

DROPLET EVAPORATION ON A HEATED STRUCTURED WALL

by

Sergey Y. MISYURA* and Vladimir S. MOROZOV

Institute of Thermophysics Siberian Branch, Russian Academy of Sciences, Novosibirsk, Russia

Original scientific paper
<https://doi.org/10.2298/TSCI180502147M>

Evaporation of water droplets on a structured surface is studied experimentally. With an increase in the wall temperature T_w from 28-70 °C the exponent n increases from 1-1.37 in the evaporation law ($j = dm/dt \sim R_0^n$, where m is the droplet mass and R_0 is the droplet radius). Usually, researchers simulating droplet evaporation consider a linear relationship between the evaporation rate, j , and the droplet radius, R_0 ($n = 1$). This paper shows an increase of the exponent, n , with a growth of the wall temperature, T_w . The diffusion vapor layer on the droplet interface and the boundary air layer on the surface of the heated cylinder with a diameter exceeding the droplet's one are formed. A neglect of free convection more than five times underestimates calculation results compared with experimental data. At droplet evaporation it is necessary to take into account convection in a vapor-gas medium and wall roughness.

Key words: natural convection, structured surface, droplet evaporation

Introduction

Droplet evaporation is important in such fields as inkjet printing [1], microelectronics [2], DNA macromolecules [3], ice production [4], and microlense fabrication [5]. The studies of droplet evaporation on structured surfaces for different droplet diameters are interesting for a wide range of technical applications. An increase in the droplet diameter leads to a significant change in wall cooling, and this, in turn, results in a shift in boiling regime [6]. Low heat transfer between porous carbon nanotubes and water droplets has been observed in [7]. The case of large droplets differs from both liquid droplets and a thin liquid layer. For this case, the evaporation rate is proportionate to the square of droplet radius. It is assumed that a change in the evaporation law is caused by a change in natural convection, which is characterized by Grashof and Prandtl numbers [8]. A sessile droplet drying at free convection is accelerated. If the heater radius many times exceeds the droplet radius, the natural convection above the droplet increases the evaporation rate by more than 20-30%, even for moderate wall temperature of about 60-70 °C [9]. Evaporation of sessile droplets is considered in [10-21]. Small droplets falling on the heated wall interact and merge with each other, while large droplets interact and form a thin liquid film. Evaporation of droplets falling on the wall is considered in [22-27]. Nucleate boiling in droplet solutions differs from single-component liquids and depends on the concentration of components [28]. Currently, an increased attention is paid to the creation of structured surfaces with high hydrophobic and hydrophilic properties and to the studies of wettability [29]. The superhydrophobic surfaces may be effectively used for various purposes: self-cleaning, increasing condensation intensity and heat transfer, preventing icing, and reduc-

* Corresponding author, e-mail: misura@itp.nsc.ru

ing wall friction. It is shown that the onset of the pinning regime (the contact droplet line is attached, $R = \text{constant}$) is delayed due to changes in the wall surface topography. Usually, the simulation of droplet evaporation is focused on the linear relationship between the mass flux of vapor and the droplet radius. The evaporation of small water droplets (1 mm diameter) on a solid wall at wall and air temperature of 22 °C, when the Marangoni flow and the convection in liquid and gaseous phases may be neglected, has been studied in detail. The growth of the droplet diameter of easily volatile liquids leads to significant intensification of evaporation due to steam convection [30]. Gas-droplet flows are widely used for fire extinction and in solving environmental problems [31-33]. The measurement technique of the thermal imager is presented in [34]. The rate of evaporation is affected both by convection and by external turbulence [35, 36]. The flow of vapor and fine droplets is formed when burning CH_4 hydrate, which leads to a decrease in the combustion temperature of the fuel [37].

In this paper, we consider separate influence of diffusion, Stefan flow and free convection on evaporation. The evaporation of large and small droplets differs fundamentally by the effect of different physical mechanisms. To date, there is no rigorous theoretical model that simultaneously takes into account several key factors, affecting droplet evaporation. The qualitative analysis carried out in this work allows classifying these factors by their importance.

Experimental data and analysis

Experimental measurements were carried out on the horizontal structured wall at air temperature of 21 °C and air pressure of 1 bar. The initial temperature of water was 21 °C. Wall roughness was measured with the help of the optical profilometer *New View 8000 Series* (measurement uncertainty of ± 15 nm). The droplet was located on the wall of the titanium alloy cylinder with the diameter of 90 mm, figs. 1(a) and 1(d).

