Verification of the method for determining heat fluxes and heat loses in experiments with a shear-driving liquid film

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Abstract. The cooling of microelectronic equipment is currently one of the most difficult problems of thermal physics. To achieve high heat fluxes and reduce the volume of liquid used, the technological solution looks especially promising, where heat is transferred from the fuel element to a thin, intensely evaporating film of liquid moving under the action of friction of a forced gas flow in a mini- or micro-channel. In recent works, the study of heat losses and determination of the true heat flux in experiments with an intensely evaporating liquid film moving under the action of a gas flow in a mini-channel was carried out. In this paper, the task is to verify the method for determining heat losses. Heat transfer in the convective flow regime is investigated. The experimental and calculated Nusselt numbers in convection experiments with complete channel flooding are in good agreement. In addition, the experimental and calculated Nusselt numbers for film flows are in good agreement. This indicates the correctness of the method for determining heat flux and heat losses in previous studies.

1. Introduction

The cooling of microelectronic equipment is currently one of the most difficult problems of thermal physics. The growth of human needs for computing power has not slowed down at all and even accelerated due to the rapid development of cloud technologies, the Internet of things and deep machine learning. The number of devices that are constantly connected to the Internet, according to the forecast of Cisco, will exceed 50 billion in 2020, the number of sensors in these devices will exceed 1 trillion. Processing exponentially increasing flows of information requires further growth in computing power.

The integrated heat flux in high-performance modern microprocessors can reach 150-200 W/cm$^2$. At the same time, the local heat flux in individual regions of the size of the order of 100 µm - 2000 µm can exceed a value of 1000 W/cm$^2$ or more [1-3]. High rates of development of electronics (chips, laboratories in chips, LED technology, lasers, radars, converters, etc.) pose a significant intensification of heat removal processes. 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 microcircuits, where several substrates with electronic components are installed in parallel with a distance of 50-100 microns. Volumetric heat flux densities in three-dimensional chips reach 10 kW/cm$^3$ [1].
To achieve high heat fluxes and reduce the volume of liquid used, the technological solution looks especially promising, where heat is transferred from the fuel element to a thin, intensely evaporating film of liquid moving under the action of friction of a forced gas flow in a mini- or micro-channel [4-5]. The latent heat of evaporation in two-phase flow usually provides much greater cooling capacity than single-phase flow. The implementation of this method – shear-driving liquid films – was presented in [5-6]. Further research showed the stable operation of such a method under various conditions, including different channel inclination angles to the horizon (0-360°) [7] and the stable operation at a different height of channel (0.17-2 mm) [8]. The limiting values of specific heat fluxes removed by cooling liquids from heat-generating surfaces are limited by the heat transfer crisis. Therefore, much attention is paid to studies of critical heat loads, as well as mechanisms for the development of critical phenomena [9]. In [10] it was showed that “shear-driving liquid films” is capable of removing heat fluxes of up to 1 kW/cm² (heating area of 1x1 cm²).

In [5-7, 9-10] the critical heat fluxes are given without taking into account heat losses. It was assumed that the heat losses were negligible. In the works [11,12] the authors used a new test section with a similar design as in [5-7, 9-10]. To determine the local heat flux, two thermocouples with junctions placed one above the other in the heater were used. The estimates obtained from measurements of the thermocouples located in the heater show that the relative heat losses into the atmosphere do not exceed 15% at heat fluxes over 400 W/cm². The estimates obtained from measurements of the thermocouples located in the steel plate show that the relative heat spreading into the steel plate do not exceed 6% at heat fluxes over 350 W/cm². In this way, the value of the total heat loses (heat loses into the atmosphere and heat spreading into the steel plate) at heat fluxes over 400 W/cm² do not exceed approximately 20%, [11,12]. The main task of the current work is verification of the used method for determining heat losses.

2. Experimental setup
An experimental test section was developed for research. The conception of the experimental test section is shown in the figure 1. Test section is a sealed channel with inlet and outlet openings for liquid and gas. Stainless steel is used to create the channel bottom. Flush-mounted into stainless steel, copper cylinder is used as a heater. The Joule heat, when passing a current through a nichrome coil, wound around the copper cylinder lower part, heats it. The square area of 10x10 mm² is the copper heater surface. The temperature on the surface of the such heater takes constant values, T_w=constant, which is confirmed by measurements of the thermocouples. The plate of transparent optical glass is the upper part of the channel. In the experiments height of the channel is equal to 1 mm, width of the channel is equal to 36 mm. The air compressor supplies atmospheric air (working gas) to the test section with an initial temperature of about 25 degrees Celsius and a relative humidity of 20-40%. The gear pump supplies the working fluid – Milli-Q water (ultra-pure distilled water) to the test section with an initial temperature of 25 degrees Celsius.

