ANALYSIS OF A RECTANGULAR MICROCHANNEL USING THE R-22 REFRIGERANT IN FORCED CONVECTION HEAT TRANSFER CONDITION

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Microscale heat exchangers are becoming an important area of interest in many fields of developing technology that require compact high heat energy removal solutions. In our work, numerical and experimental analyses are carried out for rectangular microchannels with five sets of rectangular configurations, in order to find optimum configuration of a microchannel. Analysis of a rectangular microchannel is carried out for the forced convection heat transfer condition with a constant base area of 30-mm length and 20-mm width. An experimental setup has been developed to test the microchannel with differential pressure transducers, J-type thermocouples, and a digital magnetic rotameter with microchannel manifold. A theoretical analysis was carried out with a C program code to find out the optimum theoretical dimension of the microchannel at various flow rates from 0.0016 kg/s to 0.1 kg/s. Experiments were carried out with heat inputs of 25 W to 150 W and flow inputs of 0.0016 kg/s to 0.1 kg/s. It was found that the heat transfer coefficient for a hydraulic diameter of 260 μm is equal to 12,000 W/m²·K. As compared to other configurations, the heat transfer coefficient is higher, so that the 260-μm diameter microchannel is more optimistic theoretically as compared to other configurations of a microchannel. Also, as the heat input increases, the configuration with a larger hydraulic diameter shows a less pressure drop as compared to a hydraulic diameter of 260 μm. The range of pressure drop for a microchannel is observed at 0.026 kPa. For a hydraulic diameter of 370 μm, the pressure drop is minimum, 0.01 kPa. Experimental results show that for a 30-mm-long microchannel with a hydraulic diameter of 260 μm, the temperature rise for the refrigerant R22 is in a range of 2°C to 18°C in the analysis of a single-phase refrigerant. The results for experimental temperature difference across the microchannel is 10 to 20% less as compared to theoretical results. The range of applicability allows a comparison of the refrigerant distribution in different designs of microchannel heat exchangers.

KEY WORDS: microchannel, heat transfer, pressure drop, heat transfer coefficient
NOMENCLATURE

\begin{tabular}{|l|l|}
\hline
\textbf{Quantity} & \textbf{Symbol} & \textbf{Definition} \\
\hline
height of a channel & \( a \) & \( \mu m \) \\
width of a channel & \( b \) & \( \mu m \) \\
hydraulic diameter & \( D_h \) & \( \mu m \) \\
Poiseuille number & \( f/Re \) & \\
heat transfer coefficient & \( h \) & \( W/m^2\cdot K \) \\
length of a channel & \( L_c \) & \( m \) \\
mass flow rate & \( m \) & \( kg/s \) \\
number of microchannels & \( n \) & \( \text{number} \) \\
Nusselt number & \( Nu \) & \( \Delta \) \\
Prandtl number & \( Pr \) & \( N/m^2 \) \\
heat input & \( q \) & \( \text{density} \) of water, \( \text{kg/m}^3 \) \\
mean flow velocity & \( U_m \) & \( \text{angle} \) of a microchannel, \( \text{deg} \) \\
Pressur drop in a microchannel & \( \Delta P \) & \\
\hline
\end{tabular}

Greek Symbols

\begin{tabular}{|l|l|}
\hline
\textbf{Symbol} & \textbf{Definition} \\
\hline
\alpha & \text{angle} of a microchannel, \text{deg} \\
\Delta & \text{difference} \\
\rho & \text{density} of water, \text{kg/m}^3 \\
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\end{tabular}