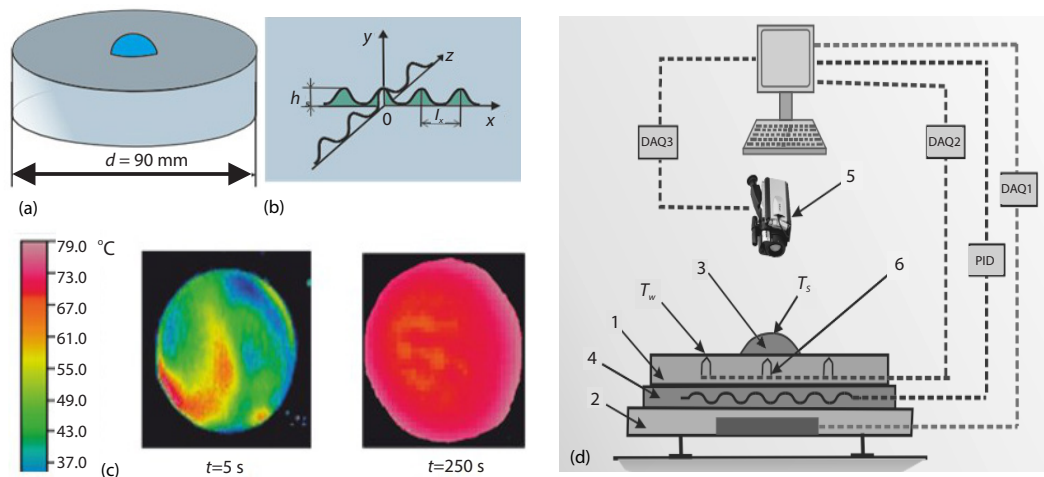


Figure 1. (a) Water droplet evaporation on the heated horizontal cylindrical wall, (b) structured surface ($l_s = 0.2$ mm, $h_s = 20$ μm), (c) thermal images of a water droplet, (d) the experimental set-up:

1 – titanium working section, 2 – scales, 3 – droplet, 4 – heater, 5 – thermal imager, 6 – thermocouples

The measuring thermocouples were located near the wall surface. The DAQ2, PID, heater, fig. 1(d), served to control the wall surface temperature T_w ($T_w \approx \text{constant}$). The droplet free surface temperature, T_s , fig. 1(d), was determined by thermal imager (NEC-San Instruments, 640×512 pixels, resolution of 10 μm). The measurement error of the used infrared camera was

within 1°C. The droplets were formed by a micro-dispenser with the maximal relative volume error below 1%. External air humidity was 30%. Degassed bidistillate was used in all experiments. The volume and mass of a droplet were determined by the weight methods. The heated metallic cylinder with a droplet were placed on the high-precision scales, fig. 1(d). The decrease in the mass of a droplet due to evaporation was registered by scales and processed by software. The value of a droplet evaporation rate j was determined from $j = \Delta m / \Delta t$, where m is a droplet mass, t is time. The droplet static contact angle on the structured wall was 68-72°. The surface profile of the structured heated wall is shown in fig. 1(b). The structured surface has the wave profile: height $h_s = 20 \mu\text{m}$ and profile period $l_s = 0.2 \text{ mm}$. The structured wall was made by mechanical treatment of its surface. Thermal images of a water droplet are shown in fig. 2(c). A noticeable convective water motion is observed at short times ($t = 5 \text{ s}$) when the liquid is heated.

Most of experimental works were carried out on a heater, whose length slightly exceeded the droplet diameter. In addition, the small droplets were investigated at a low heating level. These experimental conditions allowed neglecting the effect of free convection in the vapor-gas medium, and the diffusion transfer was the determining factor of evaporation. If the droplet size changes over a wide range, and the heater size is much larger than the droplet diameter, the droplet evaporation depends on several key factors.

