THREE-DIMENSIONAL DEFORMATIONS IN NON-UNIFORMLY HEATED FALLING LIQUID FILM AT SMALL AND MODERATE REYNOLDS NUMBERS

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Abstract. Experimental investigations of hydrodynamics and heat transfer in a liquid film flowing along an vertical substrate with a local heater under the action of gravity are performed. The onset of rivulet-like structure in a wavy film motion (Re numbers from 1 to 66) at a condition on the wall $T_W = const$ has been studied. The dependence of the wavelength on heat flux and Re number has been measured. The average heat transfer coefficient in the rivulet flow has been measured. The effect of rivulet-like structure on heat transfer enhancement is verified.

Keywords. Falling liquid film; Local heater; Rivulet-like structure; Thermocapillarity.

INTRODUCTION

Thin liquid films are widely spread in different branches of industry. For instance, they are used in low-pressure evaporators for concentrating nutritional liquids and seawater desalinators. Such apparatuses include the systems of heated vertical or horizontal tubes. The plate type falling film evaporators are already common worldwide also. The treated liquid flows down in the form of evaporating falling film. Liquid evaporation occurs effective when heat flux at the surface of the tube is large and falling film is thin and wavy, remaining still continuous over the entire tube surface. Due to gravity, long-wave instabilities usually occur on the free film surface and finite-
amplitude waves propagate in the downstream direction. For a current review of the field see papers by Chang [1] and Oron et al. [2], as well as the book by Alekseenko et al. [3].

For a given flow rate there is a heat flux above which falling film cease covering the entire tube surface, but starts to rupture and to convert into rivulets, leaving dry patches on the surface. If the thin film is disrupted, the advantages of falling film evaporators, namely keeping low wall superheat and maintaining high evaporation rate, can not be reached because the mean heat transfer rate is much lower for a wall section with dry patches than that for a section covered by continuous film. The liquid films are usually provided by sprayers, being subcooled up to the saturation temperature and are susceptible to instabilities driven by shear stresses arising from the temperature dependence of surface tension (the so-called Marangoni effect). The action of Marangoni effect results in structural changes of the flow pattern that may eventually lead to increasing or decreasing of the heat-mass transfer coefficient and to the film rupture. To avoid the reduction of falling film evaporators performance by film breakdown it is of paramount importance to understand when and why the instabilities arise. The effect of thermocapillarity on gravitationally driven flow in thin liquid layers on a solid plate has been studied theoretically by Joo et al. [4] and by Miladinova et al. [5] for uniformly and non-uniformly heated plate respectively.

Recent experimental studies [6-13] have focused on thin films falling down inhomogeneously heated plates and have revealed the occurrence of novel effects: the phenomenon of "regular horseshoe-like structures" [6-10], the phenomenon of “horizontal bump” [7, 8, 13] and the phenomenon of “lateral waves” [12]. As the temperature of the fluid surface increases, the surface tension decreases. The concomitant surface tension gradient produces Marangoni flow opposed to the gravitationally driven flow. A liquid bump is formed in the region of the upper edge of the heating element. Beyond critical heat flux, rivulets aligned with the flow start from this bump and distribute spanwise with a fixed wavelength. Two lateral waves are also formed at the lateral edges of the heating element by the spanwise Marangoni flow. In the paper by Kabov et al. [6] by means of direct measurements of the temperature field on the liquid film surface using the infrared thermography it is established that the formation of the bump has the thermocapillary nature. A region with the maximum surface temperature gradient appears in the horizontal bump zone. The appearance of a large temperature drop in the streamwise and spanwise direction is found (gradients up to 10-15 K/mm).

In an attempt to explain the phenomenon of “horizontal bump” and other phenomena in the nonuniformly (locally) heated liquid film the two-dimensional and three-dimensional models taking into account variations of surface tension with temperature are proposed by Marchuk and Kabov [14], Kuznetsov [15], Sharypov and Medvedko [16], Scheid et al. [17], Kabova and Kuznetsov [18],
Skotheim et al. [19], Kaliadasis et al. [20]. In the papers [8, 12] two-dimensional and three-dimensional models taking into account variations of surface tension and viscosity with temperature have been studied.

