

Experimental Study on Air Side Heat Transfer and Fluid Flow Characteristics of Microchannel Heat Exchanger

2011-01-1166

Published
04/12/2011Engr Sarbadaman Dasgupta, Faisal A. Siddiqui and Amir Fartaj
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doi:10.4271/2011-01-1166

ABSTRACT

Microchannel heat exchangers are taken as a great area of research interest for their elevated heat transfer characteristics, higher heating surface area to volume ratio, smaller size, lighter weight, lower fluid inventory and energy efficiency compared to conventional heat exchangers with same heat exchange capacity. Though several investigations have been conducted to find out the air side heat transfer and flow characteristics of common compact cross-flow heat exchangers with various tube and fin geometries, the studies on air side thermal and hydraulic performances of microchannel heat exchangers are rare in open literature. In the present study, the air side heat transfer and fluid flow characteristics of a multiport microchannel heat exchanger in liquid-to-air crossflow orientation were investigated experimentally. Experimental setup is consisted of a closed-loop thermal wind tunnel with a test section of microchannel heat exchanger in crossflow orientation and numbers of measuring instruments with data acquisition system. The microchannel heat exchanger is made up of fifteen extruded aluminum slabs with 68 circular channels in each slab. Slab channels have the inner circular diameters set at 1mm, and the airside frontal area is 304mm × 304mm. Hot water and ethylene glycol 50:50 mixture at a constant temperature of 74°C was cooled by the cold air at different temperatures (23°C, 33°C and 43°C) flowing through the wavy fins of microchannel heat exchanger. The mass flow rate and temperature of the liquid were maintained at a constant level by using a gear pump and an inline circulation heater respectively. The inlet air velocities through microchannel heat exchanger were varied at different temperature levels. The air side Reynolds number, Nusselt number, overall thermal resistance, pressure drop and other key parameters

were examined in the region of the air side Reynolds number 750-3165, at a constant liquid side Reynolds number of 200. The heat balance was observed less than 3% for all experimental runs, and the air side thermal resistance range was 74% to 83% of overall thermal resistance. The air side Nusselt number correlation with the Reynolds number was obtained. The air side Nusselt number was found higher in comparison with other studies.

KEYWORDS

MCH-Microchannel heat exchanger, Crossflow, Energy efficient, Pressure drop, Heat transfer, Experimental Test Bench

INTRODUCTION

Scientists and researchers have been looking at energy efficient compact heat exchangers for quite some time. Heat exchangers containing narrow flow passages such as microchannels have attracted much researcher due to its elevated heat transfer characteristics. Tuckerman and Pease (1981) introduced the concept of microchannel flow passage to remove the heat from the silicon integrated circuits [1]. Consequently numerous studies were performed to investigate the heat transfer and flow behaviors in microchannel flow passages [2,3,4,5,6,7,8,9,10,11]. Owhaib and Palm (2004) investigated experimentally single phase convective heat transfer in circular microchannels [12]. Steinke and Kandlikar (2004) reviewed the different likely techniques to augment heat transfer in microchannels [13]. Morini (2004) reviewed the different researcher's experimental results on single phase convective heat transfer in microchannels [14]. Hetsroni et al. (2005) compared the heat transfer in microchannels among experimental results

with theory and numerical results with different microchannel geometries [15]. Harries et al. (2000) designed a crossflow micro heat exchanger to expedite the heat transfer from liquid to gas which could be the utilizable in a wide range of applications especially in the areas of automotive, home heating and aerospace [16]. Kandlikar et al. (2006) categorized the flow passages as well as provided more information on multiphase fluid flow and heat transfer characteristics in microchannels [17]. Khan and Fartaj (2010) investigated experimentally the fluid flow behaviors in multiport circular straight microchannel slab in the classical laminar flow regime [18]. The main advantages of microchannel heat exchangers compared to conventional compact heat exchangers with same heat transfer duty are: 1) smaller size, 2) lighter weight, 3) larger heat transfer area density (m^2/m^3), 4) lower fluid quantity, 5) higher heat transfer coefficient and 6) energy efficiency. All these features are lucrative for designing the heat exchangers in engineering applications such as automotive radiators, heater cores, AC condensers and evaporators. Microchannel heat exchangers will be the potential candidate for the future automotive heat exchangers which will save the size, weight, cost and energy significantly.