There are free convective flows above the droplet surface figs. 2(a) and 2(b). The maximum value of the Rayleigh number for air (thermo-gravitational air-flow over the heater, fig. 2(c), with a characteristic velocity, V_a) is $Ra_a = 5 \cdot 10^5$, and the maximum value of the Rayleigh number (for free convective vapor flow over a droplet, fig. 3(c), with a characteristic velocity, V_v) is $Ra_v = 3 \cdot 10^5$. $Ra_a = g\beta\Delta T_a R_0^3 / (\nu\alpha)$, $Ra_v = g\beta\Delta T_v R_0^3 / (\nu\alpha)$, where g is the gravitational constant, β – the thermal expansion, ν – gas viscosity, α – the thermal diffusivity, $\Delta T_a = 7-49^\circ$ is the difference of temperature between the substrate (without a droplet) and the ambient air, the heater radius is 45 mm, $\Delta T_v = 6-39^\circ$ is the difference of temperature between a droplet free surface and the ambient air, the droplet radius is $R_0 = 0.2-40 \text{ mm}$. The critical Ra_{cr} , at which the gas loses its stability, and the motion starts, is about 1000-2000 [38, 39]. Thus, Rayleigh number in these experiments significantly exceed the critical values that proves the intensive free convection.

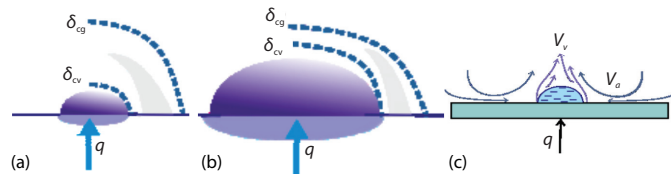


Figure 2. (a), (b) convective boundary-layers, (c) free gas convection

Two cases of evaporation are schematically shown in fig. 2: for small (a) and large (b) droplets. Here, δ_{cv} is the convective vapor layer above the droplet surface, and δ_{cg} – the convective layer of gas (air) above the heated wall surface. If the heater radius and the droplet diameter are equal, $\delta_{cg} = 0$ (there is no air convection above the heater). For a small droplet, $\delta_{cv} \approx 0$ (the vapor-gas convection above the droplet interface may be neglected). Moreover, dividing the free convection into two types is conditional and allows considering the extreme cases of evaporation for very small and large droplets. For fig. 2(a), $\delta_{cg}/\delta_{cv} \gg 1$, and for fig. 2(b), $\delta_{cg}/\delta_{cv} \approx 1$. Different ratios of thicknesses, δ , lead to different profiles of steam concentration, C , above the droplet surface. In the presence of free convection, the rate of V_0 at the border of the near-wall boundary-layer is unknown, dependent on the longitudinal coordinate x and ΔT ($\Delta T_w = T_w - T_0$, $\Delta T_s = T_s - T_0$, where T_w is the wall temperature, T_s – the interface temperature, T_0 – the temperature at the external border of the boundary-layer, δ), $V_0 \sim [gR_0(\Delta\rho/\rho)]^{1/2}$, $\Delta\rho/\rho = f(\Delta T)$, where R_0 is a characteristic size, g – the gravitational constant, and $\Delta\rho$ – a change in the gas density due to the temperature change. With an increase in temperature, T_w , the dif-

ference $\Delta T = T_w - T_0$ increases, leading to a growth in the flow velocity. As a result, the steeper velocity profile leads to an increase in convective vapor transfer. At that, the profile of vapor concentration on the droplet surface will have a higher gradient. For air, the numbers of Prandtl, Lewis, and Schmidt are close to 1, and the following similarity and equality should be satisfied $St = St_d = C_f/2$. Here, the Stanton number is $St = q_w/(\rho_0 V_0 C_p \Delta T)$, the diffusion number of Stanton is $St_d = j/(\rho_0 V_0 \Delta C)$, j – the evaporation rate, ρ_0 , V_0 – are density and velocity of gas at a boundary of the gas boundary-layer, respectively, C_f – the friction coefficient, and q_w – the heat flux on the wall, $q_{w1} = \alpha_1(T_{w1} - T_0)$ for gas and $q_{w2} = \alpha_2(T_{w2} - T_0)$ for vapor-air. Here α_1 is the heat transfer coefficient for gas over a wall without a droplet, α_2 – the heat transfer coefficient for vapor-gas, T_{w2} – the temperature under a droplet, $j = \beta(C_s - C_0)$, β – the mass transfer coefficient, C_s – the equilibrium relative steam concentration on the free surface of a droplet, and C_0 – the steam concentration at the border of the near-wall boundary-layer. For the sessile droplet there is relation between evaporation and droplet contact angle $j = dm/dt = -\pi R_0 D \Delta \rho f(\theta)$, where D is the diffusion coefficient for air-vapor and θ – the contact angle. To simplify the analysis, we can consider the case with high wettability and low contact angle, when angle function is constant $f(\theta) = 4/\pi$. At similarity, there is expression (1):