In all existing experiments with local heating the researchers have strived to set the boundary condition on a heaters \( q=\text{const} \) (constant density of heat flux). In the contrary most of the recent theoretical models on the instability of falling films taking into account variations of the surface tension [17, 19] are developed at a condition on the wall \( T_w=\text{const} \).

The studies of fundamental regularities governing film flow regimes at condition on the heater \( T_w=\text{const} \) are of interest for agro-food industry also, where the falling film evaporators of fruit juice, milk and sugar are heated by water condensation.

Most of the existing experiments on rivulet structures are done at relatively small Reynolds numbers (0.06 - 9). The distances from the nozzle to the heat source (20-40 mm) are chosen so that the heating elements are located in the region of the smooth waveless part of the film. The non-uniformly heated wavy liquid films remain uninvestigated with the exception of recent papers [10, 21]. Experiments with the heating element of square section 150×150 mm\(^2\) show that the rivulet structures appear also for Re numbers up to 300 at the area of two-dimensional and three-dimensional hydrodynamics waves. The objective of the present work is to investigate experimentally the dynamics of rivulet structures and heat transfer for wavy film motion at a condition on the heater \( T_w=\text{const} \). Experiments have been conducted on a new test section developed in the Institute of Thermophysics, Russian Academy of Sciences. One of the novelty in comparison with the previous experiments is the measurement of the average heat transfer coefficient in the rivulet flow.

**EXPERIMENTAL SETUP**

The experimental setup presents a closed circular loop including a tank with electrical pump, a test section, filter, rotameters, and valves. The test section consists of main textolite plate
36×196×335 mm³ (a) with three heating elements (b), of temperature stabilizer (c) and of film sprinkler (d), Fig.1. A photo of the test section is presented in Fig.2. The pump supply the liquid to the film sprinkler built up from an accumulative chamber, a multi-hole distributor and a nozzle with a flat calibrated slot. The width and the height of the nozzle are equal to \( a_{\text{noz}} = 146.6 \) mm and \( h_{\text{noz}} = 0.1 \) mm, respectively. Liquid flowing down the plate is accumulated in the receiver and returned to the system under the effect of gravity.

The test section is made in two modifications (Fig. 3). In the modification “A” the distance between the nozzle of the film sprinkler and the upper edge of the heater is equal to 62.9 mm. In the modification “B” a textolite lengthening plate is incorporated to the main plate. The distance between the nozzle of the film sprinkler and the upper edge of the heater is equal to 396 mm. The opportunity to place the nozzle in two positions along with variation of Re number enable the experiments with wavy films to be conducted and in doing so an opportunity to change the film wave structure has been provided as well.

The heaters present flat copper heat exchangers with rectangular channels for the heating water pumping. In the lower part of Fig.1 the scheme of the liquid flows is shown that provide the sprinkler thermal stabilization and liquid film heating. In these experiments only the lowermost heating element is employed as a heater whereas both upper heating elements are employed as the film temperature stabilizers. It means that water of initial film temperature has been pumped through them. The dimensions \((L \times B)\) of the first, second and third heating elements downstream the film measure 25×120, 11×120 и 60×120 mm², respectively.

The experiments have been performed with water film draining. The initial temperature of liquid is equal to \( T_i = 20 \) °C. The test section is opened to the atmosphere. Reynolds number has been varied from 1 to 66. The heat flux density and the difference between the heaters wall and the initial film temperatures have been varied from 0 to 5 W/cm² and from 0 to 26 °C, respectively. The average heat flux has been defined on the basis of the
The amount of heat transferred from heating water as

\[ q = G_{hw}c_{hw}AT_{hw}/S_k \]  

(1)

The area of the heated surface is equal to 72 cm². The displacement of the thermocouples along the test section is shown in Fig.3. All the thermocouples are disposed at the test section axis of symmetry. The heater wall temperature is measured with three thermocouples. The distances between the thermocouples and the upper edge of the heater \((X_t)\) came to 15, 30, and 45 mm, respectively. Characteristic temperature distribution along the plate and the heater surface is presented in Fig.4. It is noticed that the design of the heating elements has provided satisfactorily the boundary condition \(T_W = \text{const.}\)

