РОЗДІЛ 3 ХОЛОДИЛЬНІ ТА СУПУТНІ ТЕХНОЛОГІЇ

УДК 536.24

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MEASUREMENT AND CORRELATION OF FLOW BOILING HEAT TRANSFER OF R600a/COMPRESSOR OIL SOLUTION INSIDE A HORIZONTAL SMOOTH TUBE

Experimental results of local heat transfer coefficients for the boiling of real working fluids (solutions of R600a with mineral naphthenic oil ISO VG 15) in smooth tube with small diameter (5.4 mm) are presented. The tests were carried out for the inlet pressure in the range from 71.1 kPa to 77.9 kPa, heat flux was 3800 W/m^2, and mass velocity of working fluid was from 14.75 to 18.36 kg/(m^{2}s). The quantitative estimation in reduction of heat transfer coefficient of the wetted surface in evaporator at high oil concentration in the mixture is made. The influence of mass velocities of the working fluid on the values of the local heat transfer coefficients are analyzed. Based on the results obtained it was observed that increasing mass velocity leads to increase the local heat transfer coefficient of RWF both on side of wetted perimeter and vapor phase. The equation for the modeling of the local heat transfer coefficient for boiling of isobutane/compressor oil solution flow in the pipe is suggested.
Keywords: *Heat transfer coefficient – Real work fluid –Boiling – Isobutan – Refrigerant/oil solution – Vapor quality – Concentration.*

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ВИМІРЮВАННЯ І МОДЕЛЮВАННЯ КОЕФІЦІЄНТА ТЕПЛОВІДДАЧІ ПРИ КИПІННІ РОЗЧИНІВ R600a/КОМПРЕСОРНЕ МАСТИЛО ВНУТРІ ГОРИЗОНТАЛЬНОЇ ГЛАДКОЇ ТРУБИ

В статті приведені результати експериментального дослідження локальних коєфіцієнтів тепловіддачі при кипінні реальних робочих тіл (розчини холодоагента R600a з мінеральним нафтеновим мастилом ISO VG 15) в гладкій трубі невеликого діаметру (5.4 мм). Дослідження проводились при тиску в випарнику в діапазоні від 71.1 кПа до 77.9 кПа, тепловому потоці 3800 Вт/м^2 і масовій швидкості рабочого тіла 14.75 – 18.34 кг/(м^{2}с). Проведена кількісна оцінка зменшення коєфіцієнта тепловіддачі змоченої поверхні труби випарника при високих концентраціях мастила в розчині. В докладі виконаний аналіз впливу масової швидкості робочого тіла на значення локальних коєфіцієнтів тепловіддачі. На основі отриманих результатів було відмічено, що з ростом масової швидкості РРТ наблюдаеться незначне збільшення локального коефіцієнта тепловіддачі як зі сторони змоченої поверхні труби так і зі сторони парової фази. Запропоновано рівняння для моделювання локального коєфіцієнта тепловіддачі при кипінії розчину R600a/мастило в трубі, в якому використані експериментальні данні по теплофізичним властивостям розчинів холодоагент/мастило.
Ключові слова: *Коефіцієнт тепловіддачі – Реальне рабоче тіло – Кипіння – Ізобутан – Розчин холодоагент/мастило – Степінь сухості – Концентрація.*

I. INTRODUCTION

During operation of vapor compression refrigeration machine the certain amount of compressor oil always circulates with the refrigerant through the cycle of a compressor system. Lack of oil separators in a small refrigeration systems leads to an unavoidable circulation of small amount of oil through refrigeration system that together with refrigerant form refrigerant/oil solution (ROS). Mutual solubility of the compressor oil with the refrigerant has significant impact both on heat transfer in the apparatus as well as on the refrigeration machine operation in general. For this reason, it is necessary to have information about the influence of the compressor oil admixtures in the refrigerant on the boiling processes of the real working fluid (RWF) in an evaporator.

II. EXPERIMENTAL SETUP

In order to study the influence of oil impurities on the characteristics of isobutane boiling process in the evaporator complex experimental setup has been designed [1].

