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Heat transfer during condensation of moving steam in a narrow channel

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ABSTRACT

The paper presents the results of experimental investigation of heat transfer and hydrodynamics during condensation of moving steam in a narrow channel of square cross-section 2 mm× 2 mm. The channel had a serpentine shape, the channel length was 660 mm. An experimental cell simulated conditions of heat transfer in the condenser of loop heat pipes. The steam velocity at the channel inlet ranged from 13 to 52 m/s, the pressure was 1 atm. The temperature of the cooling water varied from 70 to 95 °C. The annular flow pattern was noted in the whole range of the regime parameters. There was a clear boundary between the condensation zone and the zone occupied by the condensed phase downstream. Temperature has measured along the channel, and the heat-transfer coefficients have been determined. The coefficient values varied from 10,000 to 55,000 W/K m^2 depending on the steam velocity at the channel inlet and the cooling temperature. The efficiency of the condenser – heat exchanger has been investigated.

1. Introduction

Lately much more interest has been shown in investigation of condensation processes inside pipe with a small cross-section [1]. This is primarily connected with the increased demand for compact two-phase equipments used for cooling electronic devices and their components. The present tendency of development of electronic devices is such that their dimensions decrease the power of the inner heat sources release increase. Accordingly, requirements for cooling systems have become more rigid. The use of loop heat pipes (LHP) is one of the most promising lines of attack on this problem [2].

At the same time, in a number of publications it has been repeatedly shown that LHP develops face the problem of the absence of reliable information on condensation processes in small-diameter pipes [3,4]. A fundamental knowledge and understanding of heat and mass exchange processes in condition are necessary for optimizing the design configuration of an LHP condenser.

In Ref. [1] it is mentioned that hydrodynamic and heat-exchange processes in micropipes (or microchannels) with a characteristic size up to 0.2 mm and minipipes (or minichannels) with characteristic sizes from 0.2 to 3 mm have a specific character. Therefore, although a sufficiently large number of papers describe heat exchange in condensation and regimes of a two-phase flow inside large-diameter pipes, these results cannot be used right in analyzing heat and mass exchange processes during condensation in channels with a small cross-section, where these processes are much influenced by the constrained condition of narrow channels. In this case the effect of capillary forces is also enhanced. All these effects finally determine not only the regime of a two-phase flow, but also the intensity of heat-exchange processes. The paper presents a review of two-phase flows in different pipes. The date available on the flow pattern in channels have been systematized. Maps of flow pattern are presented.

In Ref. [4] one can find a comparative analysis of today's most famous semiempirical relations for calculation of the main characteristics of a two-phase flow during condensation in a horizontal pipe, viz. relations for the pressure gradient and local heat-transfer coefficients. The author concludes that the results of calculations obtained with the use of relations of different investigations may vary widely. This is connected with the fact that most dependences were obtained in generalizing experimental data for a narrow range of parameters to be investigated and concrete liquids. Beyond this range and for other liquids such relations either prove to be invalid in general or may be used as valuation ones with a high degree of error.

Begg et al. [5] developing a loop thermosyphon investigated the problem of hydrodynamics and heat exchange in condensation of water in miniature straight tubes from 1.6 to 3.4 mm in diameter. They made experimental investigations and visual observations, which served as a basis for a modal of heat and mass transfer developed by them. In experimental investigations use was made of Pyrex tubes 302 mm long with a cooling-section length of less than 200 mm. The value of the axial heat flow due to phase changes varied from 1.71 to 12.06 W, accordingly, the mass flow rate ranged from $0.7{\cdot}10^{-6}$ to $5.0{\cdot}10^{-6}$ kg/s. The annular type of two-phase flow was registered in the experiments.

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Nomenclature

c_p	specific heat at constant pressure, J/kg K	Q	heat load, W
d	diameter, m		
h	latent heat of vaporization, J/kg	*Greek symbols*	
k	thermal conductivity, W/m K	α	heat transfer coefficient, W/m^2 K
T	temperature, °C	μ	viscosity, Pa s
m''	mass flux, kg/s m^2	ρ	density, kg/m^3
g	gravity acceleration, m/s^2		
l	length, m	*Subscripts and superscripts*	
$\dot{m}$	mass flow rate, kg/s	cs	cross-section
Nu	Nusselt number	$cool$	cooling
Pr	Prandtl number	l	liquid
S	square, m^2	v	vapor

Buz et al. [6] have developed models which allow carrying out calculations of local heat exchange at any flow regime combinations of vapor and a liquid film during full vapor condensation in straight pipes. Unfortunately, they have not made a comparative analysis of their calculated data and the results of some experimental investigations, which would allow validation of their analytical models.