1. INTRODUCTION

Microscale heat exchangers are becoming an important area of interest in many fields of developing technology that require compact high heat energy removal solutions. Such fields as MEMS, microelectronics, biomedical, fuel processing, and aerospace are all pushing the limits of thermal control and are finding ways to make smaller devices with higher heat flux potential — requiring more efficient smaller heat exchangers to cool their key working components. Vist and Petterson (2004) conducted distribution experiments using R-22 and CO\textsubscript{2} in a horizontal manifold feeding 21 parallel heat exchanger tubes using countercflowing water jacket as a heat source. Among the researches that have been conducted within the academic and industrial communities are investigations of the validity of the macroscale equations of friction factor and Nusselt number on the microscale. Over the course of the past couple of decades, many conflicting accounts of results on the validity of classical macroscale equations for microchannel fluid flow and heat transfer have been given. Among these researches are the investigations of the validity of the macroscale equations for the friction factor, transition Reynolds number, and Nusselt number on the microscope. High heat transfer rate is the need of many systems in today's world in order to improve their performance by maintaining their operating temperature below acceptable levels as in the case of high performance computer chips (below 100\degree C), in microelectronic equipment. This can be achieved by both direct geometry advantage of "higher heat transfer area" and "higher heat transfer coefficient." For various heat and mass transfer applications, a microchannel device has received increasing attention for ap-
Applications such as microelectronics, microchannel reactors, microrockets, and microbiological systems. To cope with the demand for more efficient cooling technology for the next generation of high-power electronic devices, various kinds of microscale heat exchangers, and two-phase microchannel heat sink are required. Energy conversion, recovery, and generation often require high-effectiveness heat exchangers. Functions can include recuperation, waste heat recovery, boiling a working fluid, condensation, evaporation in cooling systems, and high heat flux removal. In many applications, emphasis is placed on small, light-weight, and high-performance devices, especially if the overall systems using heat exchangers are meant for portability, must be air-lifted or for retrofitting existing systems where space constraints are dictated. Microchannels are used in microscale heat pumps for microprocessor cooling or in portable cooling devices, small-scale stationary auxiliary power units, and a cryosurgical probe for ablating tumors or treating heart arrhythmia. The microchannels attract important research interest due to the rapid growth of applications in microelectronics. Among the researches that have been conducted within the academic and industrial communities there are investigations of the validity of the macroscale equations of friction factor and Nusselt number on the microscale. Over the course of the past couple of decades, many conflicting accounts of results on the validity of classical macroscale equations for microchannel fluid flow and heat transfer have been given. Among these researches there are the investigations of the validity of the macroscale equations for friction factor, transition Reynolds number, and Nusselt number on the microscale.

2. LITERATURE REVIEW

The literature review concerned the works carried out for a microchannel with a refrigerant and water as a medium. Both experimental and theoretical works and CFD analysis of microchannels were considered. Zhou and Fang (2013) evaluated the correlations required for calculation of boiling heat transfer of R-22 flowing in horizontal channels that takes place in chemical process cooling systems, refrigeration, and air conditioning. A number of correlations for flow boiling heat transfer in channels have been proposed. The work evaluates the existing correlations for flow boiling heat transfer with 1669 experimental data points for flow boiling heat transfer of R22 collected from 18 published papers. The top two correlations related to R-22 with the mean absolute deviation of 32.7% and 32.8%, respectively. Alagesan et al. (2012) studied flow boiling heat transfer in mini- and microchannels. They made comprehensive review of flow boiling heat transfer characteristics of various working fluids in mini- and microchannels. Due to their high heat transfer and heat removal properties, two-phase flows were studied and applied in electronics cooling systems. Correlations for two-phase heat transfer established by many researchers based on their experimental studies were compared with conventional correlations in both laminar and turbulent flow regimes. A large number of working fluids, configurations of
channels, and various ranges of parameters were considered in this review. The survey involves such range of studies as investigation of a single-phase flow, estimation of heat transfer in channels of various sizes, analytical studies in small tubes/channels, two-phase flow boiling in mini- and microchannels, design and testing of microchannel heat sinks for electronic cooling. Further development of the existing research in this area has been established.

Shah (2010) studied transfer during condensation inside small channels. He considered the prediction of heat transfer during film condensation in mini- and microchannels. No well-verified method for this purpose is available. The applicability of the authors' well-validated general correlation for condensation in tubes to small channels is investigated in this paper. A wide range of data for condensation in horizontal micro- and minichannels were compared. This correlation was found to predict 500 data points from 15 studies for small-diameter channels with a mean deviation of 15.9%. These data included single round and rectangular channels as well as multiport channels with round and rectangular ports with equivalent diameters from 0.49 to 5.3 mm, 8 fluids, reduced pressures from 0.048 to 0.52, and mass flux from 50 to 1400 kg/m²·s. This indicates its applicability to minichannels. However, a large quantity of data for diameters from 0.114 to 2.6 mm showed appreciable deviations from this correlation. The discrepancy in the overlapping range of data could be due to the difficulties of accurate measurements in small channels. Heat transfer during condensation in small-diameter channels is of great practical interest due to the needs of miniaturization. Numerous experimental studies were done to measure the heat transfer rate in mini- and microchannels. Many methods were proposed for predicting heat transfer coefficients, both theoretical and empirical, but none of them was shown to be generally applicable or even applicable in a well-defined range of parameters.