The test section of experimental setup was designed as smooth stainless steel U-tube with inner diameter d_{inner}=5.4 *mm* ± 0.05 *mm*, wall thickness δ=0.3 *mm* ± 0.05 *mm* and roughness 0.5-0.8 μm. The length of the test section was L=1691 *mm* ±2 *mm*. The evaporator was divided into nine sections. On the bounds of each section copper-constantan type-T thermocouples for measuring liquid RWF boiling temperatures and evaporator wall temperature are installed. The evaporator sections are linked by short rubber hoses (distance between sections of the evaporator is not greater than 5 mm), in which the thermocouples are inserted. The thermocouples for measuring the average over the section of pipe wall temperature are wound on the tube (a few turns to improve the thermal contact). On each evaporator section the differential thermocouples are located which are required to measure the temperature difference between the evaporator wall and the boiling working fluid in the upper and lower points of the evaporator.

Voltages of the nine thermocouples are measured simultaneously by a multimeter model TE 5065 with an error ± 0.0035 *mV*.

III. DATA PROCESSING

The specific heat flux is defined by equation

$$\dot{q} = \mathrm{W}_{\mathrm{test}} / (\pi \cdot d_{inner} \cdot L) \quad (1)$$

where, W_{test} – heat load on the evaporator, *W*; d_{inner} – inner diameter of test section tube (evaporator), *m*; L – length of the test section, *m*.

Local flow boiling heat transfer coefficient of RWF in the tube is calculated by the equation

$$\alpha = \dot{q} / \Delta T \quad (2)$$

where, ΔT – local temperature difference between the wall temperature (hot junction thermocouple mounted through electrically insulating material on the surface of the pipe) and the temperature of the RWF boiling at a certain point of tube, *K*.

The RWF flow is defined from approximation equation

$$\dot{M}_{RWF} = \dot{M}_W (1.6086 \cdot \Delta T_W / \Delta T_{RWF} - 0.1331) \quad (3)$$

where, $\dot{M}_W$ is a mass flow rate of water through the calorimetric flow meter, *kg/s*, ΔT_W is a differential temperature between inlet and outlet of water, *K*; ΔT_{RWF} is a differential temperature between inlet and outlet of working fluid, *K*.

The nominal oil concentration before throttling device is calculated from the approximated equation

$$w_{inlet} = -9.5 - 70393.2 \cdot \dot{M}_{RWF} - 11268.6 \times \\ \times \dot{M}_{RWF} \cdot \ln(\dot{M}_{RWF}) + 0.00106 / \dot{M}_{RWF} \quad (4)$$

The RWF mass velocity on the test section is calculated by the equation

$$\dot{m} = 4 \cdot \dot{M}_{RWF} / (\pi \cdot d_{inner}^2) \quad (5)$$

Local vapor qualities are defined by equation [2]

$$x_{local}^{ROS} = (\dot{M}_{ROS} \cdot h_{local}^{ROS} - \dot{M}_O \cdot h_{local}^{O} - \\ - \dot{M}_R \cdot h_{local}^{R,l}) / (\dot{M}_{ROS} \cdot (h_{local}^{R,g} - h_{local}^{R,l})) \quad (6)$$

where, $\dot{M}_R$ – mass flow rate of refrigerant, *kg/s*; $\dot{M}_O$ – mass flow rate of oil, *kg/s*.

Local oil concentrations in the refrigerant in different sections of the evaporator are defined by equation [2]

$$w_{local} = w_{inlet} / (1 - x_{local}^{ROS}) \quad (7)$$

The uncertainties of the measured parameters are listed in Table 1.

Table 1 – The uncertainties of the measured parameters

Parameters	Type of instruments	Range	Uncertainties
Local temperature difference, ΔT, *K*	Type-T thermocouple	–	0.09 – 0.13
Heat flux, $\dot{q}$, W/m^2	BVP 30V/50A	2400 – 4200	2.6 – 3.1 %
Mass flow rate of RWF, $\dot{M}_{RWF}$, *kg/s*	Calorimetric flow meter	$2.72 \cdot 10^{-4} - 4.68 \cdot 10^{-4}$	6.8 – 8.7 %
Nominal oil concentration w_{inlet} %	–	0.25 – 0.45	6.9 – 9 %
Mass flux, $\dot{m}$, $kg/(m^2 s)$	–	11.9 – 21	7.1 – 8.9
Local heat transfer coefficient, α, $W/(m^2 s)$	–	500 – 3700	4.5 – 17 %

IV. CORRELATION

As the basic model, Kattan et al [3] correlation has been taken during this experimental investigation. This model appears very promising to reveal the mechanism of flow boiling of the RWF.