Visual representations of processes in the condenser of an ammonia LHP were obtained with the use of neutron radiography [7]. The vapor–liquid boundary of ammonia in the condenser was observed near the condenser exit, regardless of the cooling water temperature. Most likely the LHP was operating in the regime of constant conductance, when one can observe the maximum possible vacation of the condenser, and the length of the condensation zone does not change.

The present paper is devoted to heat-transfer processes in a serpentine-shaped condenser – heat exchanger located on a lamellar radiator. A test bed with a working cell was made for this purpose. It allowed conducted both instrumental and visual observations of condensation in a long serpentine-shaped pipe with a small cross-section or, in other words, in a narrow long out-of-straight channel. Its length was 660 mm. The channel had a square cross-section with a side of 2 mm. Thus the channel length exceeded its hydraulic diameter 330 times. The channel had a complicated form and consisted of six straight sections 90 mm in length and five turns through 180°.

The setup made it possible to simulate the conditions which are realized during vapor condition in the condenser of a loop heat pipe operating in the regime of variable conductance. The length of the condition section was regulated by the internal process of redistribution of the liquid between the channel and the reservoir (or hydraulic accumulator) depending on the conditions of cooling of the condenser – heat exchanger and the thermal power supplied to the steam generator.

2. Experimental setup and measurement procedure

2.1. Description of the experimental setup and the experimental cell

The experimental setup (Fig. 1) was a closed circulation loop which included a working cell, two pipelines for vapor and condensate flow, and also a steam generator-pump similar in design to the evaporator of a loop heat pipe. The circulation of the working fluid through the experimental loop was ensured at the expense of capillary forces. Heat to the steam generator was supplied from an electric heater. The outer surfaces of the pipelines and steam generator had thermal insulation.

An experimental view of the working cell is shown in Fig. 2. This configuration of the cell made it possible to reproduce all the main peculiarities of hydrodynamics and heat transfer typical of radiator-type condensers in loop heat pipes [8].

Vapor condensation took place in a narrow, long, out-of-straight channel. The channel had a serpentine shape and consisted of six straight sections (bends) 90 mm long and five turns through 180° with a bend radius of 7.5 mm. Its total length was 660 mm. The channel had a square cross-section with a side of 2 mm. Thus the channel length exceeded its hydraulic radius 330 times. All the coils were situated in the same horizontal plane.

The channel was formed in an aluminum plate which had the shape a flat disk of diameter 160 mm and thickness 10 mm. The lower side of the cell was cooled with running thermostatted water. Visual observations of the condensation process, photography and filming were realized through the upper glass cover. The

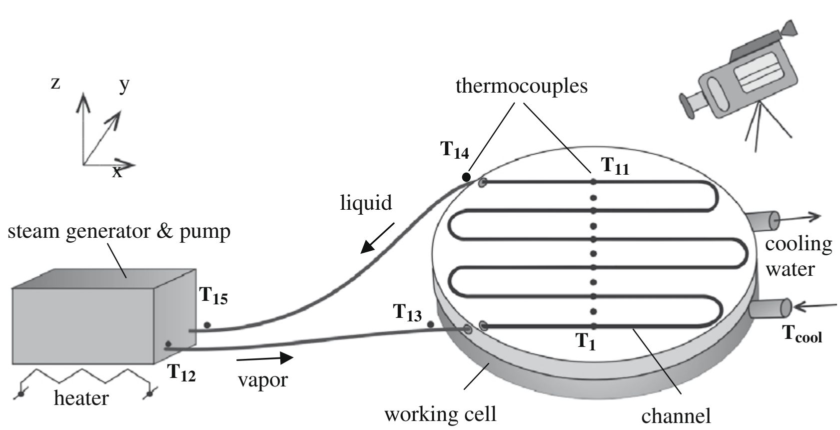

Fig. 1. Scheme of the experimental setup and the working cell.

Fig. 2. View of the working cell.

class cover was tightly pressed to the metallic base with the help of two screw joints, which can be seen in the photograph. Additional tightness of the working cell was ensured at the expense of applying silicone sealant paste along the perimeter of the butt-end surface of the cell and using a rubber packing ring.

The temperature was measured at eleven check points. Thermocouples were used as temperature-sensitive elements. Their location is indicated in Fig. 1. Thermocouples were positioned in "nests" made on the underside of the metallic base of the working cell (Fig. 3). Six thermocouples were located in the immediate vicinity to the channel. Distance between the channel bottom and thermocouple was at most 1 mm. The local heat-transfer coefficients were determined from their readings. Besides, additional thermocouples numbering five were placed between the coils of the channel. The readings of all the eleven thermocouples give sufficiently exhaustive information on the degree of homogeneity of the temperature field in the working cell, which finally makes it possible to evaluate the efficiency of such a condenser – heat exchanger as a whole.