![Figure 1](image-url)  
*Figure 1. Conception of the test section. 1 – gas inlet; 2 – liquid inlet; 3 – thermocouples; 4 – nichrome coil; 5 – copper cylinder; 6 – thermal insulation; 7 – steel plate; 8 – glass plate; 9 – outlet of liquid and gas.*
Working surface temperatures are determined by means of thermocouples embedded in the steel plate and cooper heater. The Joule power released on the nichrome coil determines the total heat flux to the heater surface. The heat flux is controlled by thermocouples measurements. Wrapping the heater with a layer of thermal insulation was used to reduce heat losses into the atmosphere (see 6 on figure 1). Large difference in the thermal conductivity of stainless steel and copper (15 W/mK for stainless steel and 400 W/mK for copper) as well as by a significant thermal resistance on the stainless-steel – copper interfaces provide a small relative heat spreadings. For better wettability of the surface of the heater and the stainless-steel plate was polished. The study of the morphology of the treated surface was carried out on the atomic force microscope Solver Pro NT MDT in semi-contact mode on various parts of the surface of the heater: RMS roughness is about 0.5 microns.

3. Results
To verify the method used for determining the heat flux and heat losses, an experiment was carried out in the convective mode with complete flooding of the channel for subsequent comparison of the experimental data with numerical calculations in the software. To determine the heat losses and the true heat flux in the experiment, a method from [11,12] was used. In the numerical simulation package, the conjugate heat transfer was calculated using the finite element method. Comparison of the experimental and numerical data were shown in figure 2. The experiment and numerical study was carried out at a liquid flow rate of 60 - 1500 ml/min (Re$_l$ = 80 – 860) for various heat fluxes.

![Figure 2](image)

**Figure 2.** Convective heat transfer in a 0.6 mm channel. Comparison of Nusselt numbers for numerical simulation (1), experiment (2) and theoretical dependence (3) from [13].

In addition, the experimental data were compared with the Nusselt numbers obtained from the theoretical calculation of heat transfer in the thermal initial section of a flat tube for the boundary conditions T = const [13] (figure 2, line 3).

$$
\overline{Nu} = 1.65 \left( Pe \frac{h}{\chi} \right)^{1/3}
$$
It should be noted that for the Reynolds numbers $Re_l = 80 – 860$ thermal initial length is $l_T = 20 – 210$ mm, hydrodynamic initial length is $l_H = 0.5 – 8$ mm. Thus, at any flow rate, for our experiment, the entire heater (distance from the liquid nozzle 38 mm, heater length 10 mm) is located in the developed hydrodynamic section, but at the initial thermal section, which justifies the possibility of comparison with the theoretical dependence from [13].

![Figure 3. Heat transfer to a liquid film moving under the action of a gas flow in a 1 mm channel. Experiment (1) and (3), Nusselt from [14,15] (2) and (4).](image)

In figure 3 Nusselt numbers calculated from experiments with a liquid film moving under the action of a gas flow ($1 – Re_l = 30$ and $U_{sg} = 8$ m/s; $3 – Re_l = 30$ and $U_{sg} = 16$ m/s) are compared with the calculation for a free-falling liquid film for the boundary condition $T = \text{const}$ [14,15].

$$Nu = 1.88 \left[ 1 + 0.029 \left( \frac{Pe}{X} \right)^{4/3} \right]^{1/4}$$

Good agreement is observed at the initial stages of the experiment. In turn, the inconsistency of the extreme points on the graphs for both flow regimes can be explained by the transition from the convective process of heat transfer to boiling in the later stages of the experiment and the associated intensification of heat transfer.

4. Conclusion
Thus, the method for calculating heat losses in [11,12] was verified. It is can be seen from the graphs in figure 2, the experimental and calculated Nusselt numbers in convection experiments are in good agreement. In addition, the experimental and calculated Nusselt numbers for film flows were compared (figure 3). Which are also in good agreement with each other. This testifies to the correctness of the method for determining the heat flux and heat losses in [11,12].

Acknowledgments
The study was performed under the support of Russian Scientific Foundation (Grant No. 18-79-10258).

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