At present study the two phase flow boiling heat transfer coefficient is defined from equation [3]

$$\alpha_{aver} = \frac{d_{in} \cdot \theta_{dry} \cdot \alpha_V + d_{inner} \cdot (2\pi - \theta_{dry}) \cdot \alpha_{wet}}{2\pi \cdot d_{inner}} \quad (8)$$

where, θ_{dry} - dry angle, *rad*; α_V is a heat transfer coefficient on the side of vapor phase, W/m^2K; α_{wet} is a heat transfer coefficient on the wetted perimeter, W/m^2K.

Heat transfer coefficient on the wetted perimeter α_{wet} is defined from equation

$$\alpha_{wet} = (\alpha_{nb}^3 + \alpha_{cb}^3)^{1/3} \quad (9)$$

In turn, α_{wet} combines the nucleate boiling α_{nb} and convective boiling α_{cb} contributions.

Calculation of nucleate boiling heat transfer coefficient is carried out by Cooper equation [4]. Using this model, it's necessary to have the data about molecular weight of oil, the information of which is missing because of commercial confidentiality. Since the compressor oils are multicomponent systems with different processing aids the experimental determination of this magnitude entails considerable difficulties [7]. For this reason, in this paper the Ivanov model [8] is proposed to use, which have a good prediction in the calculation of nucleate pool boiling of the R12/mineral oil solution at low concentrations of oil.

$$\alpha_{nb} = \left\{ 10^{6.3(1-w_R)} + \left[0.22 - 26(1-w_R)^2\right] \times \right. \\ \left. \times 1.0197 \cdot p_{suc} \cdot q^{0.75-2(1-w_R)} \right\} \quad (10)$$

where, w_R - mass fraction of refrigerant in the RWF, *kg/kg*; p_{suc} - suction pressure, *bar*.

The convective boiling heat transfer coefficient α_{cb} is predicted as follows [5]

$$\alpha_{cb} = A \cdot \left[\frac{4 \cdot \dot{m}(1-x)\delta}{(1-\varepsilon)\mu_{ROS}} \right]^n \times \\ \times \left[\frac{c_p^{ROS} \mu_{ROS}}{k_{ROS}} \right]^{0.4} \frac{k_{ROS}}{\delta} \quad (11)$$

where, A, n - empirical constants; μ_{ROS} is a dynamic viscosity of ROS, $N \cdot s/m^2$; c_p^{ROS} - specific heat, $J/(kg \cdot K)$; k_{ROS} - thermal conductivity of ROS, $W/(m\ K)$; δ - liquid film thickness, *m*.

The vapor void fraction is predicted from equation [5]

$$\varepsilon = \frac{x}{\rho_V} \left\{ \left[1 + 0.12 \cdot (1-x)\right] \left(\frac{x}{\rho_V} + \frac{1-x}{\rho_{ROS}} \right) + \right. \\ \left. + \frac{1.18}{\dot{m}} \left[\frac{g\sigma(\rho_{ROS} - \rho_V)}{\rho_{ROS}^2} \right]^{0.25} (1-x) \right\}^{-1} \quad (12)$$

where x - vapor quality, *kg/kg*; ρ_V - vapor density, kg/m^3; ρ_{ROS} - density of ROS, kg/m^3; σ - surface tension, *N/m*.

In the process of the executed researches it was revealed that coefficients A and n from equation (13) are depended from both heat flux and mass velocities.

The value of the heat transfer coefficient on the side of vapor phase – α_V, is obtained from Dittus and Boelter [9] turbulent flow heat transfer correlation

$$\alpha_V = B \cdot \left[\frac{\dot{m} \cdot x \cdot d_{inner}}{\varepsilon \cdot \mu_V} \right]^m \left[\frac{c_{pV} \mu_V}{k_V} \right]^{0.4} \frac{k_V}{d_{inner}} \quad (13)$$

where B and m – empirical constants.

IV. RESULTS AND DISCUSSION

In the studies the solution of isobutane (R600a) with compressor mineral naphthenic oil ISO VG 15 have been used as the working fluid.

At designed by the authors experimental setup were conducted studies of local heat transfer coefficients during working fluid boiling in the evaporator. Specific heat flux $\dot{q}$ was constant 3800 W/m^2. RWF mass velocity $\dot{m}_{RWF}$ varied between 14.75 to 18.34 $kg/(m^2 \cdot s)$. The measuring of the working fluid boiling process parameters was produced only after reaching equilibrium processes in the evaporator: the constant of RWF flow rate, constant in time values of pressure gauges and thermocouples.