Thermocouples in the scheme are numbered as follows. Thermocouple #1 is at the entrance into the channel, accordingly, thermocouple #11 is located near the exit from it. Odd-numbered thermocouples are situated on the channel wall, and "even" thermocouples are between the coils. During investigations records were also kept of the vapor temperature at the exit from the steam generator (T_{12}) and at the entrance into the working cell (T_{13}), and also the condensate temperature at the exit from the working cell (T_{14}) and at the entrance into the steam generator (T_{15}). The readings at the thermocouples were registered with the help of the measuring complex Agilent 34970a.

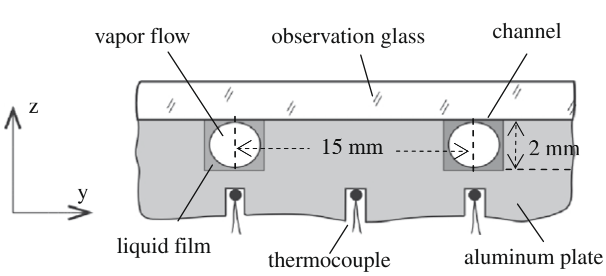

Fig. 3. Cross-section of working cell.

2.2. Experimental procedure

Experimental investigations were conducted by a procedure which made it possible to reproduce the main mechanism of formation of the condensation zone in the condenser of an LHP operating in the regime of variable conductance. This regime, according to theoretical notions of the device functioning, is characterized by the fact that under changes in the value of the heat load or cooling conditions the formation of a condensation surface is accounted for by the liquid redistribution between the condenser and the and the hydraulic accumulator (compensation chamber). For these purposes the steam generator of the experimental setup was equipped with a special reservoir, where the liquid displaced from the channel during the formation of the condensation zone in it could arrive.

At the beginning of an experiment the channel was filled with a liquid. When the steam generator was brought into operation, a vapor flow rushed along the vapor line to the entrance into the experimental cell, displaced part of the liquid from the channel and condensed on the "cold" channel walls. Eventually, such a press ended in formation in the channel of two zones which differed in structural and hydrodynamic parameters. In the upstream zone one could observe complete vapor condensation, and the downstream zone was occupied by the condensed phase. As the liquid moved to the exit from the channel, it cooled down. The formation of zones proceeded in the off-line mode. In this case the relation between the lengths of these zones was formed depending on the thermal capacity of the steam generator and the intensity of cooling of the working cell. The cooling temperature ranged from 70 to 95 °C. In all investigations the vapor pressure at the entrance into the channel was equal to 1 atm. The liquid under investigation was distilled water.

The temperature field in the cell and the length of the condensation zone were measured upon completion of all transient processes which arise after a change in the operating conditions of an experiment. Visual observations of the condensation process and photography were realized through the glass cover of the working cell, the condensation regime and the character of the motion of flow were determined.

Investigations were conducted at different flow rates of the working fluid. The flow rate was controlled by the thermal capacity Q supplied to the steam generator. The vapor-flow velocity u_v at the entrance into the channel was determined as follows:

$$u_v = \frac{Q - Q_{cp}}{h(T_{12}) \cdot \rho_v(T_{13}) \cdot S_{cs}} \tag{1}$$

where Q_{cp} is the heat expended for heating the liquid arriving at the steam generator with temperature T_{15} to the temperature of vapor T_{12} at the exit from it:

$$Q_{cp} = C_p \cdot \dot{m} \cdot (T_{12} - T_{15}), \tag{2}$$

where $\dot{m}$ is the mass flow rate. In investigations the thermal capacity supplied to the steam generator ranged from 100 to 400 W. Accordingly, the vapor flow velocity at the entrance into the channel varied between 13 and 52 m/s.

3. Results of investigation and their analysis

3.1. Flow visualization experiment

The main aim of visual investigations consisted in obtaining information on the type of condensation, the character of the flow of the working fluid in the zone of condensation and cooling. Besides, these observations are a sufficiently simple and at the same time reliable way of confirming the hypothesis on the mechanism of self-regulation of an LHP operating in the regime of variable conductance, when the length of the condensation zone changes depending on the value of the vapor flow rate at the entrance into the condenser and the conditions of the external heat exchange. In experiments the intensity of the external heat exchange between the condenser and the heat sink was controlled by the water temperature in the loop of cooling of the working cell.

