Study of total heat losses in experiments with shear-driving liquid films in a mini-channel with intense heating

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Abstract. Recent works by the authors have shown the high efficiency of using thin liquid films, moving under the action of gas in a flat mini-channel, for removing high heat fluxes. In the present work, we carefully study the total heat losses in such a system. Total heat losses are divided into two portions: heat losses from the heating coil to the atmosphere and heat spreadings from the heater to the substrate. The relative values of the total heat losses prior to the crisis are decreasing with increasing critical heat flux and do not exceed approximately 20% for the critical heat flux between 500 and 900 W/cm\textsuperscript{2}.

1. Introduction

The problem of microelectronics cooling is nowadays one of the most challenging problems of thermophysics. At this point in the production of processors, a technical process with a resolution of 22 nm is used. The number of transistors on a chip reaches an amount of about 2x10\textsuperscript{9}. The heat flux in electronic converters and inverters in hybrid cars now reaches up to 100-200 W/cm\textsuperscript{2}. The average heat flux on mass-produced computer chips and other electronic devices is up to 200-300 W/cm\textsuperscript{2}. While localized in the area of \textasciitilde100 micron\textsuperscript{2} heat flux reaches 1 kW/cm\textsuperscript{2} and higher [1]. A number of military and technical applications already require heat flux removal from localized zones of up to 3-5 kW/cm\textsuperscript{2} and more (i.e. GaN transistors). It should be noted that the heat flux on the sun surface is about 6 kW/cm\textsuperscript{2}. The situation with heat removal in microelectronics is even more complicated due to the transition of the electronics industry to the production of three-dimensional chips where multiple substrates with electronic components are installed in parallel with a distance of 50-100 microns. Therefore, development of advanced cooling techniques, capable to cope with such thermal loads, is needed.

There are three well-known ways of cooling intense localized heat sources: 1) microchannel boiling [2], 2) spray cooling [3], and 3) micro-jet cooling [4]. In [5, 6], a new way of efficient cooling was proposed. In this way of cooling the intense evaporation of a thin liquid film, moving under the action of gas flow friction in a flat micro-/mini-channel is used for heat removal. In [7], using this technique the possibility of removing heat fluxes with a density of up to 1 kW/cm\textsuperscript{2} from a heating area of 1x1 cm\textsuperscript{2} was shown, which is an order of magnitude higher than critical heat flux in falling liquid films [8]. In [9-11] stable operation of such system at different heights of the channel (from 0.17 to 2.0 mm) and at different angles of inclination to the horizon (from 0 to 360°) was demonstrated. In all works [5-7, 9-11], it was assumed that the heat losses were negligible, that is, all the electrical heat from the heating coil goes through the heater surface. However, for small flow rates of liquid and gas...
(i.e. when the heat transfer rates from the heater to the liquid film are comparatively low), the heat losses can be significant. In [5-7, 9-11] the critical heat fluxes are given without taking into account heat losses since the test sections in [5-7, 9-11] did not allow determining the local heat flux in the heater. In the current work we use a new test section with a similar design as in [5-7, 9-11] with the exception that, using two thermocouples with junctions placed one above the other in the heater, we can determine the local heat flux. The main task of the current work is to investigate the total heat losses in such experiments.

2. Experimental setup

The experimental setup consists of a test section, closed liquid line, open gas line, measuring and control systems. As a working gas, atmospheric air is used. The relative humidity of air is 20–40%, whereas the initial temperature is 25 °C. Ultrapure distilled water Milli-Q with the initial temperature of 25 °C is used as the working liquid. Figure 1 shows a schematic of the test section. The substrate is a plate of stainless steel. The copper cylinder, serving as a heater, is flush-mounted in the steel plate. Cylinder heating is carried out by passing an electric power through the nichrome coil, attached to its bottom part. For the thermal insulating of the heater, a layer of mineral wool is used. The copper heater surface is designed to simulate the surface of a typical electronic chip and has a square shape of 10x10 mm². Such construction of the heater provides a condition of the heater surface temperature to be nearly constant. The working surface (stainless steel plate together with the heater surface) was roughly polished so that the root mean square (RMS) roughness is 0.5 µm. The test section is covered by a transparent optical glass plate. The height of the channel is 1 mm.

![Figure 1. Schematic of the test section. 1 – gas inlet; 2 – liquid inlet; 3 – nichrome coil; 4 – copper cylinder; 5 – thermal insulation; 6 – steel plate; 7 – glass plate; 8 – outlet of liquid and gas.](image)

To measure the temperature, a system of chromel-copel thermocouples was built into the test section as shown in Fig. 2 (left). Thermocouples 1 and 3 are used to determine the local heat flux, \( q_{\text{loc}} \), whereas thermocouple 2 is used to determine the temperature of the surface of the heater, \( T_w \), (taking into account the depth of the thermocouple junction location). The thermocouples 4 and 5 are used to determine the temperature of the steel plate, as well as the heat spreadings from the heater to the substrate. The diameter of the thermocouple junction is 0.5 mm.

