EPJ Web of Conferences **143**, 02077 (2017) DOI: 10.1051/epjconf/201714302077
EFM 2016

A study of the flow boiling heat transfer in an annular heat exchanger with a mini gap

Tomasz Musiał[1,a], Magdalena Piasecka[1] and Sylwia Hożejowska[2]

[1]*Kielce University of Technology, Faculty of Mechatronics and Mechanical Engineering, Al. 1000-lecia PP 7, 25-314 Kielce, Poland*
[2]*Kielce University of Technology, Faculty of Management and Computer Modelling, Al. 1000-lecia PP 7, 25-314 Kielce, Poland*

Abstract. In this paper the research on flow boiling heat transfer in an annular mini gap was discussed. A one-dimensional mathematical approach was proposed to describe stationary heat transfer in the gap. The mini gap 1 mm wide was created between a metal pipe with enhanced exterior surface and an external tempered glass pipe positioned along the same axis. The experimental test stand consists of several systems: the test loop in which distilled water circulates, the data and image acquisition system and the supply and control system. Known temperature distributions of the metal pipe with enhanced surface and of the working fluid helped to determine, from the Robin boundary condition, the local heat transfer coefficients at the fluid - heated surface contact. In the proposed mathematical model it is assumed that the cylindrical wall is a planar multilayer wall. The numerical results are presented on a chart as function of the heat transfer coefficient along the length of the mini gap.

1 Introduction

In recent years, many works have been published to address the issues of the heat transfer process in flow boiling in mini gaps. The major advantage of the heat exchangers with mini gap technology is improving the effectiveness of the heat transfer processes. Effective heat transfer is very important for the cooling processes in miniature equipment, as it prevents overheating and permanent damage to components. The cooling technology that uses flow boiling processes in mini channels will contribute to further development of current trends regarding the miniaturisation of products and devices.

In Ref. [1] authors showed the results of experimental investigation into heat transfer of helium-xenon mixtures in cylindrical channels. In Ref. [2], in order to determine the heat transfer coefficient, the researches performed experimental tests during nanofluid (based on water and copper oxide particles) flow in a cylindrical channel. Ref. [3] described determination of the heat transfer coefficient in a mini channel evaporator with R-134a as a refrigerant. In Ref. [4], authors presented the study of R134a flow boiling heat transfer and evaluation of existing correlations was reported. Flow boiling heat transfer of R134a in a multiport minichannel heat exchangers was presented in Ref. [5]. In Ref. [6], authors proposed an improved semi-empirical method for determining heat transfer coefficient in flow boiling in conventional and small diameter tubes. Papers [7] and [8], presented investigations into heat transfer bubbly boiling process, harnessing environmentally friendly refrigerating media in conventional channels. Research on heat transfer coefficients was presented in [9], where pressure drop and 'dryout' for flow boiling of water in an oil heated minichannel was discussed.

In authors' earlier research [10-13], flow boiling heat transfer in rectangular minichannels were studied while Fluorinert FC-72 was heated by an enhanced surface. Liquid crystal thermography and/or infrared thermography were applied to measure the temperature of the heated minichannels wall. Local values of the heat transfer coefficients were calculated numerically using one- and two-dimensional methods.

In the present paper, one-dimensional method is proposed to determine the heat transfer coefficient in flow boiling of distilled water in an annular mini gap.

2 Experimental setup and data

2.1 Experimental stand and procedure

The experimental stand is shown in Fig. 1. It is composed of several systems: the test loop with an annular mini gap in which the working fluid (distilled water) circulates, the data and image acquisition system, the supply and control system, and the lighting system. The test loop consist of: a heat exchanger, mass flow meter, a compensating tank, a gear pump, a deaerator and a filter. The data and image acquisition system comprises: a data acquisition station, a high speed camera, the lighting and a computer with special software.

[a] Corresponding author: tmusial@tu.kielce.pl

After deareation, there is a gradual increase in the electric power supplied to cartridge heater. Data for three settings of increasing heat flux, during laminar flow of distilled water along a mini gap, are analysed. The data used were obtained from the subcooled boiling region, in which the working fluid was superheated only at the interface with the heater and subcooled at the core of the flow.