**THREE-DIMENSIONAL DEFORMATIONS**

Two types of rivulet-like structures

The previous works have shown that various pronounced rivulet flows can form in irregularly heated liquid films. Dynamic of regular structures forming is shown in Fig.5 for 10% alcohol-water solution flowing over a heater \(6.75 \times 10^{-9} \text{ mm}^2\) in size [22]. The heater is disposed in the field of smooth waveless film flow. The flow takes place in ambient stationary conditions. The liquid is appreciably subcooled. The temperature of the ambient air is sustained equal to the film initial temperature within the accuracy of \(\pm 1^\circ \text{C}\) \((T_0=8-20^\circ \text{C})\). The initial thickness of the draining liquid layer is equal to \(h_0=79 \, \mu\text{m}\). Local film heating results in stationary horizontal film thickening appearance – a bump in the field of front edge of the heating element, which size increases with the heat flux. A peculiarity of the flow is the propagation of the deformations spanwise the film flow, which amplitude is also rising with the heat flux density.

Primarily the spanwise deformations of the interface arise in the field of lateral edge of the heater as lateral stationary waves \((q=0.26 \, \text{W/cm}^2)\). At \(q=0.58 \, \text{W/cm}^2\) spanwise deformations arise in the film from the front edge of the heater (Fig.5). At \(q=0.78 \, \text{W/cm}^2\) lateral horseshoe-shaped structures arise at the edges of the heater. At the same time the horseshoe-shaped structures arise in
the middle part of the heater. There are spanwise deformations of various wavelength and amplitude within a narrow range of heat flux on the heater from $q=0.58$ to $q=1 \text{ W/cm}^2$. At $q=q_{rol}=1.01 \text{ W/cm}^2$ a line of structures of approximately identical wavelength $\Lambda$ that deform the surface to the utmost arise all over the heater. As the heat flux increases the number of the structures fitting in the heater length reduces, i.e. the wavelength increases.

For $q \geq q_{rol}$ the rivulet flow regimes occur quasi-stationary. As the heat flux varied the flow could become stationary, the rivulets could move across the heater or oscillate around some center of equilibrium.

For ease of handling such a type of rivulet flows has been denoted as “A”. In Figs. 6 and 7 there are photos of A-type rivulet flows under other conditions. The dependence of average wavelength on the heat flux has been studied in the work [23] for the flow of a dielectric liquid perfluorine-triethyl-amine over the heater of $6.5 \times 13 \text{ mm}^2$ size mounted on a vertical plate. For $Re=2$ the data fit the relation

$$\Lambda = 3.7 q^{0.39}, \quad (2)$$

where $\Lambda$ is measured in mm and $q$ is measured in W/cm$^2$. The wavelength increases from 3.4 to 4.8 mm with the heat flux increasing from 0.75 W/cm$^2$ to 1.79 W/cm$^2$. A feature of rivulet A-type flow is that the central stream of
essentially less size forming among the draining rivulets at \( q \geq q_{rol} \) is observed in the region of thin film. According to the research data available the average wavelength increases with \( Re \) [6, 22-24] and for the inclined plates as well [10, 24]. In the present work experimental results for vertical plate are only analyzed. In the work [24] the data on the average wavelength are generalized in the range of \( Re=0.42-24 \) and \( q= q_{rol} \) for various liquids and the heater sizes with the relation:

\[
\Lambda / \sigma = 3.26 \ Re^{1/6}
\]  

(3)

Another type of rivulet flows denoted as B-type flow has been defined for liquid film flow over a heater of intermediate size [21]. The experiments are taken with a heater of 150×150 mm\(^2\) size. The data have been obtained for water and for dielectric liquid FC-72. Reynolds number has been varied from 1 to 330. The initial film temperature ranged between 17 and 28 °C. The film sprinkler is mounted at the distances \( (X_n) \) of 41.5 mm and 120 mm from the upper edge of the heater. A photo of water film flow at \( Re=10.4 \) (\( X_n \) equals to 41.5 mm) is presented in Fig.8. Vertical heterogeneities on the film surface (spanwise deformations) become noticeable at relatively low density of heat flux (\( q \geq 0.3 \ W/cm^2 \)), distinguishing B-type of rivulet flows from the A-type one. The rivulets forming takes place at the end of the smooth region of the film at 70-80 mm from the nozzle where undulating movement on the film surface occurs. The average distance between the rivulets decreases from 20 to 15 mm with the heat flux growth, that essentially distinguishes B-type flow from the A-type as well.

