International Communications in Heat and Mass Transfer 89 (2017) 47–56

Contents lists available at ScienceDirect

International Communications in Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ichmt

Experimental investigation of single-phase turbulent flow of R-134a in a multiport microchannel heat sink

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ARTICLE INFO

Keywords:
Single phase flow
Heat transfer coefficient
Microchannel
Refrigerant

ABSTRACT

This experimental study aims to investigate the heat transfer characteristics of single-phase turbulent flow of R-134a refrigerant in a rectangular multi-micro channel heat sink having 27 channels where each channel has a hydraulic diameter of 421 μm. Experimental results were obtained for inlet temperatures ranging from 24 to 33 °C, mass fluxes from 1485 to 2784 $kg\,m^{-2}\,s^{-1}$ and wall heat fluxes from 3 to 24 $kW\,m^{-2}$. The results indicate that the heat transfer coefficients are found to be higher at lower inlet temperatures than those at higher ones. In addition, when equal amount of heat supplied to the heat sink, the heat transfer coefficients increase with increasing the mass flux of refrigerant. They were also compared with 12 well-known correlations and it was seen that 4 of 12 were in good agreement with each other with the average deviation < 10%. The findings demonstrate that well-known correlations in fundamental sources can be used to predict the heat transfer coefficient of R-134a during its single phase flow in a multiport microchannel heat sink under turbulent regime.

1. Introduction

Reducing the size of thermal systems has been a main subject of research during last decade. Nowadays, with the rapid development of manufacturing technology, a growing interest in designing of various types of microchannel heat sinks has emerged because of their wide applications in cooling of electronic devices. Microchannels with relatively small sizes can transfer high amounts of heat flux which promises an efficient cooling approach especially in the case of electronic devices. Here, a brief review is done on the fluid flow in multiport microchannel.

Wu and Little [1] studied the nitrogen flow in rectangular channels having hydraulic diameter between 134–164 mm. This work was one of the first studies on heat transfer in microscale. They found that classic models fail to predict the Nusselt numbers, and the real value of Nusselt number is higher than that of predicted one. They presented a correlation derived from their experimental results.

Wang and Peng [2] experimentally investigated the single-phase forced convective heat transfer of water and methanol in microchannels with rectangular cross-section. They found that the classic correlation of Dittus-Boelter can be used to predict the Nusselt number for turbulent flow if the constant coefficient in the correlation changes from 0.023 to 0.00805. Later, Peng et al. [3] examined the single-phase forced convective heat transfer and flow behaviours in microchannels having hydraulic diameters of 0.133–0.367 mm with dissimilar geometric configurations with their previous work [2]. Here, they discovered a new dimensionless variable (Z) for the turbulent heat transfer calculations; when $z = 0.5$, it was proved to be the optimum value for turbulent heat transfer irrespective of the aspect ratio.

Adams et al. [4] found that classic correlations which have been developed for flow in conventional channels are not useful to predict the Nusselt number if microchannels. They presented new correlations based on their experiments for water flow in circular channels with diameters of 0.76 and 1.09 mm. Later, Adams et al. [5] evaluated the single-phase turbulent flow of in a non-circular microchannel with a hydraulic diameter of 1.13 mm. They indicated that Gnielinski correlation is a reliable relation to estimate the value of Nusselt number in such channels.

Harms et al. [6] examined the water flow in microchannels for both laminar and turbulent regimes (Reynolds numbers among 173 to

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http://dx.doi.org/10.1016/j.icheatmasstransfer.2017.09.023

12,900). They found that classic correlations can predict the heat transfer rate in the microchannel well where the flow is laminar.

Agostini et al. [7] studied the single-phase upward flow of R-134a refrigerant in a flat aluminium multiport heat sink with 11 parallel rectangular channels (3.28 mm × 1.47 mm) where the hydraulic diameter of each channel was 2.01 mm. They found that heat transfer and pressure drop characteristics of minichannels can be predicted well with correlations developed for conventional tubes, especially for Reynolds numbers between 500 and 7000.

Owhaib and Palm [8] analysed the single-phase flow of R134a in three single microchannels having inner diameters of 1.7, 1.2, and 0.8 mm. They compared their collected results with well-known correlations for laminar and turbulent region. It was seen that the measured Nusselt numbers overlapped with classical correlations for laminar flow, but the same situation cannot be valid for turbulent flow.

Lee et al. [9] studied the thermal behaviour of single-phase flow in rectangular microchannels with different sizes (with hydraulic diameters of 318–903 mm) using water as the working fluid. Heat flux was 45 W cm^{-2} and Reynolds number varied from 300 to 3500. The experimental results were very close to results obtained by classical models (an average deviation of 5%).

Caney et al. [10] conducted a similar study with Lee et al. [9] using aluminium minichannels. The heat flux supplied to refrigerant was between 1 and 8 kW m^{-2}. They concluded that temperature profile can be found by classical correlations, but some corrections are necessary for non-uniform heat flux conditions.