Visual investigation showed that at all operating conditions the film type of condensation and annular pattern of a two-phase flow were observed. Along the whole length of the condensation zone, exclusive of a short section near the interface, the film was smooth and thin. According to evaluations, the film thickness was no more than 10–20% of hydraulic radius of the channel. Thus, most of the cross-section area of the channel was free for the vapor flow.

The film closure was observed only at the vary end of the condensation zone. In this case between the condensation and cooling zones there was a well-defined boundary in the form of a meniscus, as shown in Fig. 4. Under constant operating conditions, i.e., with established heat and mass exchange processes in the working cell, the position of the boundary between the zones was fixed. At the same time one could observe slight oscillatory displacements of the boundary of the order of 0.5–1 mm from its equilibrium position. Video filming has shown that the oscillation frequency is about one displacement a second.

The effects connected with the irregularity in the film structure were noticed only in proximity to the boundary of the zones, i.e., where the vapor velocity is minimum. Irregularities looked like local inconsiderable bulges in the condensate film. Two types of bulges were noted – asymmetrical and symmetrical. The asymmetrical type was observed where the boundary of the zones was situated right after the channel turn, as shown in Fig. 4. It can be seen that a bulge is present only on the outside of the turn rounding. In this case there was no film closure. After the turn the structure of the film flow leveled off, the bulge dispersed. The appearance of such a local asymmetrical bulge in the condenser film is most likely caused by the whirl of a two-phase flow in the curved section of the channel, which leads to an increase in the one-side dynamic action of the vapor flow on the film. In another photograph (Fig. 5) one can see a local symmetrical bulge in the condensate film. In this case the boundary of the zones was situated in one of the straight sections of the channel, far from the turn. In all probability, the reason for the appearance of a symmetrical formation in the film is the decrease of the vapor velocity in this section. In this case the effect of the capillary forces becomes predominant.

It should also be mentioned that neither in the curved nor in the straight sections of the channel did we discover any effects connected with the frustration of the condensate film or the carry-over of liquid drops by the vapor flow.

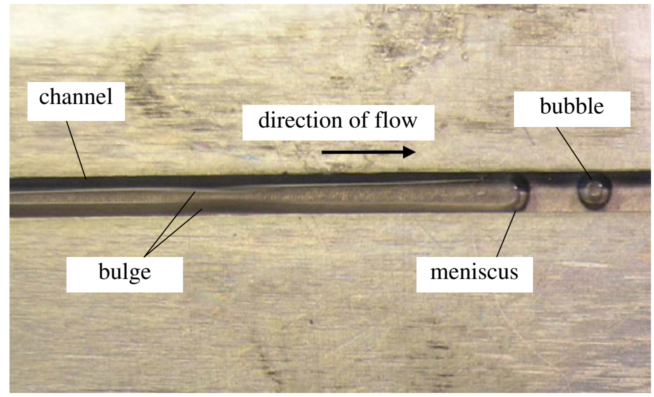

Fig. 5. Structure of a two-phase flow in a straight section of the channel.

Visual observations also showed the presence of the isolated bubbles in the condensate flow (see Figs. 4 and 5). They formed at the zone boundary. It happened irregularly and rather seldom. On forming such bubbles did not collapse. They moved together with the condensate flow to the exit from the working cell. Although, as the liquid moved to the exit from the channel, its temperature decreased; there was no appreciable decrease in the size of the bubbles. The most probable reason for the formation of such bubbles is the presence of permanent gases in the system.

Usually, for a qualitative analysis of a flow pattern during condensation in channels use is made of either a maps of flow patterns or specially developed semiempirical methods, which make it possible to determine what model should be used in investigation of heat-exchange during condensation. For instance, in Ref. [9] the criterion is suggested for determining the structure of a two-phase flow arising during condensation in a horizontal pipe. It is based on the dimensionless reduced velocity u_v^+ of the gas phase determined by the expression:

$$u_v^+ = u_v \cdot \sqrt{\frac{\rho_v}{g \cdot D \cdot (\rho_l - \rho_v)}}, \quad (3)$$

where D is hydraulic diameter of a pipe characterizing its cross-section. To formulate the conditions of this criterion, the authors used their own data and data of same other investigators [10–12]. They have established that an annular flow with predominance of tangential stress is realized when the values of the reduced velocity prove to be higher than 1.5. A flow with predominance of gravity forces is realized at $u_v^+ < 1.5$. In this case we are dealing with a stratifying pattern of a two-phase flow. The value u_v^+ of calculated at the smallest flow velocities u_v realized in our experiments was