Foley (2015) studied a copper microchannel tube for HVAC. He studied microchannel heat exchangers of latest trend used in heating and cooling technologies with about a 40% increase in the efficiency. Most microchannel heat exchangers on the market today are constructed of aluminum. The invention is a copper microchannel tube. The intent of this invention is to create a more long-term and durable heat exchanging system for commercial and residential HVAC industries. The use of a copper microchannel tube is an engineering challenge as compared to aluminum microchannel tubes due to the increased temperatures required.

Lee et al. (2005) studied microchannels ranging in width from 194 μm to 534 μm, with the channel depth being nominally five times the width in each case. Each test piece was made of copper and contained ten microchannels in parallel. The Reynolds number ranged from approximately 300 to 3500. Tuckerman and Pease (1981) were first to suggest the use of microchannels for high heat flux removal; this heat sink is simply a substrate with numerous small channels and fins arranged in parallel, such that heat is efficiently carried from the substrate to the coolant. Their study was conducted for water flowing under laminar conditions through microchannels machined
in silicon water. Heat fluxes as high as 790 W/cm² were achieved with the chip temperature maintained below 110°C.

Vardhan and Dunn (1997) studied the heat transfer and pressure drop characteristics of R-22 in microchannel tubes at the Air-Conditioning and Refrigeration Center. They studied the heat transfer and pressure drop characteristics for single-phase and two-phase flows of refrigerants in microchannel tubes. The existing facility was modified to perform tests on a single microchannel tube using water flowing in counterflow as a heat sink fluid. Data were collected for a subcooled liquid, superheated vapor, and a condensing refrigerant for circular-port microchannel tubes at four different mass flow rates. The refrigerants tested were R-22, R-134a, and R-407C. In addition to circular port tubes, square-port tubes were also tested with R-22 for both single-phase and two-phase flows. The Reynolds number for all the flows in this study ranged from 3000 to 17,000 in the liquid phase and from 30,000 to 110,000 in the vapor phase. The heat transfer coefficients and pressure drop for circular-port microchannel tubes were compared for R-22. The heat transfer coefficient for R-22 was found to be slightly less than that for other two refrigerants for condensing flow. A model was built to predict the heat transfer and pressure drop in a microchannel tube. In most cases, the heat transfer and pressure drop predicted by the model agree very well with experimental results. A sensitivity analysis revealed that the total heat transfer is not very sensitive to the refrigerant side heat transfer coefficient.

Gunasegara et al. (2010) studied the effect of geometrical parameters on the heat transfer characteristics of microchannel sheet sink with different shapes and gave a summary of their work. They studied rectangular microchannel heat sinks. The smallest hydraulic diameter had the greatest value of heat transfer coefficient. The height-to-top width ratio (H/a), the bottom-to-top width ratio (b/a), and the length-to-hydraulic diameter ratio (L/Dₜ) are the important design parameters for trapezoidal microchannels. The Poiseuille number increases when H/a and L/Dₜ decrease while b/a increases. The tip angle of triangular microchannels has a great effect on the Poiseuille number. The Poiseuille number increases with the tip angle β from 22.14 to 51.95°. Numerical simulation of the fluid flow and heat transfer characteristics in full scale microchannel heat sinks was performed in the study. The effect of different geometrical parameters such as the differences in hydraulic diameters, height, and width of three different shapes of microchannel heat sinks was comprehensively studied.

Based on the simulated results, the following conclusions can be made:

i) for rectangular microchannel heat sinks, the smallest hydraulic diameter has the greatest value of heat transfer coefficient;

ii) the heat transfer coefficient and the Poiseuille number increase with the Reynolds number. For rectangular microchannels, the heat transfer coefficient and Poiseuille number are the highest, for triangular microchannels are the lowest, and for microchannels of trapezoidal shape are in between;
iii) the pressure drop decreases with increase in heat flux from the top plate of a
heat sink for the same Reynolds number;

iv) for rectangular microchannels, the width-to-height ratio \( \frac{W_c}{H_c} \) has a significant effect on the Poiseuille number. The Poiseuille number is the highest for \( \frac{W_c}{H_c} = 0.974 \), less for \( \frac{W_c}{H_c} = 0.651 \), and least for \( \frac{W_c}{H_c} = 0.391 \).