A photo of the film flow for \( Re=22 \) (\( X_n \) equals to 120 mm) is shown in Fig.9. 2-D and 3-D waves come to the upper part of the heater. The artificial disturbances in the form of cylinders put in the film at different distances apart allow to change the instability wavelength only in a narrow range that corresponds to the boundaries of A and B flows. The distance between the rivulets has decreased by 30-40%. For non-isothermal water film the wavelength reduces with Reynolds number growth. The exponent \( n \) in the relation \( \Lambda \propto Re^n \) is ranged from \(-0.028\) to \(-0.31\). The parameter \( n \) increasing with heat flux is observed. The wavelength for the heater 150×150 mm\(^2\) is by a factor two or three larger than that for the heaters of small size.
The flows of types A and B with opposite dependence of the distance between the rivulets on the heat flux density were detected in the work [21] where the dielectric liquid FC-72 drained down the vertical heater of 150×150 mm² size. A photo of the film flow for Re=5 (X_n equals to 41.5 mm) is presented in Fig.10. The A-type flow has been observed in the upper part of the heater and the B-type flow take place in the middle and in the lowermost parts. The wavelength has varied over the heater more than twice. The distance between the rivulets alteration is apparently caused by irregularity of the heat flux distribution along the heater.

For low Reynolds numbers and for the liquid of low heat conductivity (FC-72) local density of the heat flux (q_l) differs from the average one (q) appreciably [25]. For the A-type flow (q≥0.25 W/cm²) Λ increases with Re that corresponds to the data obtained for the small size heaters. For the B-type flow (q≤0.35 W/cm²) Λ do not practically depend on Re that corresponds to the data obtained for water flow over the heater 150×150 mm². The concurrent occurrence of the A and B-flows on the heater shows that there is no pronounced boundary between these rivulet patterns. For the water film flow over the heater 150×150 mm² the A-type rivulet structures have not been observed, i.e. the rivulet flow of various types occurrence depends on the nature of the liquid and on the boundary conditions at the wall and at the film surface.

The onset of rivulet-like structures at a condition on the wall T=const

Map of water film flow patterns over the heater of 60×120 mm² size is presented in Fig.11. The regions where A and B-types of rivulet flows arise and the region where the film rupture takes

![Fig. 11. Map of water film flow patterns. a – breakdown, X_n=62.9 mm, 60×120 mm²; b – A -types, X_n=62.9 mm, 60×120 mm²; c – A -types, X_n=396 mm, 60×120 mm²; d – breakdown, X_n=41.5 mm, 150×150 mm²; e – breakdown, X_n=396 mm, 60×120 mm²; f – breakdown, X_n=120 mm, 150×150 mm², h – B-types, 60×120 mm² (Re>10, vertical line shows left boundary)](image)
place are shown. When \( X_n = 62.9 \) mm and Re\( \leq 10 \) virtually smooth film falls down the heater in the absence of heating. The structures of A-type arise there exclusively. When Reynolds numbers are low the structures are most pronounced and spread all over the heater (Fig.12). When Re\( > 10 \) the A-type structures arise only on a part of the heater.

For Re\( \leq 2 \) the film rupture accompanies rivulet generating on the upper part of the heater (Fig.13). After the film rupture an ordered rivulet flow takes place over the drained surface thus lowering the risk of dryout (Fig.12). In the field where film rupture arises randomly dry spaces of large area occur (Fig.14). It is seen from Fig.11 that the rupture on the heater of 150×150 mm\(^2\) occurs when average heat flux is substantially less than that necessary for the A-type structures generating. Probably it mainly explains why the structures of A-type escaped detection in the work [21] during the experiments with water films. The structures of B-type occur on the lower part of the heater.