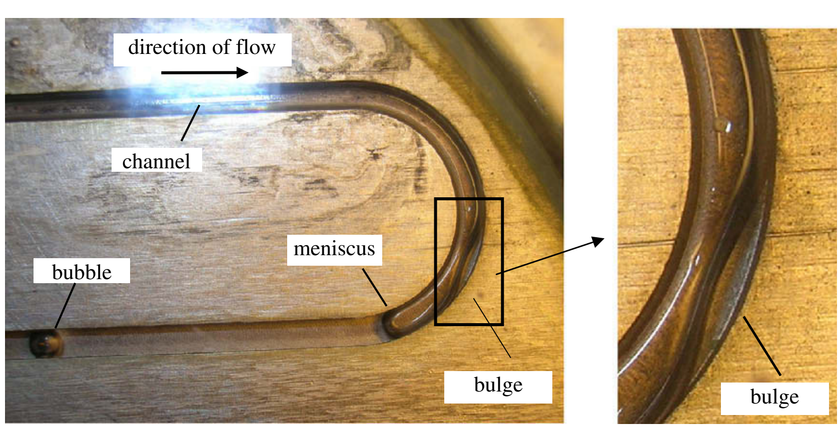

Fig. 4. Structure of a two-phase flow in a curved section of the channel.

2.6. The smallest velocity u_v corresponded to a mass flow rate $\dot{m}$ equal to 3.5×10^{-5} kg/s obtained at a thermal capacity supplied to the steam generator equal to 100 W. The character of the flow observed by us in the both the condensation and the liquid zone was fully reproduced at all velocity regimes of the vapor flow at the entrance into the channel. Thus it has been confirmed by experiment that the relation (3) used for determining the current regime of a two-phase flow may be used for a two-phase water flow moving in a narrow out-of-straight complex-shaped channels, which are used fairly often in LHP condensers.

3.2. Investigation of heat transfer during condensation

In the experiments data on the temperature distribution along the channel have been obtained, the temperature field of the working cell has been measured, the position of the boundary between the condensation and cooling zones has been established, and the length of these zones has been determined. Fig. 6 presents the dependence of the condensation zone length on the thermal capacity of the steam generator. Tests were conducted at different cooling temperatures. It has been found that for all temperature conditions of cooling the dependence $L_{cond} = f(Q, T_{cool})$ has a linear form. It is seen that at the same thermal capacity Q the length of the condensation zone L_{cond} increase with an increase in the cooling temperature. The regularity noted can be quite easily explained. The smaller the temperature difference between the vapor and the condensation surface, the larger the heat-transfer surface required for transmission of the same heat flow.

The local heat-transfer coefficient during condensation was calculated by the formula

$$\alpha_z(z) = \frac{q_{cond}}{T_v - T_w(z)}, \tag{4}$$

where T_w is the temperature of the channel wall at the point z, T_v is the vapor temperature at the entrance into the channel, q_{cond} is the heat flux removed through lateral surface of the channel in the condensation zone. The heat flux was calculated with the assumption of the uniformity of heat removal through the lateral surface of the channel

$$q_{cond} = \frac{Q_{cond}}{S_{cond}}, \tag{5}$$

where S_{cond} is the area of the condition surface. The upper glass side of the channel had a lower heat-transfer activity as compared with the other three. This circumstance was taken into account in calculating the actual condensation surface as follows

$$S_{cond} = 3 \cdot b \cdot L_{cond}. \tag{6}$$

Since the heat flow expended for the generation of a quantity of vapor is equal to the heat flow "liberated" during its complete condensation ($Q_{ev} = Q_{cond}$), we can write:

$$Q_{cond} = Q - Q_{cp}, \tag{7}$$

where Q is the heat flow supplied to the steam generator, Q_{cp} is the heat flow expended for heating the liquid arriving at the steam generator. When the condensation zone was sufficiently long, it was possible to evaluate the mean heat transfer over the length:

$$\overline{\alpha} = \frac{Q_{cond}}{S_{cond} \cdot (T_v - \overline{T}_w)} \tag{8}$$