The test section was specially designed to achieve high heat fluxes, so that embedding of thermocouples in the heater could not be made along the isotherms and was made through the holes along the copper cylinder from the bottom up, as depicted in Fig. 2 (left). In order to reduce inaccuracies in the temperature measurement (due to heat transport from thermocouple wires to the junction), the thermocouples are wound with a layer of insulating material along the entire length, with the exception of the thermocouple junctions which are in contact with the metal (through a layer of thermal paste), Fig. 2 (left). To verify the correctness of the temperature measurements by thermocouples, we compare temperatures of the heater surface obtained in the current work with those obtained in experiments [11] where thermocouples were embedded along the isotherms, as shown in Fig. 2 (right). In Fig. 3 the comparison of the heater surface temperatures, \( T_w \), measured just prior to the heat transfer crisis, vs. the critical heat flux \( q_{\text{total,cr}} \) (calculated without taking into account the heat losses) is presented (for different Reynolds numbers of liquid \( R_e \) and different superficial gas
velocities $U_{sg}$). A good agreement (in view of data scattering) is seen for $q_{total,cr}>150$ W/cm$^2$ (where the heater surface temperature is weakly dependent on $q_{total,cr}$ for both experiments).

**Figure 2.** Location of thermocouples in the test section of current work (left) and in that of [11] (right).

**Figure 3.** Comparison of the heater surface temperatures, just prior to crisis occurrence, obtained in the current work (red empty circles) with those obtained in [11] (blue filled circles).

**Figure 4.** Time variation of temperatures at different depths in the heater for $Re_l = 70$, $U_{sg} = 26.9$ m/s.

**Figure 5.** The dependence of the heat flux on time for $Re_l = 70$, $U_{sg} = 26.9$ m/s.
It should be noted that the calculation of the local heat flux, \( q_{\text{loc}} \), is based on the difference between readings of thermocouples 1 and 3, Fig. 2 (left), therefore the uncertainty in \( q_{\text{loc}} \) is much lower than uncertainty in temperature measurements, since systematic uncertainties in temperatures measured by thermocouple 1 and 3 are subtracted from each other. Thus, we can conclude that correctness of temperature measurements and especially correctness of local heat flux determination is enough justified in the current experiment.

Figure 4 shows a typical change of temperatures in the course of increasing heat flux up to the crisis; shown are the heater surface temperature, \( T_w \), and temperatures \( T1 \) and \( T3 \) (measured by thermocouples 1 and 3, respectively). Figure 5 shows typical time dependencies of the total heat flux, \( q_{\text{total}} \) (calculated based on Joule heat generated on the nichrome coil) and the local heat flux, \( q_{\text{loc}} \) (determined based on measurements by thermocouples 1 and 3) in the process of increasing heat flux.

3. Determination of total heat losses

3.1. Heat losses to the atmosphere

The heat losses to the atmosphere are the part of the generated heat that is going to the atmosphere from the nichrome coil, not reaching the heater surface. We determine the relative heat losses into the atmosphere as \( \left( 1 - \frac{q_{\text{loc}}}{q_{\text{total}}} \right) \cdot 100\% \), where the local heat flux, \( q_{\text{loc}} \), is determined based on the measurement of two thermocouples, embedded in the heater (thermocouples 1 and 3 in Fig. 2 (left), the distance between the thermocouples being 2.9 mm); while the total heat flux, \( q_{\text{total}} \), is determined based on total Joule heat generation (i.e. without taking into account heat losses). Figure 6 shows the relative heat losses prior to the crisis, vs. the critical heat flux, \( q_{\text{total},\text{cr}} \). For data in Fig. 6, both \( q_{\text{loc}} \) and \( q_{\text{total}} \) are taken at the next to the last step of the increasing heat flux (i.e. just prior to the crisis occurrence), while \( q_{\text{total},\text{cr}} \) is taken at the very last step of the increasing heat flux (i.e. at the crisis occurrence). In the experiment, the difference between the two last steps is usually less than 5% (see Fig. 5). Different points in Fig. 6 correspond to different Reynolds numbers of liquid and different superficial gas velocities. From Fig. 6 it is seen that the relative values of heat losses into atmosphere generally decrease with increasing critical heat flux and for \( 250 \text{ W/cm}^2 < q_{\text{total},\text{cr}} \leq 935 \text{ W/cm}^2 \); the relative heat losses are within \( 16 \pm 4\% \).