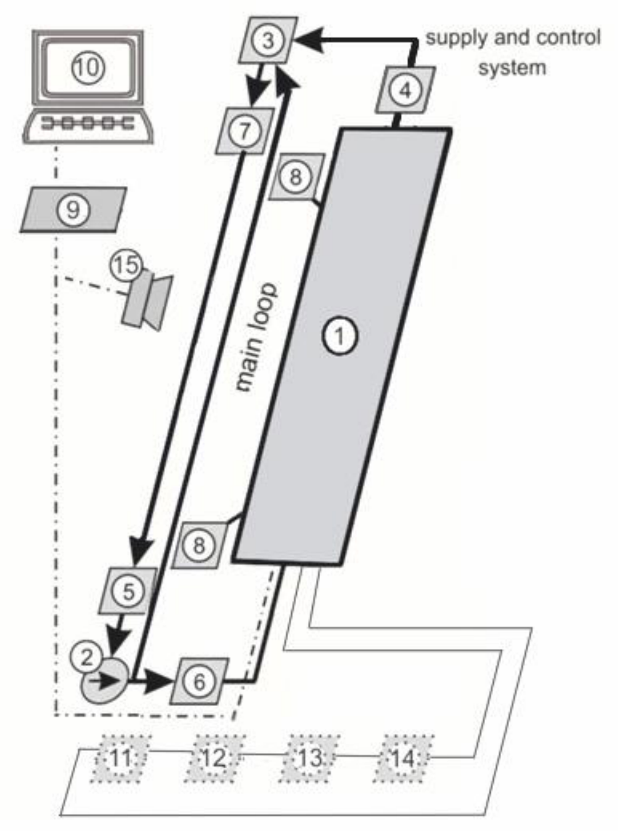

Fig. 1. The schematic diagram of the main loops at the experimental setup, 1-measurement module with a an annular mini gap; 2- gear pump; 3-compensating tank/pressure regulator; 4-tube-type heat exchanger, 5-filter, 6-mass flow meter; 7-deaerator; 8-pressure transducer; 9-data acquisition station; 10-pc computer; 11-inverter welder; 12-voltmeter; 13-ammeter; 14-shunt;15- high-speed camera.

2.2 The testing module with an annular mini gap

Figure 2 presents the testing module with an annular mini gap. The mini gap 1 mm wide was created between the metal pipe (2) and the glass pipe (1) positioned along the same axis. Inside the metal pipe (2), a cartridge heater (4) was located axially and symmetrically, powered by autotransformer with adjusted current intensity. Longitudinal slots was made in the inside surface of the pipe to fix the thermoelement sensors with a spacing of approximately 1 cm in the flow line. Thermoelement wires run in the gap between the heater and the metal pipe in the thermal conductive filler layer. The module header had a sealing (5) and the sensors of successive thermoelements and pressure transducers to measure the coolant temperature and positive pressure in the inlet and outlet to/from the mini gap.

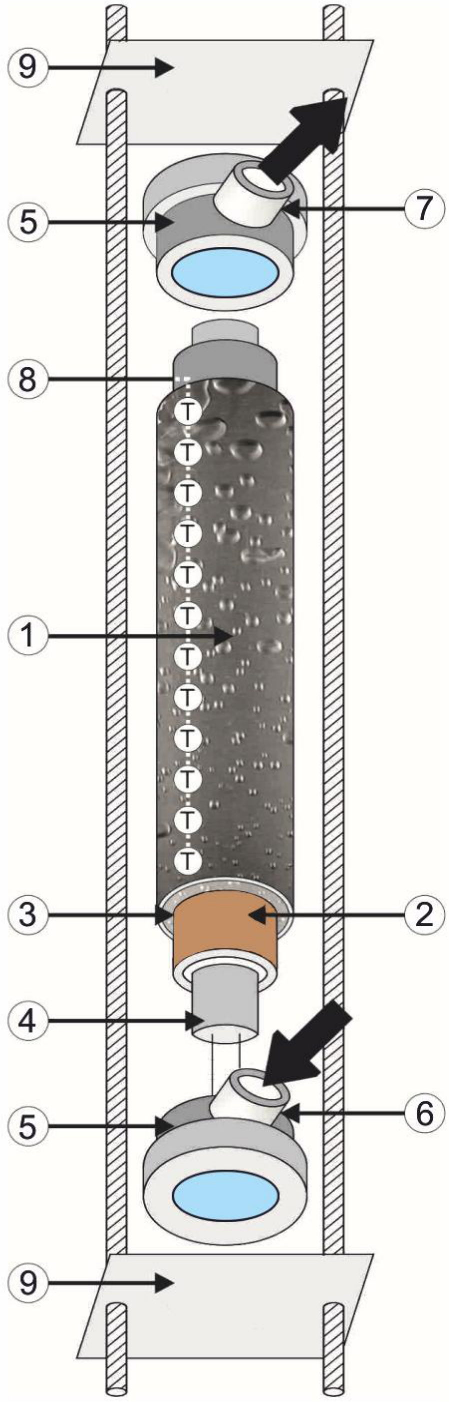

Fig. 2. The testing module with an annular mini gap layout: 1-glass pipe for viewing flow patterns, 2-metal pipe with enhanced surface, 3-annular mini gap, 4-cartridge heater, 5-module header with sealing, 6-inlet port for the medium, 7-outlet port for the medium, 8-thermoelements between the heater (4) and the metal pipe (2) in the thermal conductive filler layer, 9-stirrup bolts.

