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/m², and mass velocity of working fluid was from 14.75 to 18.36 kg/(m²·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.

**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 refrigeration 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 $d_{inner}=5.4 \text{ mm} \pm 0.05 \text{ mm}$, wall thickness $\delta=0.3 \text{ mm} \pm 0.05 \text{ mm}$ and roughness $0.5-0.8 \mu \text{m}$. The length of the test section was $L=1691 \text{ mm} \pm 2 \text{ mm}$. 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 $\pm 0.0035 \text{ mV}$.

III. DATA PROCESSING

The specific heat flux is defined by equation

$$\dot{q} = W_\text{test} / (\pi \cdot d_{inner} \cdot L) \quad (1)$$

where, $W_\text{test}$ – 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, $\Delta T$ – 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

$$M_{RWF} = M_W (1.6086 \cdot \Delta T_W / \Delta T_{RWF} - 0.1331) \quad (3)$$

where, $M_w$ is a mass flow rate of water through the calorimetric flow meter, $\text{kg/s}$; $\Delta T_w$ is a differential temperature between inlet and outlet of water, $K$; $\Delta T_{RWF}$ 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

$$w_{\text{inlet}} = -9.5 - 703932 \cdot M_{RWF} - 11268 \times M_{RWF} \cdot \ln(M_{RWF}) + 0.00106 / M_{RWF} \quad (4)$$

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

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

Local vapor qualities are defined by equation [2]

$$\chi_{\text{local}} = (M_{ROS} \cdot h_{ROS} - M_O \cdot h_O - M_R \cdot h_{RJ, local}) / (M_{ROS} (h_{RJ, local} - h_{ROS})) \quad (6)$$

where, $M_R$ – mass flow rate of refrigerant, $\text{kg/s}$; $M_O$ – mass flow rate of oil, $\text{kg/s}$.

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

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

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

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

\[ \alpha_{avert} = \frac{d_{in} \cdot \theta_{dry} \cdot \alpha_v + d_{inner} \cdot (2\pi - \theta_{dry}) \cdot \alpha_{net}}{2\pi \cdot d_{inner}} \]  

where, \( \theta_{dry} \) - dry angle, \( rad; \alpha_v \) is a heat transfer coefficient on the side of vapor phase, \( W/m^2K; \alpha_{avert} \) is a heat transfer coefficient on the wetted perimeter, \( W/m^2K \).

Heat transfer coefficient on the wetted perimeter \( \alpha_{avert} \) is defined from equation

\[ \alpha_{net} = (\alpha_{nb} \times \alpha_{cb})^{1/3} \]  

In turn, \( \alpha_{avert} \) combines the nucleate boiling \( \alpha_{nb} \) and convective boiling \( \alpha_{cb} \) 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 nucleate boiling heat transfer coefficient of R12/mineral oil solution at low concentrations of oil.

\[ \alpha_{nb} = \left( \theta_{dry} \cdot \theta_{dry} \cdot \theta_{dry} \cdot \theta_{dry} \cdot \theta_{dry} \right) \times 1.0197 \times p_{wet} \times q_{wet}^{0.75} \times (1 - w_{oil}) \]  

where, \( w_{oil} \) - mass fraction of refrigerant in the RWF, kg/kg; \( p_{wet} \) - suction pressure, bar.

The convective boiling heat transfer coefficient \( \alpha_{cb} \) is predicted as follows [5]

\[ \alpha_{cb} = A \times \left( \frac{4 \cdot m \cdot (1 - \epsilon) \delta}{(1 - \epsilon) \delta} \right)^{0.4} \times \left[ \frac{c_{p} \cdot \mu_{ROS}}{k_{ROS}} \right]^{0.4} \times \frac{k_{ROS}}{\delta} \]  

where, \( A, n - \) empirical constants; \( \mu_{ROS} \) is a dynamic viscosity of ROS, \( Ns/m^2; c_{p} \) - specific heat, \( J/(kg \cdot K); k_{ROS} \) - thermal conductivity of ROS, \( W/(m \cdot K); \delta \) - liquid film thickness, \( m \).

The vapor void fraction is predicted from equation [5]

\[ \epsilon = \frac{x}{\rho_{v}} \left[ \frac{x}{\rho_{v} \cdot \rho_{ROS}} \right]^{1/2} \]  

\[ + \frac{1.18}{m} \left[ \frac{\sigma (\rho_{ROS} - \rho_{v})}{\rho_{ROS}} \right]^{0.25} \left( 1 - x \right) \]  

where \( x \) - vapor quality, kg/kg; \( \rho_{v} \) - vapor density, kg/m\(^3\); \( \rho_{ROS} \) - density of ROS, kg/m\(^3\); \( \sigma \) - 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 – \( \alpha_v \), is obtained from Dittus and Boelter [9] turbulent flow heat transfer correlation

\[ \alpha_v = B \times \left( \frac{m \cdot \rho_{suc} \cdot d_{inner}}{\epsilon \cdot \mu_{v}} \right)^{0.4} \times \frac{k_{v}}{d_{inner}} \]  

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 \( q \) was constant 3800 W/m\(^2\). RWF mass velocity \( u_{RWF} \) varied between 14.75 to 18.34 kg/(m\(^2\)·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 \( \alpha_{cb} \) 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 \( \alpha_{avert} \) 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 \( w_{local} = 0.4 \) - 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 \( \alpha_v \) 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 $\alpha_{wet}$ from local oil concentration $w_{local}$ on the various sections of the evaporator at constant heat flux $\dot{q} \approx 3800$ W/(m$^2$K) and variable mass velocities $\dot{m} \approx 14.75-18.35$ kg/(m$^2$s).

Figure 2 – The dependence of the local heat transfer coefficient of RWF on the side of vapor phase $\alpha_V$ from vapor qualities $x$ on the various sections of the evaporator at constant heat flux $\dot{q} \approx 3800$ W/(m$^2$K) and variable mass velocities $\dot{m} \approx 14.75-18.35$ kg/(m$^2$s).
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 nonformatted 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 $\alpha_{\text{aver}}$ from oil concentration $w_{\text{local}}$ at various sections of the evaporator.

![Figure 3](image)

**Figure 3** — The dependence of average over the cross section heat transfer coefficient of RWF $\alpha_{\text{aver}}$ from oil concentration $w_{\text{local}}$ on the various sections of the evaporator at constant heat flux $\dot{q} = 3800$ W/(m$^2$K) and variable mass velocities $\dot{m} = 14.75 \text{–} 18.35$ kg/(m$^2$s).

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 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.

**REFERENCES**

ИЗМЕРЕНИЕ И МОДЕЛИРОВАНИЕ КОЭФФИЦИЕНТА ТЕПЛООТДАЧИ ПРИ КИПЕНИИ РАСТВОРРОВ 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/м² и массовой скорости рабочего тела 14,75 – 18,34 кг/(м²·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 хладагента/масло.


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Отримана в редакції 19.06.2014, прийнята до друку 23.06.2014