Celata et al. [11] investigated single-phase laminar flow in circular microtubes with hydraulic diameters ranging from 120 to 528 μm. The experimental findings demonstrated a decrease of Nusselt number with decreasing hydraulic diameter, an axial dependence associated with thermal entrance effects and an addiction of the Nusselt number also on Reynolds number. Additionally, for the smallest channel examined (50 μm ID), the sensitivity to measurement errors is so high that it is impossible to make a realistic predict about the heat transfer coefficient. In the next step, Celata et al. [12] queried single-phase heat transfer for water flow in single stainless steel microtubes with diameters varying from 0.5 mm to 0.12 mm. In the case of laminar, they saw that reducing the Reynolds number in smooth glass tubes causes an abnormal decrease in the Nusselt number. It was also seen that while stainless steel tubes exhibit relatively normal diabetic behaviour in this regime, the unexpected decline in Nusselt number is due to peripheral adhesions in the test section with heat dissipation externally attached via thin film deposited on the outer surface of the glass tube.

Qi et al. [13] studied single-phase pressure drop and heat transfer characteristics of liquid nitrogen in four micro-tubes with diameters of 1.931, 1.042, 0.834 and 0.531 mm. It was found that the local heat transfer coefficient of the liquid nitrogen through the micro-tube was 12.5% lower than that of water. Furthermore, Gnielinski correlation was modified by adding the effect of surface roughness on the heat transfer along the microchannel, which provided them to compute the Nusselt numbers with a mean absolute error (MAE) of 6.4%.

Dai et al. [14] investigated the friction and heat transfer characteristics of single-phase ethanol in two multi-port extruded (MPE) microtubes with a hydraulic diameter of 0.715 mm (rectangular). The inlet temperature and heat flux values were between 5 and 45 °C, and 3 and 9 kW m^{-2}, respectively. It was found that Nusselt numbers increase with the decrease of inlet temperature and heat flux in the study.

Zhang et al. [15] carried out an experimental study on the flow and heat transfer characteristic of six MMFT's having hydraulic diameter changing from 0.48 to 0.84 mm and aspect ratio ranging from 0.45 to 0.88. The test results were obtained for Reynolds number ranging from 120 to 3750. The effects of aspect ratio, roughness and entrance effect on heat transfer were analysed in this experiment. It was found that aspect ratio has no a significant effect on heat transfer.

Kim [16] executed this experiment to detect the flow resistance and thermal behaviour of laminar flow in 10 different rectangular microchannels having hydraulic diameters of 155–580 μm and aspect ratios of 0.25–3.8. FC770 and water were used as refrigerant in the experimental setup. The study showed that for $Re < 180$, the experimental Nu is obviously lower than the values obtained from theoretical correlations and increases with increasing Reynolds number. However, for $Re > 180$, experimental Nu are apparently higher than that of obtained by theory.

Readers who want to have more information can look at some other articles about single-phase flow heat transfer in a multi-microchannel. Refs. [17–25], they are not presented here to save the space.

Previous researches on this subject clearly show that many scientists investigated the heat transfer characteristics in a rectangular multi-micro channel heat sink. It was also seen that the heat transfer characteristics of single-phase in turbulent regime throughout rectangular multi-micro channel heat sink having refrigerant as working fluid has been a subject comparatively much less discussed in literature. In this regard, this study aims to investigate the heat transfer characteristics of single-phase R-134a in turbulent regime along a rectangular multi-micro channel heat sink having hydraulic diameter of 0.421 μm. The experimental data presented in this study are new and have not been reported in the open literature. The objective of this study was to verify whether heat transfer characteristics of single-phase flow of R-134a in a multi-micro channel can be predicted by commonly used fundamental correlations under the conditions applied. The operating test conditions have been considered to extend the existing ones' range in the field of multiport micro channel heat sink research.

2. Experimental apparatus and procedure

2.1. Experimental apparatus

The schematic diagram and photographs of experimental set-up and test section have been presented in Figs. 1–3. As shown in Fig. 1, there are 2 different loops in the experimental apparatus. The first loop is to regulate the inlet temperature of working fluid before entering the test section. This loop composed of a centrifugal pump, a variable area flow meter, and a tank containing an RTD, serpentine coil, and a cartridge heater. The second loop is designed for circulating the refrigerant. A gear pump has been used to deliver the working fluid from the tank to the test section after passing a filter/dryer which prevents any clogging in the test section. To be sure that the inlet and outlet flows are single-phase, a sight glass has been mounted throughout the test section. The working fluid absorbs the heat from the microchannel that is heated via cartridge heaters. When the refrigerant exits from the test section, it enters a plate heat exchanger for cooling and completing the loop. The experimental set-up in the presented work is a modified version of set-up used in previous studies [35–37].

The test section is a heat sink having 27 microchannels with the dimensions given in Table 1, see Ref. [37] for more details of the test section.

The test section is heated using twelve cartridge heater (200 W each) which is embedded vertically in the bottom of the heat sink and parallel to the channel length. The amount of power has been controlled by a manually adjustable DC power supply. The values of voltage and electrical current are measured by a Clamp-On power meter with an uncertainty of ± 2% for voltage and ± 1% for current. While the temperature of the refrigerant was measured using T-type thermocouples with 0.5 mm diameter at both ends of the plenums, the temperature gradient and surface temperatures were determined using same type ten thermocouples which are located at five different locations 3 mm below the channel base with 9 mm equidistant intervals. In Fig. 2 (b), the location of the thermocouples on the test section has been illustrated. The accuracy of thermocouples was ± 0.1 °C. In order to minimize heat loss to the environment, the test section was insulated by polyurethane foam.

The uncertainty measurement tools and measured parameters