Sharp et al. (2005) studied liquid flows in microchannels. They showed that nominally microchannels can be defined as channels whose dimensions are less than 1 mm and greater than 1 μm. Above 1 mm the flow exhibits behavior that is the same as for most macroscopic flows. Currently, microchannels have characteristic dimensions anywhere from the submicron scale to hundreds of microns. Microchannels can be fabricated in many materials — glass, polymers, silicon, metals — by using various processes including surface micromachining, bulk micromachining, molding, embossing, and conventional machining with microuters. These methods and the characteristics of the resulting flow channels are discussed elsewhere. Microchannels offer advantages due to their high surface-to-volume ratio and their small volumes. The large surface-to-volume ratio leads to high rate of heat and mass transfer, making microdevices excellent tools for compact heat exchangers. For example, they present a crossflow heat exchanger constructed from a stack of fifty 14 × 14 mm foils, each containing 200-μm-wide by 100-μm-deep channels machined into the 200-μm-thick stainless steel foils by the process of direct, high-precision mechanical micromachining. The direction of flow in adjacent foils is alternated 90°, and the foils are attached by means of diffusion bonding to create a stack of crossflow heat exchangers capable of transferring 10 kW at a temperature difference of 80 K using water flowing at 750 kg/h. The impressively high rate of heat transfer is accomplished mainly by the large surface area covered by the interior of the microchannel: approximately 3600 mm² packed into a 14-mm cube. A second example of the application of microchannels is in the area of MEMS devices for biological and chemical analysis. The primary advantage of microscale devices in these applications are their good match with the scale of biological structures and the potential for placing multiple functions for chemical analysis on a small area; that is, the concept of a chemistry laboratory on a chip.

Kulkarni et al. (2002) analyzed refrigerant side tradeoffs in microchannel evaporators. They showed that microchannel condensers dominated the automotive a/c market by maximizing performance for a fixed size and weight, but must operate as evaporators if they are to be used in heat pumps. The new model structure provides a user friendly interface and makes it much easier to have good initial guesses. Two different approaches for predicting wetted surface heat and mass transfer are discussed and compared, and the effects of inlet humidity and inclination angle were explored experimentally. Superheat measurements were used, together with the model to detect significant refrigerant maldistribution that reduced the capacity by approximately 3%.
Frost patterns were used to observe two-phase flow distribution in a microchannel evaporator, and the balance between inertial, gravitational and shear forces was investigated for vertical headers.

Cho and Cho (2007) carried out experiments on the performance of microchannel evaporators with refrigerant R-22. The experiments were performed with both a vapor compression system and a refrigerant circulation system. Each evaporator was made of two parallel heat exchangers connected with several return pipes. The parallel flow heat exchanger had 41 microchannel tubes inserted between inlet and outlet headers. The microchannel tube had 8 rectangular ports with a hydraulic diameter of 1.3 mm. For the vapor compression system, the flow area ratio and the number of return pipes had a great effect on the cooling capacity. Type 3 with a flow area ratio of 73% and 58% showed the best cooling capacity. It had 12 return pipes and 3 circuits. There is a merging manifold in it. The cooling capacity increased proportionally as the vertical inclination angle of the evaporator increased due to the gravity force. Dharaiya and Kandlikar (2012) carried out numerical investigation of heat transfer in a rectangular microchannel under H2 boundary conditions in a fully developed laminar flow for five different cases, i.e., 4 walls, 3 walls, 2 walls, 1 wall, and 2 adjustable walls. They found that the numerical data obtained for four-wall H2 boundary conditions follow a linear trend as compared to the H1 and T boundary conditions. The value of the fully developed laminar flow Nusselt number for H1 and T thermal boundary conditions shows an exponentially decreasing trend with increase in the aspect ratio where H2 boundary conditions show linear increasing trends. In the present work, a rectangular microchannel with the aspect ratio from 0.1 to 10 is simulated numerically by using CFD software. Saenen and Baelmans (2013) studied the size effect of a portable two-phase electronics cooling system. They presented the results of numerical study and compared with an analytical model. The parameter study performed on the small size of the cooling system shows that increase in the boiling temperature leads to a decrease in the vapor quality or mass flow rate. An analytical model is developed to predict the size effect in a simple way. It is shown that there is a link between the connecting tubing volume, heat exchanger two-phase volume, accumulation, etc. Saenen and Baelmans (2013) carried out a numerical study of the size effect and portable system for usage and handling in easy manner.

