Pool boiling heat transfer on minichannels with porous structure

Norbert Dadas[1], *Robert* Pastuszko[1*], and *Robert* Kaniowski[1]

[1]Kielce University of Technology, Faculty of Mechatronics and Mechanical Engineering, al. 1000-lecia Państwa Polskiego 7, PL-25-314 Kielce, Poland

Abstract. The work focused on the investigations and comparison of pool boiling heat transfer on the specimens with in 5 mm deep and 1 mm wide parallel mini-channels to the specimens with additional porous material (copper foam or mesh) between the fins. The experiments were carried out with boiling FC-72 at atmospheric pressure. Many works show that enhanced surfaces with open minichannels improve heat transfer performance substantially. The use of additional porous material led to the highest heat transfer coefficients in the range of 100 to 230 kW/m². The coefficients obtained were 15% higher and the superheat was 15% lower compared to the plain smooth minichannel without porous material.

1 Introduction

The widespread miniaturization of components, mainly electronic ones, and increasing their efficiency lead to the intensification of the heat they generate. The heat from the microprocessor can reach 2000 – 3000 kW/m^2, and in some cases maximum heat fluxes up to 15000 kW/m^2 have been reported [1]. Cooling by phase change during boiling ensures efficient passive heat dissipation and fluid circulation [2]. Extension of the heat transfer surface in the form of micro- and minichannels is very common, but the boiling mechanism is not fully described. The use of boiling on surfaces with minichannels allows for obtaining much higher heat transfer coefficients compared to forced convection. The liquid enters the channels at random locations [3]. Porous structures are added to increase the surface of heat transfer and to initiate a capillary pressure that directs the fluid flow. The pores in the material increase the area of bubble nucleation [4]. The decrease in temperature on the heating surface is due to the formation, growth, and detachment of the vapour bubble [5]. Partial filling of channels with porous surfaces in the form of meshes or copper foam should direct the flow in the minichannels. The added filling will allow the liquid to be sucked in, and the formation, growth, and detachment of bubbles will occur in the gaps between the porous material.

2 Experimental setup

The drawing of the experimental setup for testing the pool boiling of working fluids is shown in Fig. 1.

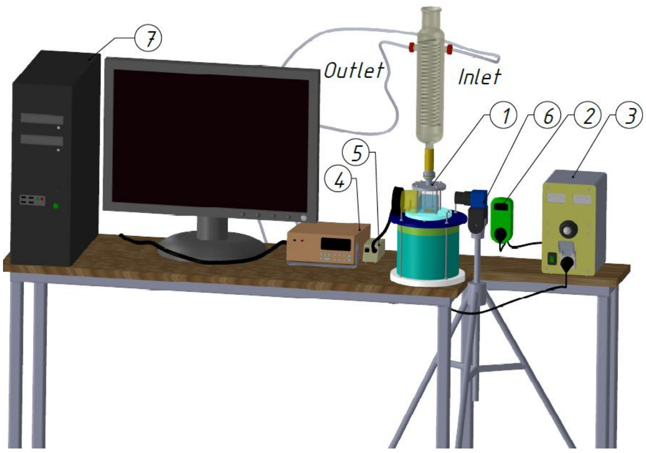

Fig. 1. Experimental setup, 1 – main module, 2 – wattmeter, 3 – autotransformer, 4 – data logger, 5 – light, 6 – high speed camera, 7 – PC.

The autotransformer with a wattmeter allows determining the power supplied to the cartridge heater placed in the copper cylinder of the main module (Fig. 2.) Two sheathed K-type thermocouples (NiCr-NiAl) are placed in boiling liquid above the specimen (T1, T2). They allowed the saturation temperature of the fluid to be checked. The stability of temperature during the experiment indicated that the saturation temperature was maintained. The condenser allowed the fluid to return to the glass vessel, which made the main module work as a thermosiphon. Four K-type thermocouples (T5 – T8) were placed at specified distances in the heating cylinder and two in the grooves under the specimen (T3, T4).