Air has the low thermal conductivity compared to most liquids. Low thermal conductivity of air leads to low heat transfer rates per unit surface area [19]. Thus the thermal resistance of the air side is dominating the overall thermal resistance of the heat exchanger. Commonly over 90% of overall thermal resistances exist in the air side for typical air cooled heat exchangers [20]. Extension of heat transfer area by using fins of different geometries is very common in heat exchanger design to improve the efficiency. The air side hydraulic and thermal characteristics of microchannel heat exchanger are rare in open literature. However, many researchers investigated experimentally and numerically air side flow and heat transfer characteristics of conventional compact heat exchangers such as fin-and-tube heat exchangers with different tube and fin geometry [21,22,23,24,25,26,27,28,29]. Park and Jacobi (2009) proposed j and f factor correlations for flat-tube louver-fin heat exchangers to predict the air side performance by analyzing experimental data of previous works. Their proposed correlations provide more accurate predictions compared to prior correlations those suggested for similar kind of heat exchanger (30). Wang et al. (2000) presented the data reduction method for analyzing the air side performance of fin-and-tube heat exchangers [31]. Junqi et al. (2007) studied experimentally the air side heat transfer and flow characteristics of wavy fin and flat tube air-water cross-flow heat exchangers with different fin pitches, fin lengths and fin heights in the region of the air side Reynolds number 800 to 6500, at a constant water flow rate. They obtained the general correlations for j and f factors [21]. Paeng et al. (2009) investigated experimentally and numerically the air side

forced convective heat transfer coefficient of plate fin and tube heat exchanger with the air side Reynolds number range from 1082 to 1649. They derived correlation equations for Nusselt number and compared it with some available correlations [24]. Tang and Yang (2005) studied experimentally the thermal performance of single-row fin-and-tube heat exchanger. They found that the dominant thermal resistance existed in the air side [27]. Taler (2005) predicted numerically the heat transfer correlations for cross-flow compact heat exchangers. He computed air-side heat transfer coefficient from overall thermal resistance equation. Both Gnielinski and Dittus-Boelter correlations were used to calculate the tube-side heat transfer coefficient [29]. Khan et al. (2004) experimentally investigated the air side heat transfer and fluid flow characteristics of cross-flow cooling of air via an in-line elliptical tube array. They stated the Nusselt number and Reynolds number correlation based on their experiment [32].

While a number of studies have been conducted to understand the heat transfer and flow behaviors in microchannels, the literature examining the air side heat transfer and flow characteristics of microchannel heat exchangers are scarce. The current study investigated experimentally the air-side heat transfer and flow characteristics of liquid-to-air cross-flow microchannel heat exchanger. The width of each multi-port microchannel slab is 100mm along the air flow path with flat surfaces on the top and the bottom. The flowing air blankets the flat heating surfaces with uniform temperature distribution. This uniform surface temperature is established due to flow of hot liquid through many narrow channels in the slab. Hot water and ethylene glycol mixture (50:50) was used as the working fluid in the liquid side. The hot liquid was forced to circulate through microchannels at a constant inlet temperature and a constant flow rate. Frontal air velocities were manipulated in five steps ranging from 3m/s to 11m/s and allowed to pass thorough the rectangular plain wavy fin in the core of microchannel heat exchanger to cool the hot liquid maintaining the specified inlet air temperature level. Three temperature levels were selected as 23°C, 33°C, and 43°C. Air side heat transfer coefficient (h_a), Nusselt number (Nu_a), effectiveness (ϵ), NTU, overall thermal resistance (R_{total}) were calculated using the experimental data. The correlation for Nusselt number with Reynolds number was obtained, and all of these are presented in this paper.

EXPERIMENTAL SETUP

The major components of the experimental setup (Fig. 1) for this study includes a closed loop thermal wind tunnel, test chamber, variable speed gear pump, circulation heater, data acquisition system composed of large number of measuring instruments and the serpentine finned microchannel heat exchanger. The wind tunnel is manufactured with a contraction ratio of 6.25 and can achieve air velocity of 30 m/