$$\beta \sim \alpha \sim a Ra_l^n + f(Ma) + 4rD(\rho_v + \rho_a) \ln(1 + B_M) + a_2 Ra_v^{n_2} + a_1 Ra_a^{n_1} \quad (1)$$

where B_M is the Spalding mass number, $B_M = (\rho_{vs} - \rho_{v0})/\rho_a$, ρ_a – the density of gas mixture, Ra_a – the Rayleigh number for free-convection motion of air on the heated wall with characteristic radius, Ra_v – the Rayleigh number with characteristic radius R_0 (droplet radius) for free air-vapor convection on the droplet interface, Ra_l – the Rayleigh number for liquid convection in a droplet, and Ma – the Marangoni number for a droplet. First four terms in expression are interdependent. Thus, a change in convection of liquid will affect vapor diffusion and *vice versa*, a change in convection of gas will affect convection in liquid. Most often, only the third term (vapor diffusion and Stefan flow) is considered in simulation. In recent years, researchers consider the combined effect of the third and fourth terms on the rate of droplet evaporation. However, disregard of other key factors in expression (1) is possible only in special cases, namely, if the following conditions are fulfilled: the evaporation rate is low, heater radius is close to droplet radius, heater temperature is close to ambient temperature, channel height is low, droplets are small, etc. Depending on these conditions, some factors may be neglected. At the same time, the choice of key factors is the subject of research for every particular problem. A purely diffusional model may be true only in a very limited number of cases. Let us try to estimate the predominant factors at evaporation. For droplet radius $R_0 = 0.5$ mm, the heat and mass transfer in the droplet is determined by conductive heat transfer and ratio of characteristic velocities $U_M/U_a \approx 0.15$ (for $\Delta T = T_w - T_s = 15$ °C), where U_M is the average circulation rate in the droplet due to Marangoni forces, and U_a is the characteristic rate of conductive heat transfer [35]. If ratio U_M/U_a is much less than unity, convection may be neglected. Moreover, under these conditions, we can neglect thermo-gravitational convection $Ma/Ra \ll 200$. For $R_0 \approx 1$ mm and $\Delta T = 30$ °C, thermal diffusivity and Marangoni number make an equivalent contribution to heat transfer in a droplet. With an increase in R_0 and ΔT , the convection will play the determining role for heat and mass transfer in a liquid [35].

The average rate of droplet evaporation in a wide range of initial radii of droplet base R_0 and at wall temperature $T_w = 28$ °C, 52 °C, and 70 °C is shown in fig. 3. The T_w corresponds to the wall temperature under the droplet. In all cases, most evaporation time corresponds to the pinning regime ($R_0 = \text{constant}$). The evaporation rate for a small droplet ($R_0 < 1$ mm) depends linearly on the initial droplet radius ($j = dm/dt \sim R_0^1$). For the droplet size R_0 of

0.7-2 mm, the experimental points start deviating from the linear relationship. With increasing wall temperature from 28-70 °C, n increases from 1.23-1.37. Experimental data for varying exponent n vs. wall temperature under the droplet are shown in fig. 4(a). Each point is obtained as an average value for three repeating experiments. The error interval is indicated in the diagram. According to the figure, the maximal derivative dn/dT_w corresponds to the wall temperature, when it is the closest to ambient air temperature $T_w = 23$ °C. When approaching $T_w = 70$ °C, the derivative reaches an extremum. This behavior of the exponent n is caused by the competition of two factors: vapor diffusion and convection in gas. Both factors are non-linearly related to an increase in temperature. Vapor diffusion is determined by parameters, D , (diffusion coefficient) and $\Delta\rho$, $D \sim T^{1/5}$, and ρ_s is connected with temperature T_s in the form of power dependence. Convective velocity, V_0 , is proportional to $(\Delta\rho_s)^{1/2}$. Obviously, with increasing temperature, the role of diffusion component (diffusion transfer of vapor) will increase. As a result, the exponent n with increasing T_w will increase, reach an extremum and then decrease. With a decrease in T_w , an error of n increases, so it is difficult to determine n experimentally, as we approach $T_w = 23$ °C. However, if we extend the experimental curve towards the smaller values of T_w , n will approach 1. It is found that when the droplet evaporates on an unheated wall, the free convection of vapor occurs at $R_0 > 20$ mm, and n increases discontinuously ($n = 2$) [8].