* Corresponding author: tmprp@tu.kielce.pl

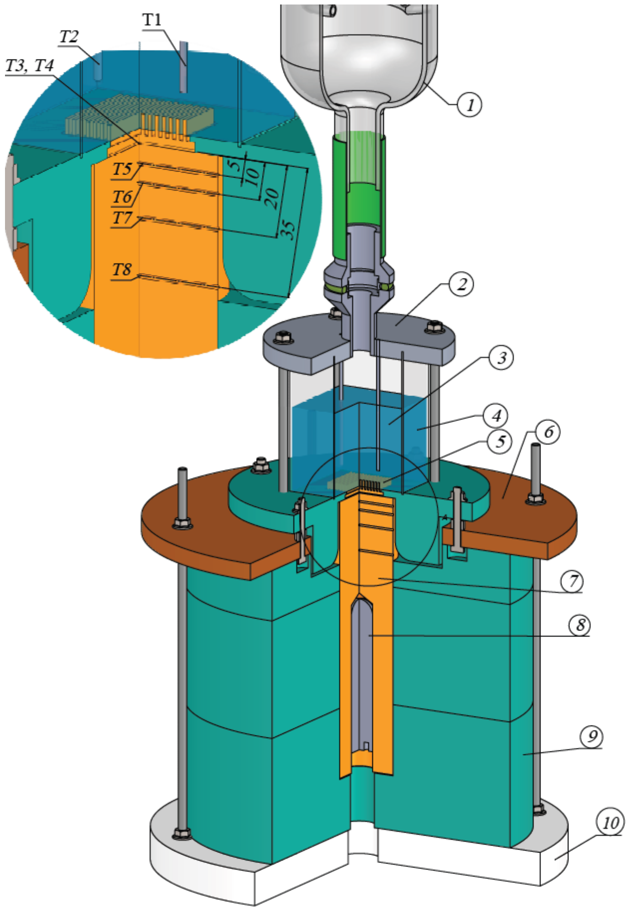

Fig. 2. Visualization of the main module of the test stand; 1 – condenser, 2 – top flange, 3 – boiling liquid, 4 – glass vessel, 5 – specimen, 6 – annular flange, 7 – copper cylinder, 8 – cartage heater, 9 – insulation, 10 – bottom flange. Dimensions in millimeters.

They allowed the determination of a one-dimensional heat flow in the copper cylinder. The heat flux was calculated according to the following formula:

$$q = \frac{\lambda_{Cu} \cdot \pi d^2}{4A_{bs}} \cdot \frac{T_8 - T_5}{\delta_8 - \delta_5} \quad (1)$$

The extrapolated superheat, defined as the temperature difference between the reference surface and the saturated liquid, was as follows:

$$\Delta T = \frac{T_3 + T_4}{2} - q\left(\frac{\delta_{bs}}{\lambda_{Cu}}\right) - \frac{T_1 + T_2}{2} \quad (2)$$

According to Newton's law of cooling, for one- dimensional heat conduction, the heat transfer coefficient was defined as:

$$\alpha = \frac{q}{\Delta T} \quad (3)$$

The MC sample contained pieces of copper mesh or foam in various configurations. The dimensions of specimen are shown in Fig. 3, and an example of the arrangement of porous structures is shown in Fig. 4.

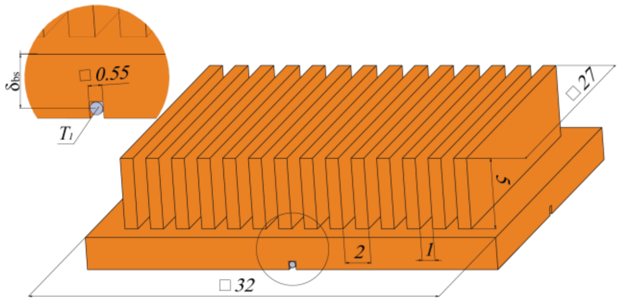

Fig. 3. MC specimen with dimensions in millimetres.

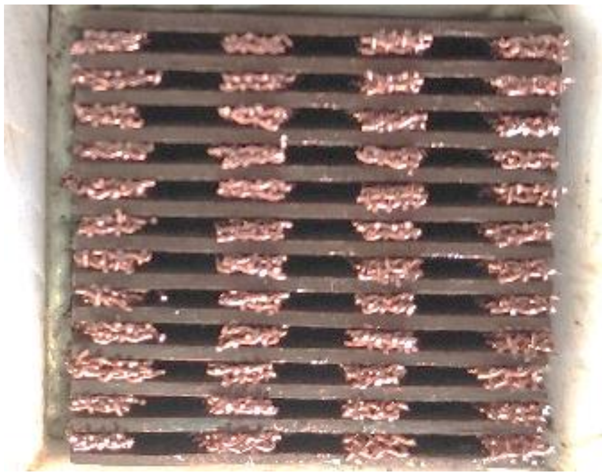

Fig. 4. Example of the wire mesh arrangement for the MCC-M-7.6-0.5-2 specimen.