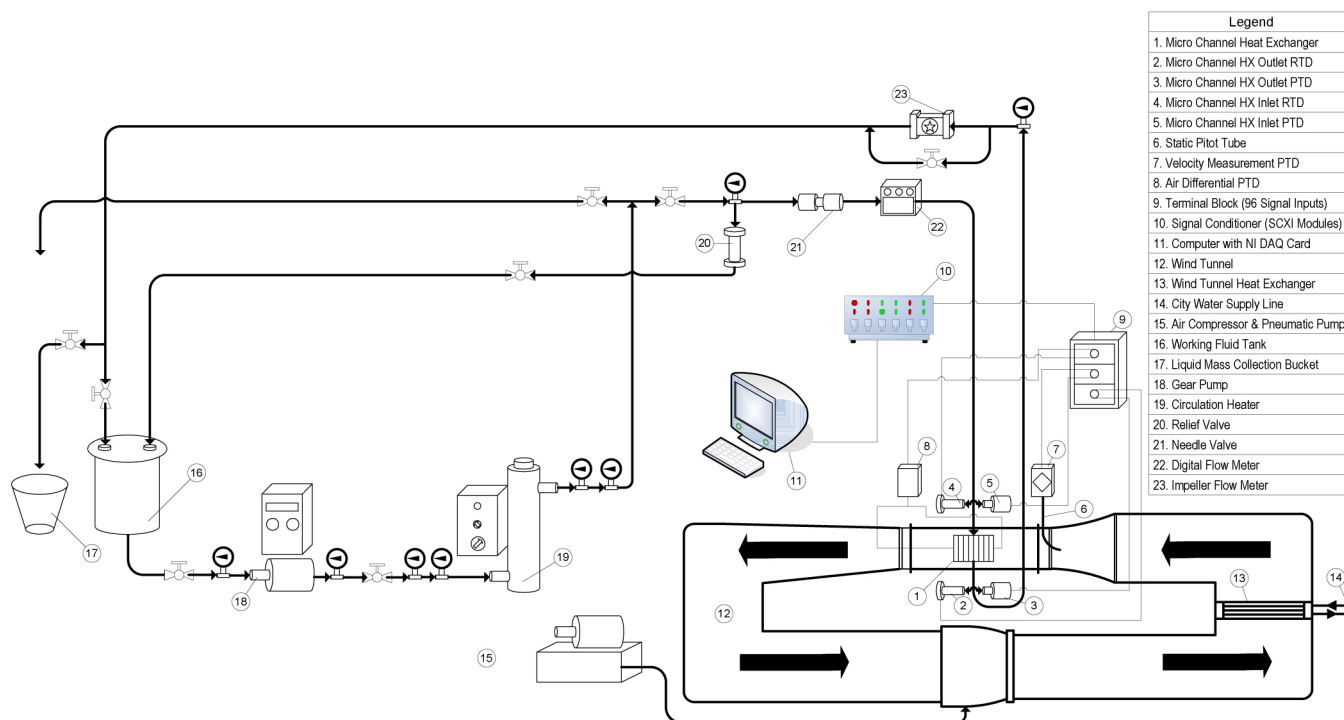


Figure 1. Schematic diagram of the experimental setup

s in absence of any obstruction. In presence of the serpentine microchannel heat exchanger, the top air velocity is just over 11 m/s. The test chamber is made of Plexiglas of low thermal conductivity ($k = 0.19 \text{ W / m.}^\circ\text{C}$) walls to minimize heat loss. The test chamber is essentially a straight duct capable of docking on the wind tunnel and housing the test heat exchangers. It has a length of 60 cm, height of 30 cm, and width of 30 cm.

The serpentine heat exchanger and its fins are made of aluminum, and its serpentine shape is achieved by carefully bending a 68 channel long slab into 5 small straight slab section of alternate flow direction. Each equally spaced channel within the slab is 1 mm in diameter and the slab is 2 mm thick. The heat exchanger is composed of three sets of such long serpentine slabs producing a total of 15 smaller straight alternate flow slab sections (Fig. 2). Fluid flows through the heat exchanger inlet pipe to the inlet header where flow is distributed to the three serpentine slabs. Fluid exits from these slabs to combine in exit header before flowing out through the exit tube.

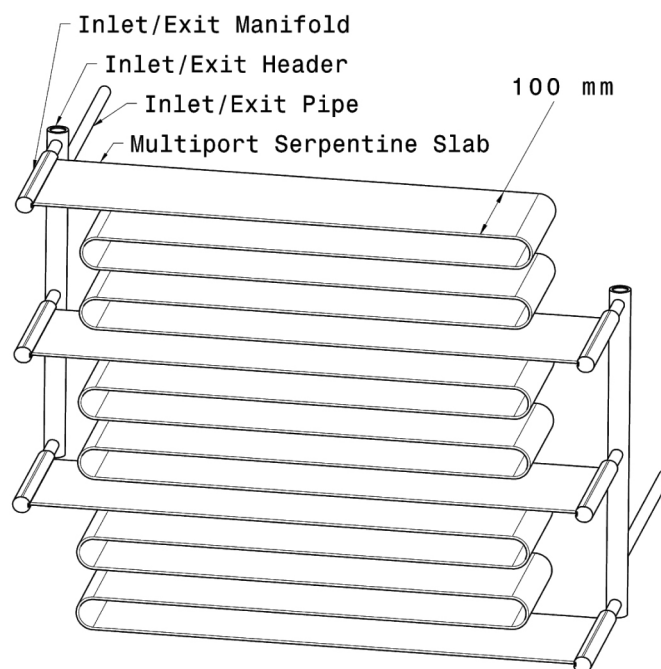


Figure 2. Microchannel slabs with inlet and outlet header

Each smaller slab sections are equally spaced within the heat exchanger and the space between them is filled with fins. The finned area length between slabs is 30 cm and fin density is 12 fin per 2.54 cm (12 fins per inch). The slab array is arranged such that the flow through the channels inside the

port is perpendicular to the air flow across the heat exchanger. Additional insulations were placed around the test chamber to prevent heat loss to the surrounding. Arrangements of rectangular wavy fins between two adjacent microchannel slabs are shown in [Figure 3](#).