Convective boiling heat transfer α_{cb} is a function of mass velocities and it almost does not depend from heat flux [10]. In order to study the influence of mass velocity on the intensity of boiling process of RWF in the evaporator, a series of experiments at constant heat flux were conducted. The dependence of local heat transfer coefficient of RWF on the side of wetted perimeter α_{wet} from oil concentration w_{local} on the various sections of the evaporator are presented in the Figure 1, where graphs of relative deviations of the experimental data are presented.

Figure 1 shows that increasing mass velocity leads to increase the local heat transfer coefficient of RWF on the side of wetted perimeter at $w_{local} \approx 0.4$ – 1.2%. This fact might be associated with increasing the convective boiling component in the pipe. Furthermore, increasing of mass velocity of RWF leads to shift the maximum of heat transfer coefficient towards lower values of the oil concentrations in time of RWF boiling.

The Figure 2 shows the dependence of local heat transfer coefficient of RWF on the side of vapor phase α_V from vapor qualities x on the various sections of the evaporator.

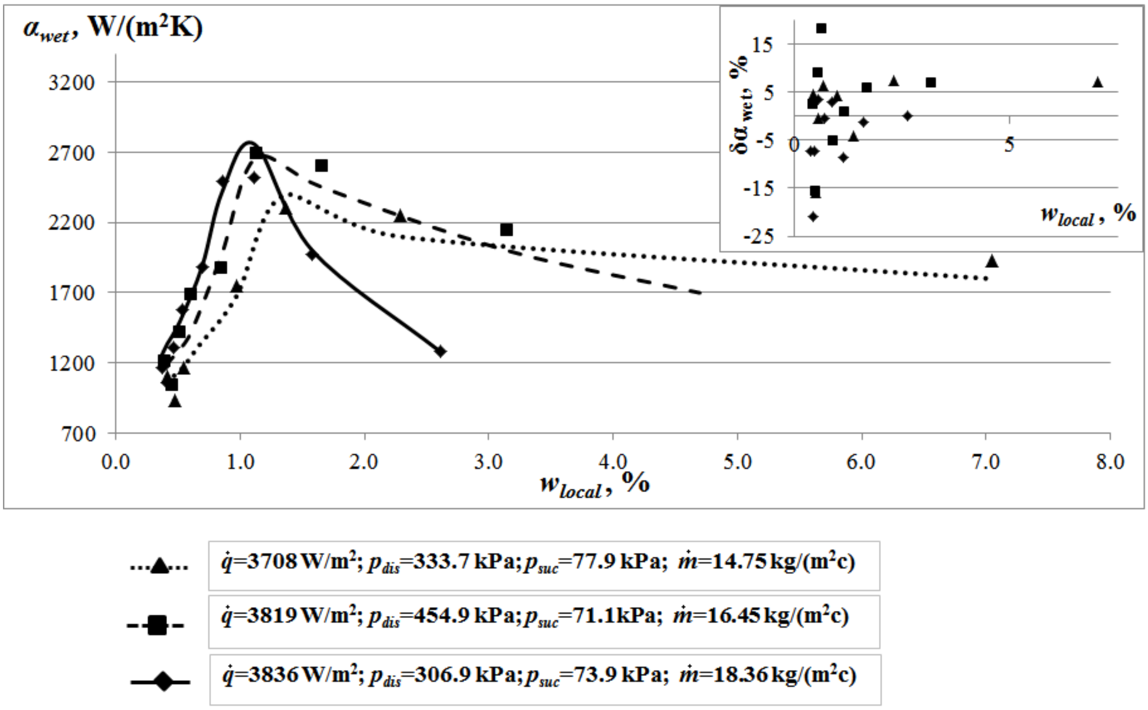

Figure 1 *– The dependence of the local heat transfer coefficient of RWF on the side of wetted perimeter α_{wet} from local oil concentration w_{local} on the various sections of the evaporator at constant heat flux $\dot{q}$ ≈3800 W/(m²K) and variable mass velocities $\dot{m}$ ≈14.75-18.35 kg/(m²s).*