The foam and mesh fillings had the following variable parameters: pitch (p_p), thickness of a single layer (δ_p), number of meshes placed side by side in the channel (n_p), length of the filling (l_p) and mesh aperture (a_p). Fig. 5 shows the surface MCC-M-3.5-0.5-2 with markings described in Table 1.

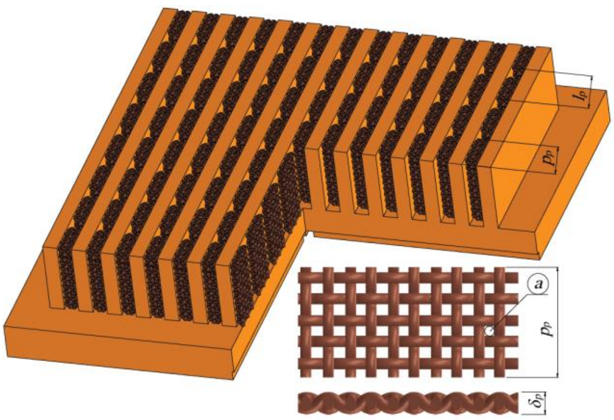

Fig. 5. 3D model of specimen MCC-M-3.5-0.5-2.

Specimens having a porous structure inside the minichannels are marked with M or F (mesh or foam). For specimens with porous material pieces in the minichannels, the code consists of three variables from Table 1: pitch, thickness of a single layer, number of meshes placed side by side in the channel.

Table 1. Specimen code and specification.

Sample code	p_p, mm	δ_p, mm	n_p	l_p, mm	a_p, mm
MC	-	-	-	-	-
MCC-M	-	1	1	27	0.7
MCC-F	-	1	1	27	0.25
MCC-M-3-0.5-2	3	0.5	2	2	0.3
MCC-M-3.5-0.5-2	3.5	0.5	2	2.5	0.3
MCC-M-7.6-0.5-2	7.6	0.5	2	3.8	0.5
MCC-F-3.6-0.3-3	3.6	0.3	3	1.5	0.25
MCC-F-3.5-0.5-2	3.5	0.5	2	2	0.25
MCC-F-4.2-0.5-2	4.2	0.5	2	2	0.25

During the tests, the linearity of the temperature values was checked in all specimen. The coefficient $R^2 \approx 0.99$ indicated one-dimensional thermal conductivity (Fig. 6).