and calculate mean value of Nusselt number

$$\overline{Nu} = \frac{\overline{\alpha} \cdot b}{k_l}. \tag{9}$$

An equivalent diameter of channel b was used as a characteristic size in the formula 9. Fig. 7 presents experimental data on the heat transfer rate depending on the vapor flow velocity at the channel entrance under different cooling conditions. An accuracy of the data obtained according to estimation was no more than 15%. The graph also presents values of the Nusselt number calculated by dimensionless correlation for predicting intensity of heat transfer during film condensation inside pipes suggested by Cavallini and Zecchin [13], Boiko [14] and Shah [15]. The mean values of the Nusselt numbers were carried out by the value of a vapor quality which was in the range from 0 to 1 in the correlation equations. In the graph one can see that at the low vapor velocities at the channel entrance the experimental data for all cooling temperatures T_{cool} are above all the values obtained by the calculated dependences of Cavallini, Boiko and Shah. Besides the higher T_{cool}, the more visible these distinctions are. With increase of the vapor velocity the situation changes. The experimental points at high values of Re are close enough to the correlation curves of Cavallini and Boiko. If to compare and analyze data as a whole, i.e., in the whole range of the investigated velocity regimes of the vapor flow, the following tendency can be observed: with the decreasing temperature of cooling the coordination becomes better of experimental data with data obtained by Cavallini and Boiko correlations.

The graph 8 presents data on the variation of the local heat-transfer coefficient along the length of the condensation zone. For the length of the condensation zone use was made of its reduced value: $Z^* = z/L_{cond}$. In the graph one can see dependences $\alpha = f(Z^*)$ for operating conditions at which the condensation zone was sufficiently long, and at the least four thermocouples registered the temperature conditions of vapor condensation. According to the data obtained the following tendency of the behavior of heat-transfer coefficients is observed. With increasing thermal

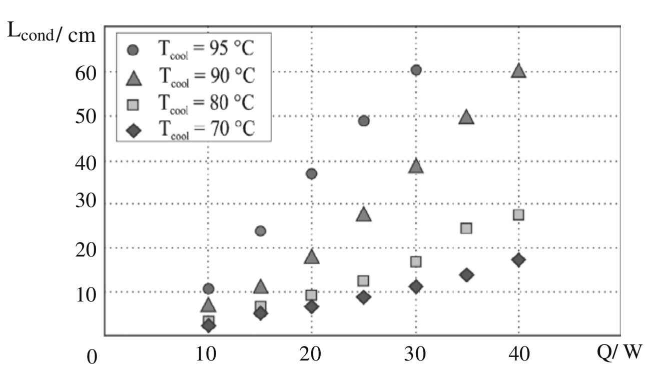

Fig. 6. Condensation zone length as a function of thermal capacity at different cooling temperature.

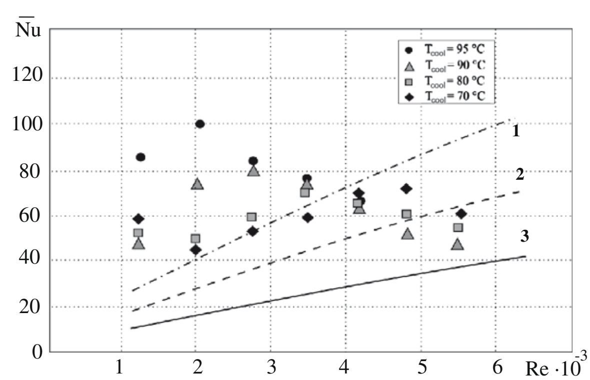

Fig. 7. Comparison of Nusselt number reported by Cavallini (line 1), Boiko (line 2) and Sash (line 3) with present experimental date.

capacity Q heat-transfer coefficients at the same cooling temperature decrease. Such a decrease in the intensity of heat-transfer processes during condensation is evidently caused by the increasing thickness of the condensate film on the wall of the channel. Besides, it can be seen that the higher the cooling temperature T_{cool}, the higher the values of heat-transfer coefficients at the same thermal capacity (Fig. 8).

3.3. Temperature distribution in serpentine-shaped condensers – heat exchangers located on a lamellar radiator

One of the criteria making it possible to evaluate the efficiency of the design of any condenser – heat exchanger is the degree of homogeneity of its temperature field. An example is the temperature distribution in the working cell shown in Fig. 9. The temperature was registered at eleven check points (see thermocouples position in the Fig. 1). This graph makes it possible to evaluate the dynamics of temperature variations in the cell under changes of the thermal capacity supplied to the steam generator. The results presented have been obtained for a cooling temperature of 90 °C. For other temperature T_{cool} (70, 80 and 95 °C) local temperature values at check points differ from those presented in Fig. 9, but on the whole the temperature distribution has a similar form. From the graph it is seen that the type of temperature dependences varies at a varying thermal capacity. At high values of Q the dependence initially has a gently inclined, almost horizontal section, where the temperature varies only slightly. It is followed by a section where the temperature decreases. At low thermal capacities, on the contrary, first one can observe an intense increase in the temperature, and then it becomes stable.