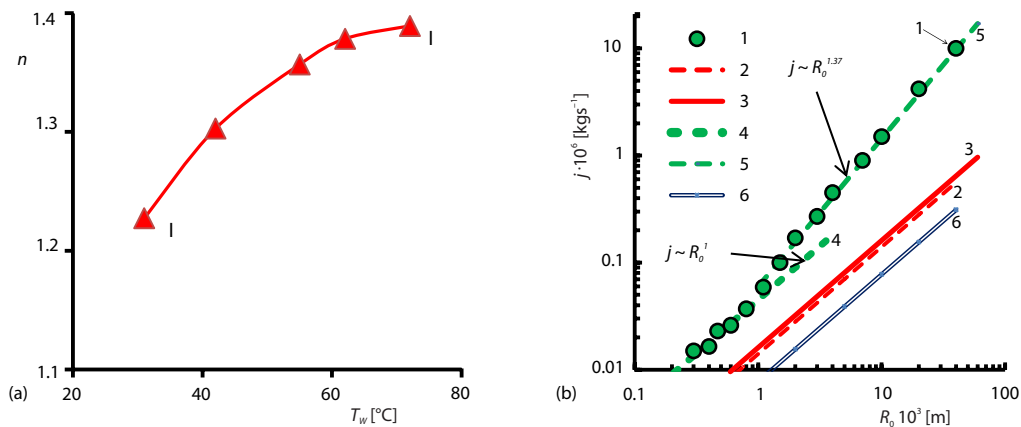


Figure 4. (a) Exponent n in the law of droplet evaporation vs. wall temperature T_w (for $R_0 > 3$ mm), (b) effect of different factors on evaporation rate ($T_w = 70$ °C): 1 – experimental data, curves 2-5 are obtained by relationships 2-5, and curve 6 is obtained regardless of Marangoni number

The calculated curves for evaporation rate, j , are shown in fig. 4(b). The structured roughness causes an increase in R_0 , therefore, the droplet height, h , decreases. A decrease in h leads to an increase in T_s and ρ_s . The structured wall raises circulation in liquid, which increases the temperature and equilibrium vapor density $\rho_s = f(T_s)$. For small droplets ($R_0 < 1$ mm), the effect of natural convection in a liquid (Ra_l) is negligibly low. In this case, we can neglect Marangoni and Rayleigh numbers in eq. (1).

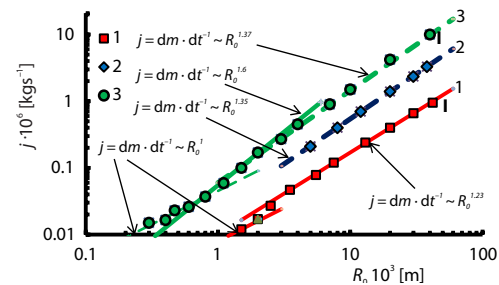


Figure 3. Rate of droplet evaporation ($j = dm/dt$) depending on the initial radius of droplet base R_0 : 1-3 – experimental data; for 1 – $T_w = 28$ °C, 2 – $T_w = 52$ °C, 3 – $T_w = 70$ °C

Curves 2 correspond to eq. (2) taking into account steam diffusion only:

$$j_1 = 4R_0 D \Delta \rho \quad (2)$$

For large radii ($R_0 > 2$ mm), Marangoni and Rayleigh numbers play a decisive role in heat and mass transfer in the droplet, and this leads to a decrease in ΔT and an increase in T_s . Since, in calculations the values of T_s are taken from the experiment (thermal imaging measurements), the interface temperature is to consider all key factors, and calculations may be made by eq. (2). Curve 3 correspond to eq. (3) (takes into account diffusion and Stefan flow):