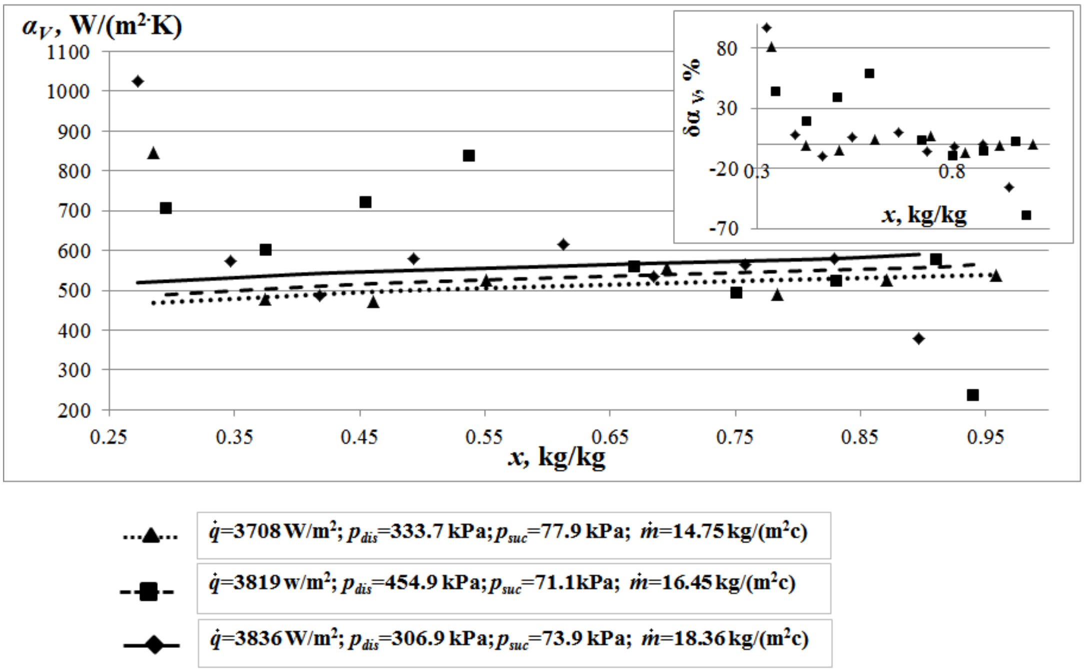

Figure 2 *– The dependence of the local heat transfer coefficient of RWF on the side of vapor phase α_V from vapor qualities x on the various sections of the evaporator at constant heat flux $\dot{q}$ ≈3800 W/(m²K) and variable mass velocities $\dot{m}$ ≈14.75-18.35 kg/(m²s).*

As shown in Figure 2, the increasing of mass velocity leads to insignificant increase of the local heat transfer coefficient of RWF on the side of vapor phase. It should be noted, that suggested by Dittus and Boelter [9] correlation has a poor prediction of the experimental data in the nonformated stratified flow area that, probably, occurs at the input of the throttling device.

In the process of the executed researches it was revealed that intensity of ROS boiling at the bottom of the tube was considerably higher than at the top. It shows that the RWF flow in the evaporator was stratified or wavy stratified.

The Figure 3 shows the dependence of average over the cross section heat transfer coefficient of RWF α_{aver} from oil concentration w_{local} at various sections of the evaporator.