Once the data presented in Figs. 6 and 9 have been analyzed, it becomes clear that the initial thermostabilized section at high Q corresponds to the zone where vapor condensation is observed, whereas in the supercooling zone temperature variations prove to be more considerable. Here the temperature decreases rapidly tending to the value of T_{cool}. The observed effect of temperature stabilization of the initial section is evidently caused by the high velocity of the vapor flow in the channel at high Q. This ensures good convective heat transfer both inside the vapor phase and between the vapor and the channel walls. A result of such processes is the sufficiently homogeneous temperature field of the initial section.

At low Q the length of the condensation zone is inconsiderable, therefore the initial thermostabilized section either is poorly noticeable in the graph or is entirely absent. Instead, one can observe a decrease in the temperature. The condensate temperature decreases to the value of T_{cool}. After the attainment of this minimum value there are no more changes in the temperature. The data presented in the graph 9 testify that the efficiency of the condense – heat exchanger at small thermal capacity is low. For instance, at a capacity of 100 W almost 40% of the condenser – heat exchanger has a temperature close to the cooling one T_{cool}. Remaining cold, this part does not participate in the heat-transfer processes and is thermopassive.

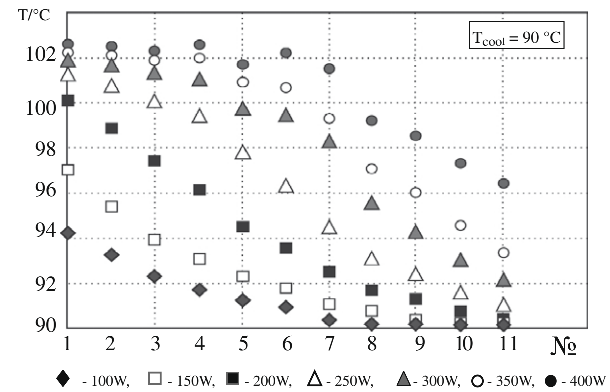

Fig. 9. Temperature distribution in the working cell at different heat flows.

The peculiarity revealed, which lies in the different character of behavior of the temperature dependence, may prove quite useful. With its help one can evaluate the length of the condensation zone and determine the approximate position of the boundary between the condensation and cooling zones. Although such an approach is indirect and rather rough, it may prove to be quite suitable, for instance, when it is impossible to organize visual observations of condensation processes in the condenser – heat exchanger.

To evaluate temperature inhomogeneities, one can also use the results presented in Fig. 10. Shown here is the dependence of the temperature difference ΔT between T_1 and T_{11} on the thermal capacity for different cooling conditions T_{cool}. It can be seen that at low Q the temperature difference ΔT is practically the same for all T_{cool}. Temperature drops increase with increasing thermal capacity of the steam generator. This is accounted for by the increase of the temperature in the upper part of the channel, whereas at the exit from the channel it remains invariant and close to the temperature in the cooling loop. For a temperature T_{cool} equal to 70, as well as for 80 °C the conditions linear increase of ΔT is retained in the whole range of Q, whereas for T_{cool} = 90 °C and T_{cool} = 95 °C dependences $\Delta T = f(Q)$ have a maximum on passing through which the value of ΔT begins to decrease. For T_{cool} = 90 °C a decrease in ΔT begins after the attainment of a steam generator thermal capacity of 250 W, for T_{cool} = 95 °C it happens earlier, at

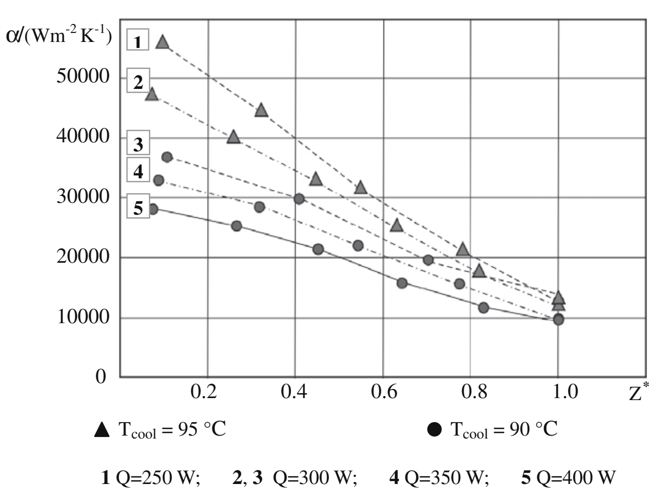

Fig. 8. Variation of a local heat-transfer coefficient along the condensation zone.