\( X_n \) increasing up to 396 mm results in significant wave generating even for Re\( < 10 \). In the absence of heating 3-D solitone-shaped waves come to the heater. For Re=10.5 and for low heat flux the wave structure does not change virtually when flowing over the heater (Fig.15). The 3-D solitone-shaped waves become deformed with heat flux increasing and finally transform to vertical dynamic heterogeneities that generate rivulets. The A-structures occur just nearby the upper edge of the heater at \( q > 1 \) W/cm\(^2\). \( X_n \) alteration does not affect appreciably the film rupture. For higher \( X_n \) the A-structures generating occurs at just higher values of \( q \) (Fig.11).

In Fig.16 the dependence of dimensionless distance between the rivulets on the heat flux for various Re is presented. The new data are compared to the results of previous works. The intervals between rivulets are normalized to corresponding capillary constant for every liquid. For the A-structures observed in the experiment \( \Lambda \) increases with heat flux in agreement with the works [21-
Fig. 15. Heat flux effect on film flow patterns, water, Re=10.5, Xn=396 mm. It is seen that slope of the lines is a function of Re, of liquids type and of heaters size. The effect of capillary forces relaxes with Reynolds number increasing. For the B-structures observed in the experiment Λ reduces with the heat flux growth in according with the data from [21].

Analysis of Re influence on dimensionless distance between rivulets is shown in Fig.17. The new data are compared to those published in some previous works. Exfoliation of the data is

Fig.16. Dependence of dimensionless distance between the rivulets on the heat flux.

- water, 150×150 mm², Re=44.2, Xn=41.5 mm, B-type;
- water, 150×150 mm², Re=10.4, Xn=41 mm, B-type;
- water, 60×120 mm², Re=2.05, Xn=62.9 mm, A-type;
- water, 60×120 mm², Re=4, Xn=62.9 mm, A-type;
- water, 60×120 mm², Re=1.07, Xn=62.9 mm, A-type;
- water, 60×120 mm², Re=22, Xn=62.9 mm, B-type;
- water, 60×120 mm², Re=6.5, Xn=396 mm, B-type;
- water, 60×120 mm², Re=6.5, Xn=396 mm, A-type;
- water, 60×120 mm², Re=7.23, Xn=62.9 mm, A-type;
- water, 60×120 mm², Re=10.5, Xn=396 mm, B-type;
- FC-72, 150×150 mm², Re=5, Xn=41.5 mm, A-type;
- calculation on Eq. (2).
observed. For the A-structures observed in the experiment $\Lambda$ increases with Re according to the data obtained for local heaters in the works [6, 23, 24] and for the middle size heater [21]. It should be noticed that the new data on the A-structures for two different values of the parameter $X_n$ underlie the generalization for local heaters (3).

The results of two sets of experiments where the wavelength overlies the data obtained for the heaters of small sizes are also presented in Fig.17. The relation $\Lambda \times \text{Re}$ shows another tendency in these sets. When rivulet flow of B-type occurred $\Lambda$ practically do not depend on Re or reduces with Re growth according to the data obtained for water flow over the heater of $150 \times 150 \text{ mm}^2$ size [21]. The parameter $X_n$ in Figs. 16 and 17 varies from 41.5 to 396 mm. For Re=$\text{const}$ the value $\Lambda/l_\sigma$ reduces with $X_n$ growth when the structures of both types are generated. A similar effect has been observed in the work [21] for water flow over the heater of $150 \times 150 \text{ mm}^2$ size.

![Fig.17. Dependence of dimensionless distance between the rivulets on Re.](image)

There are two modes of rivulet forming in a non-isothermal falling liquid film. In the field of smooth film flow over small-size heaters the flow is free of initial disturbances of large amplitude. When heat flux ranges up to some threshold value A-type regular structures occur mainly due to the thermocapillary mechanism of rivulet forming. When liquid film flows along an extended heater the rivulets occur as a result of combined effect of thermocapillary forces and of waves formation on the interface (thermocapillary-wavy mechanism of rivulet formation). The development of rivulet flow is going gradually with heat flux growing and with growing of the distance from the upper edge of the heater. By this means the variety of wavelength and the character of its dependence on
heat flux and on Reynolds number are the criteria of the flows division into two types. Distance between the rivulets increases with heat flux for the A-type flows and reduces for the B-type flows.