$$j_2 = 4R_0 D(\rho_v + \rho_a) \ln(1 + B_M) \quad (3)$$

For qualitative analysis it is convenient to consider the limiting cases for a large droplet, when its radius is approximately equal to the heater radius, and for a small droplet ($R_0 < 1$ mm). For the case of a large droplet ($R_0 = d$, where d is the heater diameter, fig. 1), only the vapor-diffusion layer δ_{cv} , fig. 2(b), will develop above the droplet surface. In this case, the diffusion will be determined by Ra_v . For the case of a small droplet $\delta_{cv}/\delta_{cg} \ll 1$, fig. 2(a), only the convection over the heater (Ra_a) assumingly plays a decisive role. As it can be seen from fig. 4(b), Curve 4 ($j \sim R_0^1$) corresponds to experimental points (1) for small droplets. Curve 4 (j_4) (for small droplets) is obtained as a sum of evaporation fluxes due to diffusion, Stefan flow and free convection of air (j_3) over the heated wall:

$$j_4 = j_2 + j_3 = j_2 + a_1 (Ra_a) n_1, \quad j_3 = a_1 \left(\frac{g\beta\Delta T}{\nu^2} \right) (R_0^3)^c = a_1 \left[\frac{g\beta\Delta T R_0^3}{\nu^2} \right]^{0.33} \sim (R_0)^1 \quad (4)$$

Thus, the exponent for the Rayleigh number is not equal to 0.25. The mass transfer will be more dependent on physical properties of gas and ΔT than the heat transfer. A significant violation of analogies between heat transfer, mass transfer and friction for the case of porous injection through a permeable wall has been shown experimentally and theoretically. In the case of porous injection, the flow of gas j through the wall does not depend on heat and mass transfer. When the droplet evaporates, the flow is associated both with gas convection and heat transfer in liquid as well as with thermal conductivity of solid wall and this greatly complicates simulation and analysis.

According to expression (1), the vapor flux in the case of large droplets takes the form of expression (5):

$$j_7 = j_5 + j_6, \quad j_5 = aRa_l^n + f(Ma) + 4rD(\rho_v + \rho_a) \ln(1 + B_M), \quad j_6 = a_2 Ra_v^{n_2} \quad (5)$$

For large droplets with $R_0 \approx 20$ mm, fig. 4(b), the experimental data are almost 10 times higher than the calculation by Curve 3. Due to the predominant role of gas convection in eq. (5), the vapor flux can be presented in the approximate form expression (6):

$$j_7 \approx j_6 = a_2 \left(\frac{g\beta\Delta T}{\nu^2} \right)^d (R_0^3)^d = a_2 \left[\frac{g\beta\Delta T (R_0^3)}{\nu^2} \right]^{0.457} \sim (R_0)^{1.37} \quad (6)$$

High value of $n_2 \approx 0.46$ indicates a significant increase in the role of gas convection for large droplets relative to small ones and a higher dependence of mass transfer on ΔT . Curve 6 in fig. 4(b) is plotted regardless of Marangoni forces. In this case, ΔT increases, and T_s decreases from 60-45°C. As a result, steam flow j halves.

It is important to note that exponents n_1 and n_2 in eq. (1) are determined by the competing influence of three key factors: gas-liquid interaction (convection in gas), convection and thermal conductivity in liquid, and interaction between liquid and solid body (a wall thermal

inertia). The thermal inertia of the wall is important for non-stationary heat transfer and leads to a decrease in the wall temperature under the droplet. Only some time after a cold drop is put on a hot wall, the wall temperature under the droplet reaches the quasi-stationary thermal regime. The time of thermal relaxation depends on thermal conductivity of the wall and the droplet size.

Therefore, at $T_w = 70$ °C, neglecting convection in liquid leads to a 70-80% underestimation of the evaporation rate, a neglect of thermal inertia of the wall results in a 30% reduction in j , and neglecting free convection in gas for large droplets leads to a multiple underestimation of evaporation rate.

Conclusion

With an increase in the wall temperature from 31-72 °C and with an increase in the initial droplet diameter, the exponent n in the evaporation law increases from 1-1.37. Under the transitional regime, the exponent $n = 1.6$ and reaches its maximum. Usually, a linear relationship of the mass-flow of steam and droplet radius is considered in the simulation of droplet evaporation. The exponent n increases with an increase in T_w and R_0 . There are two boundary layers in a free-convective flow: the diffusion steam layer on the droplet surface δ_{cv} and the boundary-layer of air on the surface of the heated cylinder δ_{cg} with a diameter exceeding that of a droplet. A neglect of free convection more than 10 times underestimates simulation results compared with experimental data. Since convection has a predominant influence in the steam-gas medium, but not in a liquid droplet, the role of roughness and morphology of the wall structure is a less significant factor. Thus, to simulate the evaporation process it is important to take into account the dynamic and thermal prehistory of the convective air-flow. If we place the key criteria according to their degree of influence on the rate of droplet evaporation, their sequence will be as follows.