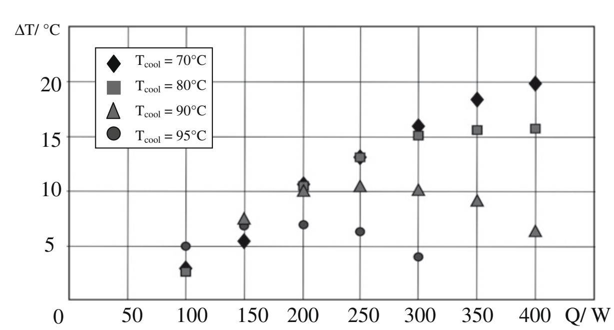

Fig. 10. Temperature drop ΔT as a function of thermal capacity at different cooling temperature T_{cool}.

200 W. Such a behavior means that up to the inflection point the inhomogeneity of the temperature field increases with increasing Q. On passing through this point the temperature field of the working cell begins to level off. The reason for such changes at high cooling temperatures is the sufficiently active liberation of the channel from the liquid with increasing Q. The length of the condensation zone increases to an extent of the condensation zone increases to an extent that "hot" vapor penetrates practically to the entire length of the channel, providing eventually a more uniform heating-up of the entire working cell. In such a state the whole heat exchanger works actively for external heat rejection. Evidently maximum point must be present in all graphs $\Delta T = f(Q)$. the lower the cooling temperature, the more to the right is the shift of the maximum point along the axis of abscissas. In our investigations at low cooling temperatures (for $T_{cool} = 70$ °C and $T_{cool} = 80$ °C) the corresponding thermal capacities have not been attained, therefore these dependences have no maximums in the graph.

Thus, from the above it follows that the efficiency of a condenser – heat exchanger depends on the degree of liberation of the channel from the liquid or, in other words, on how "open" the channel is for vapor condensation. Consequently, an important point for an LHP operating in the mode of variable conductance becomes the provision of condensations beneficial for liberating the condenser from the liquid. In this case its operation is most efficient, the LHP operating temperature decreases, accordingly, the heat transfer device has higher heat-transfer characteristics.

4. Conclusion

Visual and instrumental investigations of condensation of moving steam were conducted in a narrow serpentine-shaped channel at a constant pressure at the entrance into the channel equal to 1 atm. The channel length was 660 mm. The channel had a square cross-section 2 mm × 2 mm. investigations were conducted at different operating conditions. The vapor flow velocity at the entrance into the channel ranged from 13 to 52 m/s, which corresponds to variations of the Reynolds number between 1423 and 5528. the cooling temperature varied from 70 to 95 °C. The investigations demonstrated the following:

- The annular condensation flow in the condensation zone was noted in the whole range of operating conditions. There was a clear, well-defined boundary between the condensation and cooling zones, which was a meniscus. The liquid film in the condensation zone was thin. According to evaluations, its thickness was no more than 10–20% of the cross-section linear dimension of the channel. The condensate film was even end smooth everywhere, excluding a short section close to the end of the condensation zone. Here one could observe an inconsiderable bulge in the film. Such an irregularity in the film structure was caused by a decrease in the vapor velocity. For this reason the dynamic action of the vapor flow on the film diminished, and the effect of capillary forces increased. The character of the flow observed was totally reproduced at all velocity parameters of the vapor flow at the entrance into the channel.
- Sporadic bubbles were observed in the condensate flow. They formed at the zone boundary. It happened irregularity and seldom. If a bubble formed, it did not collapse. It moved with the liquid flow throughout the cooling zone to the exit from the channel, and its size did not practically change. The most probable reason for the appearance of bubbles is the presence of permanent gases.
- The cooling conditions and the value of the steam generator thermal capacity influence the length of the condensation zone. The dependence of the condensation zone length on the steam generator capacity has a linear form.
- Visual observations of the heat and mass exchange process in the working cell have shown that when an LHP operates in the regime of variable conductance, the condensation zone length varies at the expense of redistribution of the working fluid between the condenser and the hydraulic accumulator, under changes in the conditions of heat rejection or the value of the thermal capacity supplied to the steam generator.
- Local values of heat-transfer coefficients during condensation varied from 10,000 to 55,000 W/K m^2 depending on the vapor velocity at the entrance into the channel and the cooling temperature. The mean values of the coefficients ranged from 15,000 to 39,000 W/K m^2.
- The efficiency of a condenser – heat exchanger with a variable condensation zone length was conducted. It has been discovered that the homogeneity of the temperature field in such a heat exchanger depends on the length of the condensation zone. The highest efficiency of the heat exchanger is achieved at the maximum length of the condensation zone, i.e., in the case of the maximal liberation of the channel from a liquid.

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