**RIVULET-LIKE STRUCTURE EFFECT ON HEAT TRANSFER**

The heat flux dependence on the difference ($\Delta T$) between average temperature of the heater and the initial film temperature at $X_n=62.9$ mm (heater of $60 \times 120$ mm$^2$ size) is shown in Fig.18 for $Re=1$ and $Re=4$. The A-structures emergence leads to alteration of heat transfer. The temperature difference increases in this field while heat flux remains approximately constant. After the film rupture and transition to cooling of the heater surface by separated rivulets the temperature difference increasing brings to the heat flux growth again. Such independence of heat flux on temperature drop in the field of regular structures generating has been observed in the works [9, 26] where local heat transfer coefficient has been measured.

The heat flux dependence on temperature difference is shown in Figs.19 and 20 for $Re=10$ and $Re=22$. Heat flux independence on temperature difference is observed in the field of the film rupture as well. Rivulet flow of types A and B both generating does not affect practically on heat transfer (see Fig.11 also). The $X_n$ effect on heat transfer is shown also in Figs.19 and

![Fig. 18. Heat flux dependence on temperature difference, $X_n=62.9$ mm](image1)

![Fig. 19. The heat flux dependence on temperature difference, $Re=10$. a - $X_n=396$ mm, b - $X_n=62.9$ mm.](image2)
20. At $X_n=396$ mm as a result of three-dimensional wavy film flow the field of the heat flux independency on the temperature difference disappears. Heat flux increases monotonically with the temperature difference growth.

The dependence of heat transfer coefficient on heat flux at $X_n=62.9$ mm is shown in Fig. 21 for Reynolds numbers 4, 10 and 22. For Re=4 heat transfer coefficient does not depend practically on heat flux until the A-structures arise. After the rivulet regime forming heat transfer coefficient reduces by 10%. After the film rupture it increases again to the initial value. For Re=10 and Re=22 the structures of types A and B generating may has no impact on the heat transfer coefficient. Heat transfer varies considerably only in the field of the film rupture, where the flow of the separated rivulets on the heater take place. The heat transfer coefficient can reduce by 10-30%. The substantial scatter in the data is observed that is fraught with multiformality of rivulet motion.

Figures 22 and 23 show the comparison of experimentally obtained data to the theoretical relation by Nusselt [27]:

![Fig.20. The heat flux dependence on temperature difference, Re=22. a - $X_n=396$ mm, b - $X_n=62.9$ mm.](image)

![Fig. 21. The dependence of heat transfer coefficient on heat flux, $X_n=62.9$ mm.](image)
\[ \alpha_F = 0.0236 \frac{\mu c_p}{L} 4 \text{Re}_F + 2.074 \]

(Average heat transfer coefficient that is used in this work has been estimated using (4) by the relation:

\[ \alpha = \frac{\alpha_F}{1 + \frac{\alpha_F L}{\mu c_p \text{Re}_F}} \]

(5)

For \( X_n = 63 \text{ mm} \) (Fig.22) almost all of the experimentally obtained data overlie the line defined by Eqs. (4)-(5). The difference comes up to 50% for \( \text{Re}>7 \). The heat transfer enhancement is obviously due to significant wave generating in the film. This disagreement with the theory increases up to 300% for \( \text{Re}<7 \). In our opinion it corresponds to the growth of evaporation effect with reducing of the film thickness and the film velocity. Reynolds number has been varied from 6.5 to 44 in experiments with \( X_n=396 \text{ mm} \) (Fig.23). All of the data obtained experimentally overlie the line provided by Eqs. (4)-(5). The heat transfer enhancement increases up to 70%. The comparison between Fig.22 and Fig.23 confirms the hypothesis that the heat transfer enhancement is due to the effect of the flow structure waviness.