![Figure 6](image)

**Figure 6.** The relative heat losses to the atmosphere (just prior to the crisis) versus the critical heat flux (determined based on the electrical power generated on the nichrome coil) for different Reynolds number of liquid (shown in legend) and different gas velocities (\( U_{sg} = 2.6 \div 54 \text{ m/s} \)). Blue dotted line shows a generalization of the data.
3.2. The heat spreading to stainless steel plate

Thermocouples 4 and 5 embedded in the substrate, Fig. 2 (left), were used to determine the part of heat, “spreading” from the heater into the stainless steel plate. The thermal conductivity of the stainless steel is 15 W/mK which is about 30 times lower than that of copper (400 W/mK). We determine relative heat spreadings as a ratio between the heat that goes to the steel plate, \( q_{steel}S_{cont} \), and the heat that goes to the heater surface, \( q_{loc}S_{heater} \). Where \( q_{steel} \) is average heat flux from the heater to the steel plate, determined using thermocouples embedded in the steel plate and in the heater, \( S_{cont} \) – area of contact between the copper heater and the stainless steel plate, \( S_{heater} \) – area of the heater surface.

Fig. 7 shows the relative heat spreadings to the substrate, \( \frac{q_{steel}S_{cont}}{q_{loc}S_{heater}} \cdot 100\% \), prior to the crisis, depending on the critical heat flux, \( q_{total,cr} \), for different Reynolds numbers of liquid and different superficial velocities of gas. For data in Fig. 7, both \( q_{steel} \) and \( q_{loc} \) are taken at the next to the last step of the increasing heat flux (i.e. just prior to the crisis occurrence, see Fig. 5). From Fig. 10 it is seen that the relative values of heat spreadings decrease with increasing critical heat flux as \( \sim 1/q_{total,cr} \). This is because the absolute value of the heat spreadings to the substrate, \( q_{steel}S_{cont} \), prior to the crisis is almost non-dependent on the critical heat flux and does not go beyond 35 W/cm\(^2\) in our experiments. According to thermocouple measurements, the temperature of the heater surface prior to the crisis is also weakly dependent on the critical heat flux and does not exceed 136 °C, Fig. 3. From Fig. 10 one can see that for \( q_{total,cr} > 400 \) W/cm\(^2\), the relative spreadings to the substrate do not exceed about 10\%, while for \( q_{total,cr} > 700 \) W/cm\(^2\), the spreadings are less than about 5\%.

Figure 7. The relative heat spreadings to the substrate (just prior to the crisis) versus the critical heat flux (determined based on the electrical power generated on the nichrome coil) for different Reynolds numbers of liquid (shown in legend) and different gas velocities (\( U_{sg} = 2.6 \div 54 \) m/s).

4. Critical heat flux with account for total heat losses

The equation for “true” heat flux \( q_{whl} \) (with account for total heat losses) can be represented as:
However, this equation cannot be directly used for calculation of “true” value of the critical heat flux, as at the moment of the crisis the heater is dried out so that the temperature of the heater starts to raise very sharply (see the last step in Fig. 4) leading to incorrect determination of $q_{loc}$ (see the last step in Fig. 5). To obtain “true” value of the critical heat flux, $q_{cr}$, we are based on the critical heat flux, $q_{total,cr}$ (without taking into account heat losses), determined at the moment of the crisis, and adjust it for the total heat losses, using $q_{whl}$ determined at the previous step (just prior to crisis), where we can correctly determine $q_{loc}$ (see the next to the last step in Fig. 5). Thus, we get the following equation for “true” value of the critical heat flux:

$$q_{whl} = q_{loc} - \frac{q_{steel}S_{cont}}{S_{heater}}$$

where $q_{total,cr}$ is taken at the moment of the heat transfer crisis occurrence, whereas $q_{whl}$, $q_{total}$, $q_{loc}$, $q_{steel}$ are taken just prior to the crisis occurrence. It should be noted, that during our experiments, when approaching to the critical heat flux, the increment in the heat flux is usually equal or less than 5%, so the above-mentioned equation can be considered as a good estimation of “true” value of the critical heat flux.

5. Conclusion

The total heat losses in our experiments with locally heated liquid films moving under the action of the gas flow in a mini-channel have been determined. Total heat losses are divided into heat losses into the atmosphere and heat spreadings into the substrate. For the critical heat flux between 250 and 935 W/cm$^2$ the heat losses into the atmosphere are within 16 ± 4%. Relative values of the heat spreadings into the substrate decrease in inverse proportion to the critical heat flux. For the critical heat flux above 400 W/cm$^2$, the heat spreadings do not exceed 10%. The relative values of the total heat losses decrease with increasing critical heat flux and do not exceed about 20% for the heat flux between 500 and 900 W/cm$^2$.

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