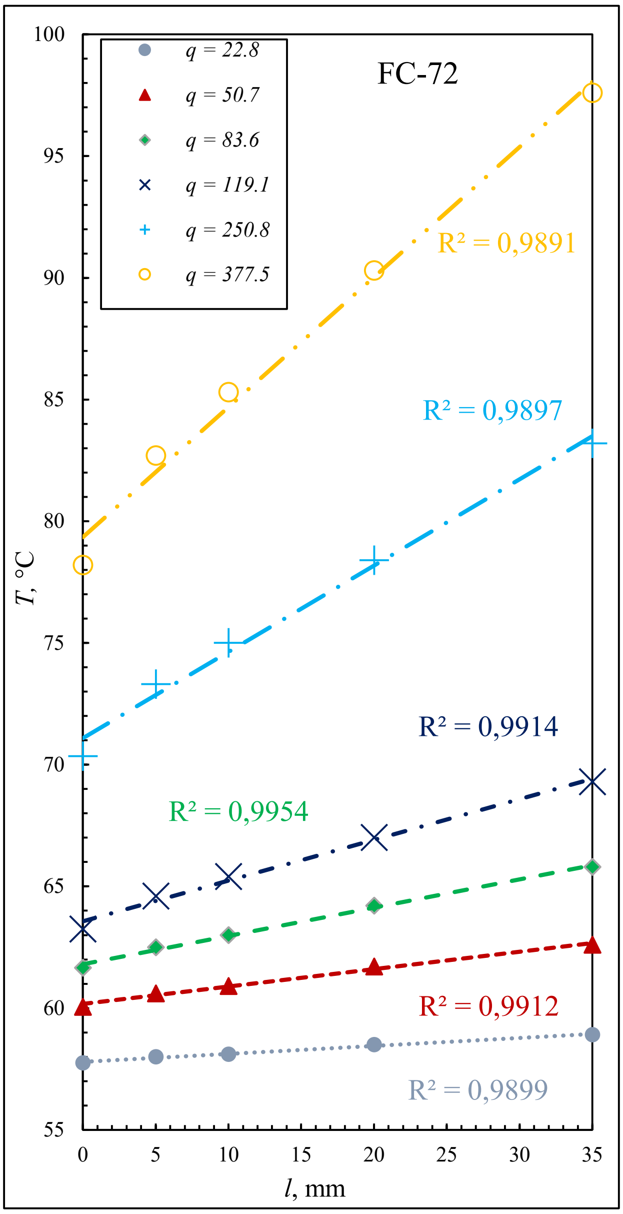

Fig. 6. Temperature vs. thermocouple location in the heating cylinder, MCC-F-4.2-0.5-2, boiling FC-72.

3 Results

The surface with open minichannels (MC) significantly improved the heat transfer coefficient compared to the plain surface (Figs. 7 and 8). The highest heat transfer coefficient for MC was about 25 kW/(m²K) – it was five times better than for the plain surface. Additional porous fillings improved HTC at a heat flux less than 220 kW/m²; the best results were obtained at 170 kW/m² for specimen MC: HTC was about 20 kW/(m²K) for MCC-M-3.5-0.5-2 and about 23 kW/(m²K) for MCC-M-3-0.5-2. For this range of heat flux, the additional mesh structure improved HTC by 15% compared to open minichannels. Minichannels with foam fillings had better performance than empty minichannels (about 15% for heat flux near 110 kW/m²). The superheat obtained was two to three times lower than for a plain smooth surface for heat fluxes smaller than 50 kW/m². Comparing the specimens with the additional filling to MC, no significant differences in superheat were observed for heat fluxes less than 120 kW/m². From the lowest values of heat flux to about 110 kW/m², the superheat values were comparable. When this value was exceeded, the MC specimen had lower superheat, but specimens with additional filling were able to withstand higher heat flux. For MCC-F-3.5-0.5-2, the heat flux obtained was about 400 kW/m² before the boiling crisis occurred. It was 25% higher than for the MC surface.

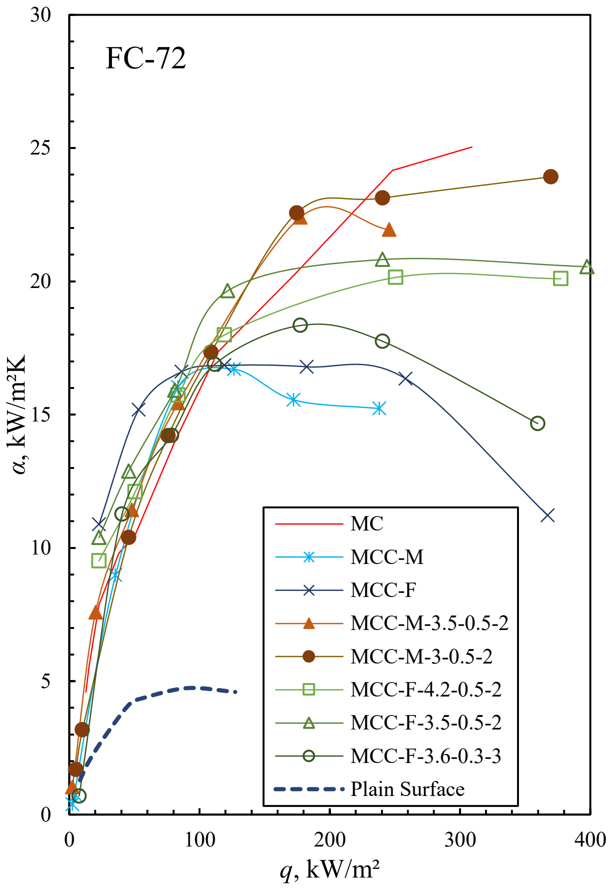

Fig. 7. Boiling curves for FC-72, heat transfer coefficient vs. heat flux.