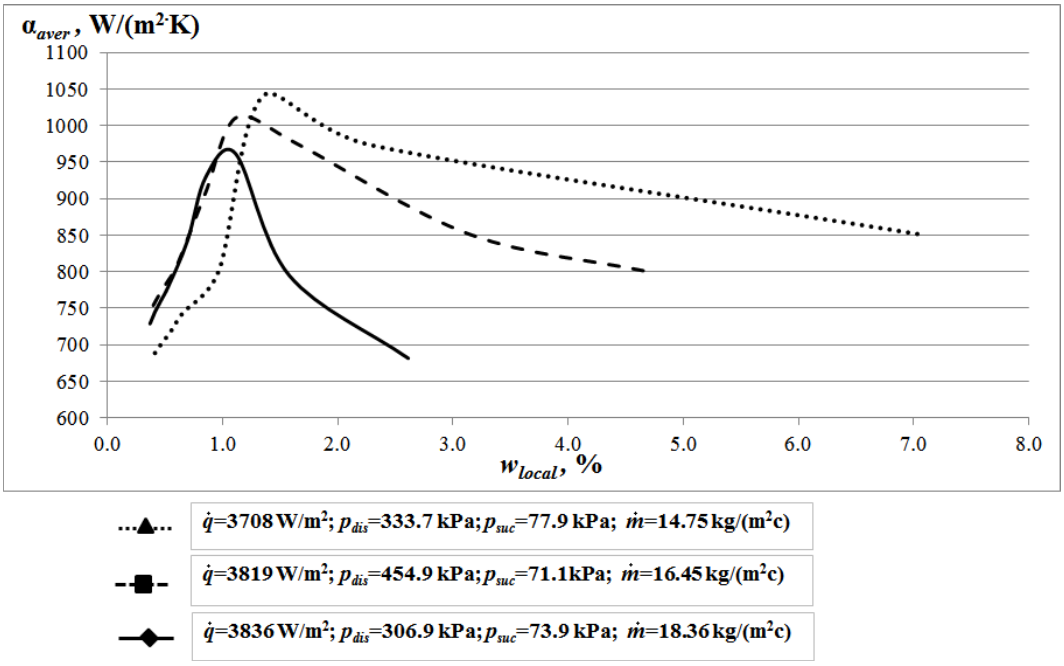

Figure 3 *– The dependence of average over the cross section heat transfer coefficient of RWF α_{aver} from oil concentration w_{local} on the various sections of the evaporator at constant heat flux $\dot{q}$ ≈3800 W/(m²K) and variable mass velocities $\dot{m}$ ≈14.75-18.35 kg/(m²s).*

It should be noted, that the heat transfer coefficient decreases rapidly at high concentrations of oil. Formation of oil enriched boundary layer ROS near the inner wall of the evaporator change the laminar flow to the turbulent and significantly decreases heat transfer by convection at high concentrations of oil.

V. CONCLUSIONS

The influence of the compressor oil ISO VG 15 admixtures in the refrigerant on the boiling processes of the real working fluid in the evaporator are experimentally investigated at present study. The technique of definition of the local heat transfer coefficient depending on the concentration of oil and vapor qualities are developed. The quantitative estimation in reduction of heat transfer coefficient of the wetted surface in evaporator at high oil concentration in the mixture is made. The influence of mass velocities of the working fluid on the values of the local heat transfer coefficients is analyzed. Based on the results obtained it was observed that increasing mass velocity leads to increase the local heat transfer coefficient of RWF both on side of wetted perimeter and vapor phase. The equation for the modeling of the local heat transfer coefficient for boiling of isobutane/compressor oil solution flow in the pipe is suggested. The experimental data of thermophysical properties R600a/ISO VG 15 are used.

The experimental results show that impact of the compressor oil admixtures on the processes of the ROS boiling in the evaporator has multiple-factor nature. Heat transfer coefficient at the RWF boiling depends on the concentration of oil in the refrigerant, heat flux, foaming process, flow rate and flow regime of the working fluid in the evaporator. Usage Kattan et al. model [3] does not consider the nonformation stratified flow at the input of the evaporator.

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ИЗМЕРЕНИЕ И МОДЕЛИРОВАНИЕ КОЭФФИЦИЕНТА ТЕПЛООТДАЧИ ПРИ КИПЕНИИ РАСТВОРОВ R600a/КОМПРЕССОРНОЕ МАСЛО ВНУТРИ ГОРИЗОНТАЛЬНОЙ ГЛАДКОЙ ТРУБЫ

В статье приведены результаты экспериментального исследования локальных коэффициентов теплоотдачи при кипении реальных рабочих тел (растворы хладагента R600a с минеральным нафтеновым маслом ISO VG 15) в гладкой трубе небольшого диаметра (5.4 мм). Исследования проводились при давлении в испарителе в диапазоне от 71.1 кПа до 77.9 кПа, тепловом потоке 3800 Вт/м^2 и массовой скорости рабочего тела 14.75 – 18.34 кг/(м^{2}с). Проведена количественная оценка уменьшения коэффициента теплоотдачи смоченной поверхности трубы испарителя при высоких концентрациях масла в растворе. В докладе выполнен анализ влияния массовой скорости рабочего тела на значение локальных коэффициентов теплоотдачи. На основе полученных результатов было отмечено, что с ростом массовой скорости РРТ наблюдается незначительное увеличение локального коэффициента теплоотдачи как со стороны смоченной поверхности трубы, так и со стороны паровой фазы. Предложено уравнение для моделирования локального коэффициента теплоотдачи при кипении раствора R600a/масло в трубе, в котором использованы экспериментальные данные по теплофизическим свойствам растворов хладагент/масло.
Ключевые слова: *Коэффициент теплоотдачи – Реальное рабочее тело – Кипение – Изобутан – Раствор хладагент/масла – Степень сухости – Концентрация.*

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Отримана в редакції 19.06.2014, прийнята до друку 23.06.2014