Taking into account the data obtained in the present work and the results of the previous works we can conclude that thermocapillary forces define the rivulet flows generating in essentially subcooled up to the saturation temperature liquid films. The appearance of rivulet flows of two types cause no significant alteration of heat transfer in subcooled falling films. However rivulet flows influence strongly to the film rupture and promote uniform wetting of the heated surface by the liquid, thus bringing to the critical heat flux growth. In the present work the amount of subcooling is equal to 80°C. Nevertheless it is expected that even for water films being essentially subcooled a significant effect of evaporation takes place for low Reynolds numbers (Re<7).
Considerable influence of the wave flow structure, i.e. of 3-D dynamical structures, on heat transfer takes place as a whole.

CONCLUSIONS

1. As a result of this and previous works a range of probable wavelengths has been detected for rivulet flow in the wide range of parameters significant for non-uniformly heated falling liquid film (Re, q, L, Xn, lσ...).

2. Dimensionless wave length is mainly defined by Reynolds number and heat flux \( \Lambda/l_\sigma = F(\text{Re}, q) \). Disturbing factors such as the distance between the nozzle and the upper edge of the heater and the inserts in the liquid reduce the range of instability (the range between the upper and the lower bound of the wave length).

3. The lower and upper boundaries of the probable wavelengths are defined by two regimes of instability: thermocapillary (A – type of rivulet flow) and wave -thermocapillary (B - type of rivulet flow).

4. The wave flow structure, i.e. 3D dynamic structures forming as a result of thermocapillary forces and wave film flow, affect on the heat transfer significantly. This influence may range up to 70%. For the small Reynolds numbers considerable effect of evaporation takes place additionally even for essentially subcooled up to the saturation temperature water films. Heat transfer enhancement may range up to 300% comparing with the classic Nusselts formula.
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NOMENCLATURE

- $B$ - heater width, mm
- $c_p$ - thermal capacity, J/(kg K)
- $c_{hw}$ – thermal capacity of heating water, J/(kg K)
- $g$ - gravitational acceleration, m/s$^2$
- $G$ – film flow rate per unit width, kg/(m s)
- $G_{hw}$ – flow rate of heating water, kg/s
- $h_0$ - initial film thickness, m
- $h_{noz}$ - height of the film sprinkler nozzle, mm
- $l$ - capillary length=$\left(\frac{\sigma}{\rho g}\right)^{1/2}$, m
- $Nu$ – Nusselt number=$\frac{h_0\alpha}{\lambda}$, dimensionless
- $L$ - streamwise heater length, m
- $q$ –average heat flux, W/cm$^2$
- $q_{rol}$ - heat flux corresponding to the A-type structures formation, W/cm$^2$
- $q_l$ –local heat flux, W/cm$^2$
- $Re$ - Reynolds number=$\frac{G}{\mu}$, dimensionless
- $Re_F$ - Reynolds number calculated using bulk temperature of the liquid, dimensionless
- $S_k=L\times B$ – area of the heated surface, m$^2$
- $T_0$ - initial temperature of liquid film, °C
- $\Delta T_{hw}$ – outlet-inlet temperature difference of heating water, K
- $T_W$ – average temperature at the heating element, K
- $\Delta T$- difference between average temperature of the heater and initial film temperature, K.
- $X_n$ – distance between the nozzle and the upper edge of the heater, m
- $X_l$ - distances between the thermocouples and the upper edge of the heater, m

Greek symbols

- $\alpha$ - averaged heat transfer coefficient=$\frac{q}{(T_W-T_0)}$, W/(K m$^2$)
- $\alpha_F$ - averaged heat transfer coefficient calculated using bulk temperature of the liquid, W/(K m$^2$)
- $\Lambda$ - wavelength of rivulet-like structure, m
- $\lambda$- liquid thermal conductivity, W/(m K)
- $\mu$ - liquid dynamic viscosity, kg/(m s)
- $\nu$ - liquid kinematics viscosity, m$^2$/s
- $\rho$ – liquid density, kg/m$^3$
- $\sigma$–surface tension, N/m
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