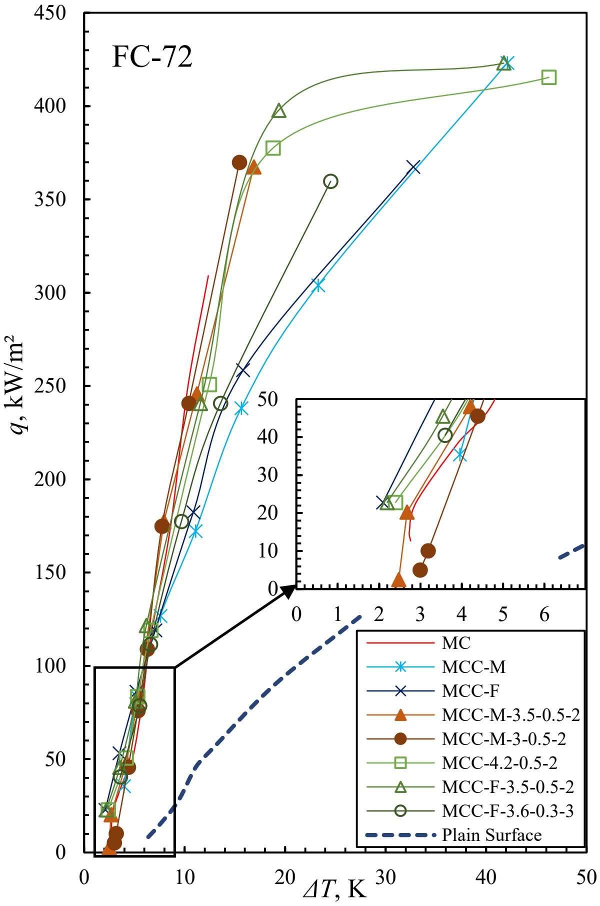

Fig. 8. Boiling curves for FC-72, heat flux vs superheat.

4 Conclusions

Based on the research carried out, the following conclusions can be drawn:

- The use of the surface with open minichannels significantly increased HTC and critical heat flux during pool boiling, compared to the plain surface. This occurs because surfaces with minichannels enable an increase in the number of nucleation centres and effective vapour removal. Most bubbles form between fins, but such formations are random.
- Additional porous structures in the spaces between the fins allow systematizing both bubble formation and liquid inflow. Most bubbles nucleated, grew, and detached in the channels created by porous filling. The porous filling acted as a capillary structure that improves suction of the working fluid into the minichannel space.
- Additional porous structures delayed the boiling crisis even at higher superheats. Stabilizing the flow of the working fluid and a larger area of extended surface increased the HTC.
- Minichannels fully filled with porous material demonstrated worse performance than open minichannels. This is probably due to the smaller empty spaces between the additional fillings. The flow of the liquid and the formation and growth of the bubbles occur in this case in a similar manner as on surfaces with capillary porous structures.

The tests were carried out with only one working fluid and one size of the specimens with minichannels. Sometimes, even small changes can significantly improve the heat transfer coefficient or superheat. Further tests including other working fluids and channel widths will be performed in the future.

Nomenclature

A – area, mm^2
a – aperture diameter, mm
d – copper cylinder diameter, mm
l – length, mm,
MC – minichannel,
n – number of mesh layers,
p – pitch, mm,
T – temperature, K,
q – heat flux, kW/m^2,

Greek symbols

ΔT – difference of temperature, K,
λ – thermal conductivity, W/(mK)
δ – thickness, mm,
α – heat transfer coefficient, W/(m^{2}K),

Subscripts

p – porous
Cu – copper
bs – base of sample
1–8 – numbers of thermocouples

Reference

1. K.C. Leong, J.Y. Ho, K.K. Wong, Appl. Therm. Eng. **112** (2017),
2. L. L. Manetti, A. S. O. Henriques Moita, R. R. de Souza, E. M. Cardoso, Int. J. Heat Mass Transf. **152** (2020),
3. A.K. Das, P.K. Das, P. Saha, Appl. Therm. Eng. **29** (2009),
4. C. M. Patil, S. G. Kandlikar, Heat Transf. Eng. **35** (2014),
5. D. Eok Kim, D. In Yu, D. W. Jerng, M. Hwan Kim, H. Seon Ahn, Exp. Therm. Fluid Sci. **66** (2015)