- Convection in a steam-gas medium;
- The effect of wall roughness, wettability, and convection in a liquid on changes in the temperature of the droplet interface T_s ;
- Thermal inertia of a metal wall with high thermal diffusivity.

However, for a wall with low thermal diffusivity, the role of 3rd factor increases sharply.

Acknowledgment

This work was carried out at the Kutateladze Institute of Thermophysics SB RAS and financially supported by the Russian Science Foundation (project number 15-19-10025).

References

- [1] Sirringhaus, H., *et al.*, High-Resolution Inkjet Printing of All-Polymer Transistor Circuits, *Science*, 290 (2000), 5499, pp. 2123-2126
- [2] Arcamone, J., *et al.*, Evaporation of Femtoliter Sessile Droplets Monitored with Nanomechanical Mass Sensors, *Journal Phys. Chem., B* 111 (2007), 45, pp. 13020-13027
- [3] Fang, X., *et al.*, Drying of DNA Droplets, *Langmuir*, 22 (2006), 14, pp. 6308-6312
- [4] Wu, Y., Modeling and Experimental Study of Vapor Phase-Diffusion Driven Sessile Drop Evaporation, *Appl. Therm. Eng.*, 70 (2014), 1, pp. 560-564
- [5] Bonaccorso, E., *et al.*, Fabrication of Microvessels and Microlenses from Polymers by Solvent Droplets, *Appl. Phys. Lett.*, 86 (2005), 12, 124101
- [6] Misyura, S. Y., Wall Effect on Heat Transfer Crisis, *Exp. Therm. Fluid Sci.*, 70 (2016), Jan., pp. 389-396
- [7] Bogya, E. S., *et al.*, Surface Pinning Explains the Low Heat Transfer Coefficient between Water and Carbon Nanotube Film, *Carbon*, 100 (2016), Apr., pp. 27-35
- [8] Carrier, O., *et al.*, Evaporation of Water: Evaporation Rate and Collective Effects, *Journal Fluid Mech.*, 789 (2016), July, pp. 774-786