Basu et al. (2011) studied the flow boiling of R134a in circular microtubes for critical heat flux conditions and found the experimental critical heat flux data and plotted the different graphs for an internal diameter of 0.50 mm, 0.90 mm, and 1.60 mm. The critical heat flux (CHF) increases with the mass flux and on increase in the inlet subcooling. The CHF decreases on increase in the quality in 0.90-mm and 1.60-mm ID tubes. The CHF was not clearly studied in 0.50-mm ID tubes. Experimental data were compared with correlations like the Katto–Ohno correlation, Bowring correlation, Thome correlations, and Zhnge correlations. The experimental
CHF data were mapped using the Hasan flow map. The Hasan flow map was used for flow region in a 0.50-mm ID tube at a CHF combined with intermittent, churn, and annular flow. The Thomsen map identified the flow regime as annular in all three test sections.

Bowers et al. (2012) studied the refrigerant distribution effect on the performance of a microchannel evaporator. They experimentally studied the flow pattern for various evaporator coil orientations with the refrigerant R410A flowing over coils. To better understand the effect exerted by orientation on refrigerant flow distribution, experiments were conducted to study the R410A distribution in three different orientations. Liquid refrigerant distribution was improved when a heat exchanger was rotated at 90° such that the header was oriented vertically, with heat flow entering at the bottom. Due to this, a modified condition test with no humidity was evaluated that showed that the improvement in the refrigerant flow distribution offers quantifiable improvement in the coil performances. Al-Hajri et al. (2008) performed the characterization of two selected refrigerants in a flat plate microtubes condenser for improvement of thermal devices and those in the aerospace industry. A number of graphs were plotted, i.e., pressure drop vs. mass flux, heat transfer coefficient vs. temperature, etc. In order to characterize the condensation heat transfer coefficient and pressure drop in a microchannel with a high hydraulic diameter and aspect ratio, a rectangular microchannel made of copper, with dimensions of 190 mm × 2.8 mm × 0.4 mm was fabricated. The tests conducted for the R134a and R245fa showed that the heat transfer coefficient and pressure drop for both refrigerants decrease with increase in the saturation temperature. The heat transfer coefficient was slightly affected by the change in the inlet superheating. A comparison of two refrigerants demonstrated that the heat transfer rate of the refrigerant R245fa is on the average by 15% better than that of R134a. The pressure drop of the refrigerant R245fa was almost three times higher than that of R134a for the same operating condition. Rosa et al. (2009) studied the single-phase heat transfer in a microchannel. They showed the importance of the scaling effect for studying the effects of enhancement, temperature properties, rarefaction, compressibility, conjugate heat transfer, viscosity heating, and surface roughness in microchannels. They obtained numerical and experimental results for heat transfer in microchannels by analyzing all of the aspects. In experiments with a single channel and when all scaling effects can be neglected, the Nusselt number is in good arrangement with the prediction by available correlations for microchannels. Mathematical mesoscopic and microscopic models for the subcontinuum effect in flow fields have been reviewed and explained. On the other hand, microscopic and mesoscopic models still require an enormous quantity of computational resources and run time, which make them still unsuitable for typical, geometrical complex, industrial fluid domain. Nomerotski et al. (2013) studied experimentally and made a graphical analysis of evaporative CO$_2$ cooling using microchannels etched in silicon for an LHCb VELO detector. They made microchan-
nals on silicon by using the process of etching on the chip (plate) for experimental analysis. The authors reported on the proof of the principle of measurement employing evaporative CO₂ cooling in silicon microchannels in the cortex of an upgraded LHCb VELO detector. We were able to remove 12.9 W of power before drying out a prototype sample. Habeeb and Al-Turaihi (2013) carried out an experimental study and made CFD simulation of two-phase flow around a triangular obstacle in enlarging channels and made a theoretical analysis of a bubbly flow around triangular bodies located horizontally. It was observed that the discharge increases turbulence flow. A complete study was made of different parameters like water discharge, pressure drop, flow pattern, bubble diameter, and air–water flow arrangements. Zeinali Heris et al. (2012) studied the CuO/water nanofluid heat transfer through a triangular duct. They analyzed numerically force-convection laminar flow of CuO/water nanofluid in a triangular duct. Nanoparticles added to a base fluid increase the heat transfer coefficient of fluid, and random movement of nanoparticles inside the fluid changes the structure of the flow field and leads to heat transfer enhancement. The results of the numerical solution show that a decrease in the nanoparticle size increases the Nusselt number in the case of specific concentration and an increase in the nanoparticle concentration increases the Nusselt number at a constant particle size and specifically increases the thermal conductivity. Equilateral triangular ducts yield a higher heat transfer coefficient than other triangular ducts.