3 Heat transfer coefficient determination

It is assumed that the steady state heat transfer process in the testing module is stationary. Particular elements of the module create a system of planar layers with different thickness and thermal conductivity. Diagram illustrating the assumptions adopted for the planar multilayer wall is presented in Fig. 3.

The cartridge heater (with radius r_1) is the heat source with power output q_V.

For $r = r_1$ at each point $z_1, z_2, ..., z_M$, the condition below is satisfied

$$\tfrac{1}{2} r_1 q_V = \lambda_h \frac{\partial T_h(r_1, z_i)}{\partial r} = \lambda_s \frac{\partial T_s(r_1, z_i)}{\partial r} \tag{1}$$

Since difference $r_2 - r_1$ is very small, it is possible to replace the partial derivative with a finite difference:

$$\frac{\partial T_s(r_1, z_i)}{\partial r} \approx \frac{T_{s,i} - T_h(r_1, z_i)}{r_2 - r_1} \tag{2}$$

where $T_{s,i} = T_s(r_2, z_i)$ are the temperatures measured using the thermocouples at points (r_2, z_i), Fig. 3a.

Combining equations (1) and (2) we obtained formula for the temperature $T_h(r_1, z_i)$:

$$T_h(r_1, z_i) = T_{s,i} + q_V \frac{(r_2 - r_1) r_1}{2\lambda_s} \tag{3}$$

The relationship between the cartridge heater temperature and the metal surface temperature for the planar multilayer wall Fig. 3a can be written as:

$$T_h(r_1, z_i) - T_F(r_3, z_i) = \frac{q_V r_1}{2} \left(\frac{(r_2 - r_1)}{\lambda_s} + \frac{(r_3 - r_2)}{\lambda_F} \right) \tag{4}$$

From equations (3) and (4) we get

$$T_F(r_3, z_i) = T_{s,i} - \frac{q_V r_1}{2} \frac{(r_3 - r_2)}{\lambda_F} \tag{5}$$

From equation (5) and from the Robin boundary condition, we got the formula for the local heat transfer coefficients at points $z_1, z_2, ..., z_M$, at the interface between the heated surface and the fluid

$$\alpha(z_i) = \alpha_{1,i} = \frac{0.5 r_1 q_V}{T_{s,i} - 0.5 r_1 q_V \dfrac{(r_3 - r_2)}{\lambda_F} - T_f} \tag{6}$$

In equation (6) it was assumed that the fluid temperature T_f changes linearly in the mini gap, from the inlet temperature $T_{f,in}$ to the outlet temperature $T_{f,out}$. Both temperatures are known from the measurements.

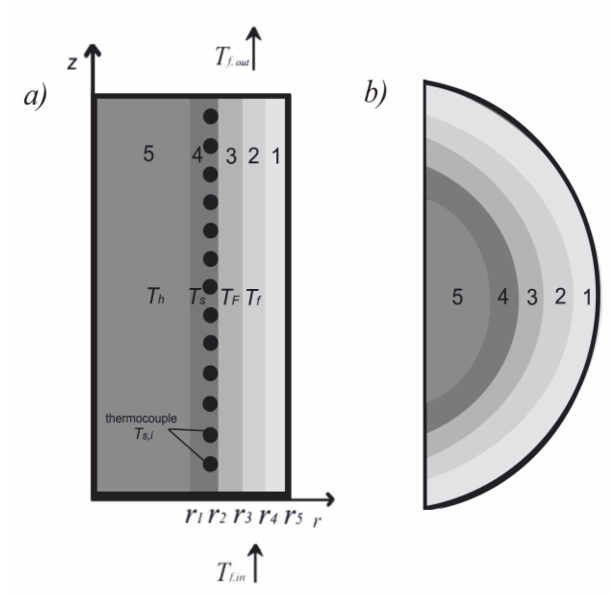

Fig. 3. Diagram illustrating the assumptions adopted for the planar multilayer wall: a) horizontal cross-section, b) vertical cross-section, 1- glass pipe, 2 - mini gap, 3- metal pipe, 4- thermal paste, 5- cartridge heater.