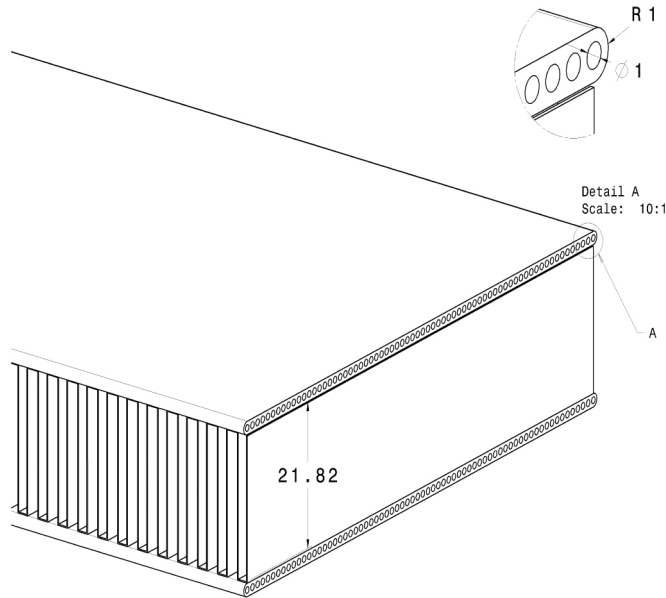


Figure 3. Microchannel slabs with fins

Hot 50% ethylene glycol-water mixture at 74 °C entered the heat exchanger through the inlet manifold and exited from the outlet manifold located at opposite ends of the heat exchanger. Cool air entered the test chamber at three different temperatures (23°C, 33°C, and 43°C). The air-side inlet and exit temperatures were measured using grids of 9 and 25 equally spaced and strategically placed thermocouples at the inlet and exit of the test chamber, respectively. Temperature measurements at the liquid side involved using Resistance Temperature Detectors (RTDs) at the inlet and outlet pipes of the heat exchanger. Thermocouples were placed at the bends of the heat exchanger slabs to measure the surface temperature at particular bends and also evaluate average surface temperatures. The air-side pressure drop across the heat exchanger was measured by connecting pressure taps located at side walls of test chamber. The liquid-side pressure drop is evaluated by measuring the inlet and exit pressures with Pressure Transducers (PTDs) located at the inlet and outlet pipes of the heat exchanger. All sensors readings such as the thermocouples, RTDs and PTDs were collected by data acquisition system.

The air flow circulation inside the thermal wind tunnel was allowed to stabilize over a period of approximately one hour prior to take any measurements. This ensured stable flow inlet velocity and temperatures. The liquid (glycol-water mixture) flow regime was preselected to be in laminar region with mass flow rate of 0.0345 kg/s which corresponds to a

Reynolds number of 200. The liquid flow rates were measured through digital flow meter. Liquid flow rates were also measured by collecting flow out of heat exchanger to a bucket and recording the collection times with a stop watch in addition to digital flow meter readings. The flow rate was averaged over 2 to 3 samples for each test run to attain a reliable value. For each of the three air-side temperature settings, the air flow rate was varied between five preselected values (3 m/s - 11 m/s), providing a range of Reynolds number between 750 to 3165. The air flow rate was evaluated by measuring air-side velocity using a pitot tube in conjunction with flow and pressure acquisition system.

DATA REDUCTION AND CALCULATION METHOD

The main reason of data reduction in this paper is to obtain the air side heat transfer characteristics such as overall thermal resistance ($R_{total} = 1/UA$), effectiveness (ϵ), NTU , Nusselt number (Nu_a) of the core section of microchannel heat exchanger from the experimental data. All data were collected under steady conditions while running the experiment. Thermodynamic properties of air and liquid (50% of water - ethylene glycol mixture) are calculated as per average temperature of inlet and outlet temperatures of air and liquid respectively.

HEAT TRANSFER RATE ($\dot{Q}$)

By knowing the inlet and outlet temperatures of hot liquid and cold air, the heat transfer rate $\dot{Q}$ can be calculated as an arithmetic average value of heat transfer rate of liquid, $\dot{Q}_l$ and heat transfer rate of air, $\dot{Q}_a$ from the following equation:

$$\dot{Q} = \frac{\dot{Q}_a + \dot{Q}_l}{2} \quad (1)$$

Where,

$$\dot{Q}_a = \dot{m}_a c_{p,a} (T_{a,o} - T_{a,i}) \quad (2)$$

$$\dot{Q}_l = \dot{m}_l c_{p,l} (T_{l,i} - T_{l,o}) \quad (3)$$

OVERALL THERMAL RESISTANCE (R_{total})

The heat transfer relation for multipass and cross-flow heat exchanger [33] as