2.1 Conclusions from Literature Review

The experimental results presented in the graphs for the temperature difference across the microchannels are by 10 to 20 times less than the theoretical results for different heat inputs and flow rates over 20-mm-, 30-mm-, and 40-mm-long pipes.

Many researchers have focused their attention on quantifying distribution and its governing variables through mass flow measurements. These measurements tend to be invasive and limit the ability to study the effects of refrigerant distribution in heat exchangers on both heat exchanger and system performance. Heat transfer coefficients increase with the exit pressure at the same exit quality for moderate and high mass fluxes. This is in agreement with findings in the literature for conventional channels and minichannels. For the most part, heat transfer coefficients display similar qualitative trends at reduced and atmospheric pressures. Depending on the mass flux and heat flux, both nucleate and convective dominant boiling mechanisms have been detected. A transition between nucleate and convective boiling was quantified. For subatmospheric pressure data, nucleate boiling is dominant for low Re and Bo numbers similar to atmospheric pressure data. However, the transition from the nucleate to convective dominant boiling heat transfer mechanism occurs at lower Re and Bo numbers than at atmospheric pressure.
3. THEORETICAL ANALYSIS

3.1 Introduction

Analysis of a rectangular microchannel was carried out for forced convection heat transfer condition on assumption that the base area was 30 mm in length and 20 mm in width with a microchannel manufactured on the top surface by the EDM process. In the theoretical analysis, heat is supplied to copper plates by a heater and water is flowing over these plates. The heat is lost from the copper plates by conduction due to the flow of water through it and by convention due to the surrounding air. In the theoretical analysis it is assumed that heat loss is negligible and flow of water over the plates is laminar.

A theoretical analysis can be done by theoretical calculations using various formulas and assumptions. We used the results of these calculations to compile tables and graphs. In what follows various conclusions can be drawn on the basis of these results. The assumptions for analysis are:

- the flow is laminar;
- the refrigerant used is R22 and flow through the cycle is single-phase only;
- the supplied heat is constant;
- both heat conduction and convection in a microchannel are considered;
- heat loss to the surrounding medium is negligible.

In the calculations the following terms are considered for the analysis of microchannels:

- heat input to a microchannel is varied from 25 W to 150 W.
- a flow rate considered during analysis is from 1 to 6 L/min.
- a microchannel length is 30 mm.

3.5 Heat Input vs. Heat Transfer Coefficient

Heat input vs. heat transfer coefficient for different flow rates is analyzed with five sets of hydraulic diameters, as shown in Figs. 1–5. As the heat input increases, there is an increase in the heat transfer coefficient. The graph shows the horizontal characteristics. In nature, for different flow rates the heat transfer coefficient increases with the diameter, as shown in Figs. 1–5. For a flow rate of 0.0166 kg/s in the microchannel with a hydraulic diameter of 260 μm the heat transfer coefficient is maximum and equal to 10,000 W/m²·K and for a 330-μm-diameter microchannel it is minimum and equal to 2000 W/m²·K. Similarly, for a flow rate of 0.063 kg/s, for a hydraulic diameter of 260 μm the heat transfer coefficient is 10,000 W/m²·K and for 330-μm hydraulic diameter it is observed to be about 2200 W/m²·K. For flow rates of 0.06 kg/s and 0.1 kg/s, i.e., as the flow rate increases with the heat input, the heat transfer coefficient also increases. It is seen that for a higher hydraulic diameter the heat transfer coefficient increases with the flow rate. For a lower hydraulic diameter the heat transfer coefficient decreases with the flow rate.
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FIG. 1: Model of a microchannel

FIG. 2: Heat input vs. heat transfer coefficient for a flow rate of 0.0016 kg/s

FIG. 3: Heat input vs. heat transfer coefficient for a flow rate of 0.033 kg/s
FIG. 4: Heat input vs. heat transfer coefficient for a flow rate of 0.066 kg/s

FIG. 5: Heat input vs. heat transfer coefficient for a flow rate of 0.1 kg/s

FIG. 6: Heat input vs. pressure drop for a flow rate of 0.0166 kg/s
coefficient is maximum, as shown in Figs. 1–5. It is seen that for different flow rates for a hydraulic diameter of 260 μm the heat transfer coefficient is maximum and for rest hydraulic diameters the heat transfer coefficient is in range from 14,000 W/m²·K to 2000 W/m²·K.

