increasing with rising mass flux and saturation pressure. The frictional pressure drop is doubled by increasing the mass flux from 40 to 60 kg/m²s at a saturation pressure of 520 kPa, the condensation heat transfer coefficient does not change as much.

7. Conclusions and outlook
Based on the characterisation of the two-phase flow in the corrugated gap for every of the three identified flow patterns a model for the prediction of the heat transfer and the pressure drop shall be derived and proofed with own experimental and literature data.

7.1. Film flow
For the film flow an adapted model of the film wise condensation model of Nußelt [17] is suggested. The basic form of the analytical Nußelt model is as follows

$$\text{Nu}_{\text{lam, film}} = \frac{L}{k_1} \left[ \frac{\varphi \rho L (\rho_l - \rho_g) k_1^2 \Delta \varphi}{\mu (T_{\text{sat}} - T_{\text{wall}}) L} \right]^{1/4}$$

with the density of the liquid and gas phase $\rho$, the viscosity $\mu$ and the length of the condensation film $L$. The model should be expanded for the effects of the increasing turbulence in the condensate film and the wave formation of the film surface due to the fast streaming gas phase. Especially the two phase flow in PHE offer a high level of turbulence. Approaches to implement these enlargements are explained by Carey [18] and in the Baehr-Stephan [19]. Another equation for the heat transfer in the turbulent condensate film with Reynolds numbers higher than 400 is suggested by Isashenko [20] based on the energy- and momentum equation. The local heat transfer can be calculated as

$$\text{Nu}_{\text{x, tur}} = 0.0325 \text{Re}^{1/4} \Pr^{1/2}$$

for Reynolds numbers between 400 and $7 \cdot 10^7$ and Prandtl numbers between 1 and 25. The author quotes the Reynolds number of the condensate as

$$\text{Re} = \left[ 89 + 0.024 \frac{\text{Pr}^{1/2}}{\text{Pr}_0} \left( \frac{\text{Pr}}{\text{Pr}_0} \right)^{1/4} (Z - 2300) \right]^{4/3}$$

with $\text{Pr}$ as the Prandtl number at saturation temperature and $\text{Pr}_0$ at wall temperature. In addition we have

$$Z = \frac{c_p L (T_{\text{sat}} - T_{\text{wall}})}{\Delta \varphi} \frac{1}{\text{Pr}} \frac{x}{\left( \frac{\rho_l}{\rho} \right)^{1/3}} \text{Pr}_0 \space (y)$$

All correlations have to be proved with more experimental and literature data to find an approach for the laminar and turbulent film heat transfer and pressure drop.

7.2. Bubbly flow
For the bubbly flow a homogeneous model is suggested, described by Carey [18], which assumes no slip between the liquid and the vapour phase and which treats the two-phase flow as a single phase flow through the plates. The fluid properties of the single phases are weighted and averaged and then used as apparent fluid properties for the two phase flow. The new fluid properties are used with single phase correlations to predict the heat transfer and pressure drop.

7.3. Slug flow
The modelling of the slug flow is more difficult because of the non steady state character of the flow. The periodic change of a film flow and a churn flow region makes it necessary to average two different flow models depending on the frequency and length of the slugs.
The measurements will be continued to fill up the databank. The models will be further developed according to Figure 2.

8. Acknowledgement
This project is founded by the German Research Council (Deutsche Forschungsgemeinschaft DFG) with the project number KA 1211/21

References
[1] Yan YY, Lio HC and Lin TF 1999 Condensation heat transfer and pressure drop of refrigerant R134a in a plate heat exchanger Int. J. Heat and Mass Transfer 42 993-1006
[2] Han DH, Lee KJ, and Kim YH 2003 The characteristics of condensation in brazed plate heat exchangers with different chevron angle J. of Korean Phys. Soc. 43 66-73
[3] Jokar MH, Hosni MH and Eckels SJ 2006 Dimensional analysis on the evaporation and condensation of refrigerant R134a in minichannel plate heat exchangers Applied Thermal Engineering 26 993-1006
[4] Kuo WS, Lie YM, Hsieh YY and Lin TF 2005 Condensation heat transfer and pressure drop of refrigerant R410A flow in a vertical plate heat exchanger Int. J. Heat Mass Transfer 48 5205-5220
[5] Shi ZY, Chen JP, Grabenstein V and Kabelac S 2010 Experimental investigation on condensation heat transfer and pressure drop of R134a in a plate heat exchanger Heat Mass Transfer 46 1177-1185
[6] Akers WW, Dean HA and Crosser OK 1959 Condensing heat transfer within horizontal tubes Chemical Engineering Progress Symposium Series 55 (29) 171-176
[7] Vlasogiannis P, Karagiannis G, Argyropoulos P and Bontozoglou V Air-water two-phase flow and heat transfer in a plate heat exchanger 2002 Int.J. of Multiphase Flow 28 757-772
[8] Tribbe C and Müller-Steinhagen HM 2001 Gas/liquid flow in plate-and-frame heat exchangers – part II: Two-phase multiplier and flow pattern analysis, Heat Transfer Eng. 22 12-21
[9] Winkelmann D 2010 Condensation of pure refrigerants and their zeotropic mixtures in plate heat exchangers Diss. Technische Universität Berlin
[10] Grabenstein V and Kabelac S 2010 Experimental investigations and modelling of condensation in plate heat exchangers Int.Heat Transfer Conf. (Washington DC, USA, 08-13 August 2010) ASME
[11] Grabenstein V and Kabelac S 2010 Strömungsformen in zweiphasig durchströmten Plattenwärmeübertragern DKV-Tagung (Magdeburg, Germany, 08-13 August 2011) DKV (Deutscher Kälte- und Klimatechnische Verein)
[12] Djordjevic EM, Kabelac S and Serbanovic SP 2008 Heat transfer coefficient and pressure drop during refrigerant R134a condensation in a plate heat exchanger Chemical Papers of the serbian society of science 62 78-85
[13] Freund S and Kabelac S 2010 Investigation of local heat transfer coefficients in plate heat exchangers with temperature oscillation IR thermography Int. J. of Heat Mass Transfer 53 3764-3782
[14] Kakac S and Liu H 1998 Heat Exchangers Selection, Rating and Thermal Design (Boca Raton: CRC Press)
[15] Collier JG 1982 Convective Boiling and condensation, (McGraw-Hill) 2nd ed.
[16] Moffat RJ 1988 Describing uncertainties in experimental results Experimental Thermal Fluid Science 1 3-17
[17] Nübelt W 1916 Die Oberflächenkondensation des Wasserdampfes, VDI-Z. 60 569-575
[18] Carey VP 2008 Liquid-vapor phase-change phenomena (New York:Taylor & Francis) 2nd ed.
[19] Baehr HD and Stephan K 2010 Wärme- und Stoffübertragung (Berlin: Springer) 7th ed.
[20] Isashenko VP 1977 Heat transfer during condensation (russ.) Energia Moscow