4 Results

Calculations were conducted for the subcooled boiling region at three settings of the volumetric heat flux, with all parameters of the experiment given in Fig. 4. The local values of heat transfer coefficient were found to decrease with the mini gap length. The increasing heat flux supplied to the heated surface caused heat transfer coefficients to decrease from above 4 kWm^{-2} K^{-1} to below 3 kWm^{-2} K^{-1} (for higher values of the volumetric heat flux). At the volumetric heat flux of 57.5 [MW/m^3], the heat transfer coefficient shows a more inclined plot along the mini gap length, takes the highest value at the beginning sections of the mini gap, with the lowest value of all three cases at the end section.

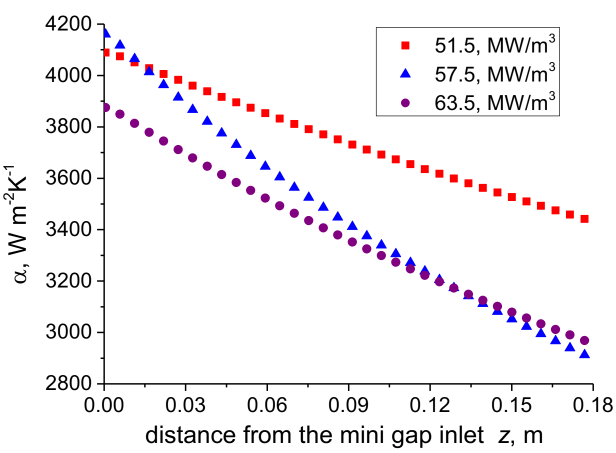

Fig. 4. Heat transfer coefficient vs. the mini gap length, experimental parameters (average): mass flow rate of 3.4 kg·s^{-1}, inlet pressure of 130 kPa, inlet liquid subcooling of 26 K, three settings of the volumetric heat flux are shown in figure.

5 Conclusions

The present paper discusses the flow boiling heat transfer research conducted in an annular w mm wide mini gap with distilled water as the working fluid. The studies were carried out for three settings of the increasing heat flux supplied to the heated surface, in the subcooled boiling region. The one-dimensional mathematical method was proposed to describe stationary heat transfer in the gap. Known temperature distributions in the heated surface and in the working fluid helped to determine, from the Robin boundary condition, the local heat transfer coefficients at the fluid - heated surface contact. In the proposed mathematical approach, the cylindrical wall is assumed to be a planar multilayer wall. The results were presented on the chart as a function of the heat transfer coefficient along the mini gap length. Observations indicated that the heat transfer coefficient decreased with the distance from the gap inlet.

In their further research, the authors intended to extend the results obtained over the saturated boiling region and focus on numerical modelling of heat transfer with the aid of the CFD module.

Acknowledgments

The research has been financially supported by the National Scientific Center from funds granted by virtue of decision No. DEC-2013/09/B/ST8/02825.

References

1. S. L. Elistratov, O. V.Vitovskii, E. Y. Slesareva, *J. Eng. Thermophys.* **24,** 33–35 (2015).
2. A. V. Minakov, V. Y. Rudyak, D. V.Guzei, & A. S. Lobasov, *High Temp.* **53,** 246–253 (2015).
3. E. B. Dollera, E. P. Villanueva, *IOP Conf. Ser. Mater. Sci. Eng.* **88,** 12027 (2015).
4. Y. Xu, X. Fang, G. Li, D. Li, Y. Yuan, *Int. J. Heat Mass Transf.* **92,** 1143–1157 (2016).
5. J. Kaew-On, K. Sakamatapan, S. Wongwises, *Exp. Therm. Fluid Sci.* **35,** 364–374 (2011).
6. D. Mikielewicz, J. Mikielewicz, J. Tesmar, *Int. J. Heat Mass Transf.* **50,** 3949–3956 (2007).
7. T. Bohdal, *Annu. Set Environ. Prot.* **15,** 107–126 (2013).
8. T. Bohdal, *Int. J. Heat Fluid Flow* **21,** 449–455 (2000).
9. L. A. Campbell, S. G. Kandlikar, *Proc. 2nd Int. Conf. Microchannels Minichannels* (2004).
10. M. Piasecka, *Int. J. Refrig.* **56,** 198–212 (2015).
11. M. Piasecka, *Int. J. Heat Mass Transf.* **81,** 114–121 (2015).
12. S. Hożejowska, M. Piasecka, *Heat Mass Transf.* **50,** 1053–1063 (2014).
13. M. Piasecka, K. Strąk, B. Maciejewska, *Heat Transf. Eng.* **38** (3)**,** DOI 10.1007/s00231-016-1895-1 (2017).