- [9] Saada, M. A., *et al.*, Numerical Investigation of Heat Mass Transfer of an Evaporating Sessile Drop on a Horizontal Surface, *Journal Phys. Fluids*, 22 (2010), 11, pp. 112-115
- [10] Brutin, D., *et al.*, Infrared Visualization of Thermal Motion Inside a Sessile Drop Deposited onto a Heated Surface, *Exp. Therm. Fluid Sci.*, 35 (2011), 3, pp. 521-530
- [11] Seine, K., *et al.*, On the Effect of the Atmosphere on the Evaporation of Sessile Droplets of Water, *Journal Phys. Fluids*, 21 (2009), 6, 062101
- [12] Shahidzadeh-Bonn, N., *et al.*, Evaporating Droplets, *Fluid Mech.*, 549 (2006), Feb., pp. 307-313
- [13] Mollaret, R., *et al.*, Experimental and Numerical Investigation of the Evaporation into Air of Drop on a Heated Surface, *Chem. Eng. Res. Des.*, 82 (2004), 4, pp. 471-480
- [14] Dunn, G. J., *et al.*, The Strong Influence of Substrate Conductivity on Droplet Evaporation, *Journal Fluid Mech.*, 623 (2009), Mar., pp. 329-351
- [15] Murisic, N., Kondic L., On Evaporation of Sessile Drops with Moving Contact Lines, *Journal Fluid Mech.*, 679 (2011), July, pp. 219-246
- [16] David, S., *et al.*, Experimental Investigation of the Effect of Thermal Properties of the Substrate in the Wetting and Evaporation of Sessile Drops, *Colloids Surfaces*, 298 (2007), 1-2, pp. 108-114
- [17] Hu, H., Larson, R. G., Evaporation of a Sessile Droplet on a Substrate, *Journal Phys. Chem., B* 106 (2002), 6, pp. 1334-1344
- [18] Bleiker, G., Specht, E., Film Evaporation of Drops of Different Shape Above a Horizontal Plate, *Int. J. of Therm. Sci.*, 46 (2007), 9, pp. 835-841
- [19] Birdi, K. S., *et al.*, A Study of the Evaporation Rates of Small Water Drop Placed on a Solid Surface, *Journal Phys. Chem.*, 93 (1989), 9, pp. 3702-3703
- [20] Toda, S., A Study of Mist Cooling, 2nd Report: Theory of Mist Cooling and Its Fundamental Experiments, *Heat Transfer Jpn. Res.*, 3 (1972), 1, pp. 1-42
- [21] Bonacina, C., *et al.*, Dropwise Evaporation, *Trans. ASME. J. Heat Transfer*, 101 (1979), 3, pp. 441-446
- [22] Wachters, L. H., Westerling, N. A., The Heat Transfer from a Hot Wall to Impinging Water Drops in Thespheroidal State, *Chem. Eng. Science*, 21 (1966), 11, pp. 1047-1056
- [23] Tartarini, P., *et al.*, Dropwise Cooling: Experimental Tests by Infrared Thermography and Numerical Simulations, *Applied Thermal Engineering*, 29 (2009), 7, pp. 1391-1397
- [24] Seki, M., *et al.*, Transient Temperature Profile of a Hot Wall Due to an Impinging Liquid Droplet, *ASME J. Heat Transfer*, 100 (1978), 1, pp. 167-169
- [25] Bussmann, M., *et al.*, Modeling the Splash of a Droplet Impacting a Solid Surface, *Physics of Fluids*, 12 (2000), 12, pp. 3121-3132
- [26] Senda, J., *et al.*, The Heat-Transfer Characteristics of a Small Droplet Impinging upon a Hot Surface, *JSME International Journal, Series II*, 31 (1988), 1, pp. 105-111
- [27] Pasandideh-Fard, M., *et al.*, Capillary Effects during Droplet Impact on Solid Surface, *Phys. Fluids*, 8 (1996), 3, pp. 650-659
- [28] Nakoryakov, V. E., Misyura, S. Y., Bubble Boiling in Droplets of Water and Lithium Bromide Water Solution, *Journal of Engineering Thermophysics*, 25 (2016), 1, pp. 24-31
- [29] Gau, H., *et al.*, Liquid Morphologies on Structured Surfaces: From Microchannels to Microchips, *Science*, 283 (1999), 5398, pp. 46-49
- [30] Carle, F., Contribution of Convective Transport to Evaporation of Sessile Droplets: Empirical Model, *Int. J. of Thermal Sciences*, 101 (2016), Mar., pp. 35-47
- [31] Valiullin, T. R., Strizhak, P. A., Influence of the Shape of Soaring Particle Based on Coal-Water Slurry Containing Petrochemicals on Ignition Characteristics, *Thermal Science*, 21 (2017), 3, pp. 1399-1408
- [32] Voitkov, I. S., *et al.*, Temperature of Gases in Trace of Water Droplets during Their Motion in a Flame, *Thermal Science*, 22 (2017), 1, pp. 335-346
- [33] Volkov, R. S., *et al.*, Fire Extinction of Forest Fuels by Droplets and Water Film, *Thermal Science*, 22 (2017), 1, pp. 347-358
- [34] Chandramohan, A., *et al.*, Spatiotemporal Infrared Measurement of Interface Temperatures during Water Droplet Evaporation on a Nonwetting Substrate, *Appl. Phys. Lett.*, 110 (2017), 4, pp. 041605
- [35] Misyura, S. Y., Contact Angle and Droplet Heat Transport during Evaporation on Structured and Smooth Surfaces of Heated Wall, *Applied Surface Science*, 414 (2017), Aug., pp. 188-196
- [36] Lebedev, *et al.*, Effect of Flow Acceleration and Initial Turbulence Level on Velocity Fluctuations, *Fluid Dynamics*, 28 (1993), 5, pp. 624-629
- [37] Misyura, S. Y., The Influence of Porosity and Structural Parameters on Different Kinds of Gas Hydrate Dissociation, *Scientific Reports*, 6 (2016), July, 30324

- [38] Kutateladze, S. S., *Fundamentals of Heat Transfer Theory*, Atomizdat, Moscow, 1979
- [39] Borishansky, V. M., Heat Transfer to Liquid, Flowing Free from the Surface Heated above the Temperature of Boiling, in: *The Problems of Heat Transfer at a Change of Substance Aggregate State*, (Ed. S. S. Kutateladze), Gosudarstvennoe Energeticheskoe Izd., Moscow-Leningrad, 1953