3.6 Heat Input vs. Pressure Drop

From the graph of heat input vs. pressure drop we have obtained the following results (Figs. 6–8). As the heat input increases, the pressure drop changes drastically from a negative to a positive value. For the different flow rates the pressure drop tries to attain positive values. The graph lines are horizontal in nature. With increase in the hydraulic diameter, the pressure drop decreases. The graph of pressure difference vs. heat input for a flow rate of 1 L/min and for various hydraulic diameters is shown.
It has been observed that for a hydraulic diameter of 260 μm the pressure drop is maximum, i.e., 0.025 kPa, and remains constant for all heat inputs. For a hydraulic diameter of 343 μm, the pressure drop is minimum. It is seen from the graph that for all hydraulic diameters the pressure drop is constant for all heat inputs. The graph of pressure difference vs. heat input is drawn for the flow rate 2 L/min and for various hydraulic diameters. It has been observed that for a hydraulic diameter of 260 μm the pressure drop is maximum and remains constant for all heat inputs. But for a hydraulic diameter of 343 μm, the pressure drop is minimum. It is seen from the graph that for all hydraulic diameters the pressure drop is constant. The graph of pressure difference vs. heat input was drawn for the flow rate 6 L/min and for various hydraulic diameters. It has been observed that for a hydraulic diameter of 260 μm the pressure drop is maximum and remains constant for all heat inputs. But for a hydraulic diameter of 343 μm, the pressure drop is minimum. It is seen from the graph that for all hydraulic diameters the pressure drop is constant.

3.7 Heat Input vs. Temperature Difference

Different graphs in Fig. 9–12 are given. Five sets of configurations of the microchannel were considered.

As the heat input increases, the difference in temperature also increases. The graph shows linear characteristics. We have obtained graphs for different flow rates from 1 L/min to 6 L/min. With increase in the hydraulic diameter, the temperature difference increases, as shown in Figs. 9–13. The graph of the temperature difference $\Delta T$ vs. heat input $q$ for various hydraulic diameters is shown. The flow rate here is 1 L/min. It has been observed that for a hydraulic diameter of 0.000370 m the temperature difference is maximum, i.e., 80°C for 100 W, while for a heat input of 20 W, $\Delta T$ is 16°C.

**FIG. 9:** Heat input vs. pressure drop for a flow rate of 0.1 kg/s
A Rectangular Microchannel Using the R-22 Refrigerant

FIG. 10: Heat input vs. temperature difference for a flow rate of 0.016 kg/s

FIG. 11: Heat input vs. temperature difference for a flow rate of 0.033 kg/s

FIG. 12: Heat input vs. temperature difference for a flow rate of 0.066 kg/s
For a hydraulic diameter of 343 \( \mu m \), \( \Delta T \) is 70°C for 100 W and 15°C for 20 W, whereas for a hydraulic diameter of 260 \( \mu m \), \( \Delta T \) is 60°C for 100 W and 12°C for 20 W. The temperature difference \( \Delta T \) is minimum for a hydraulic diameter of 260 \( \mu m \), i.e., 12°C at 20 W and 58°C at 100 W. It is seen from the graph that for all hydraulic diameters the temperature difference varies linearly with respect to the heat input. The graph of \( \Delta T \) vs. \( q \) for a flow rate of 2 L/min and various hydraulic diameters is given. It is observed that for a hydraulic diameter of 343 \( \mu m \), \( \Delta T \) is maximum, i.e., 80°C for a 100-W heat input, while \( \Delta T \) is minimum for a hydraulic diameter of 260 \( \mu m \) at 20 W. It is seen from the graph that for all hydraulic diameters \( \Delta T \) varies linearly with respect to the heat input. The graph of \( \Delta T \) vs. \( q \) for a flow rate of 6 L/min and various hydraulic diameters is given. It is observed that for a hydraulic diameter of 343 \( \mu m \), \( \Delta T \) is maximum, i.e., 71°C for a 100-W heat input, while \( \Delta T \) is minimum for a hydraulic diameter of 260 \( \mu m \) at 20 W. It is seen from the graph that for all hydraulic diameters \( \Delta T \) varies linearly with respect to the heat input.

4. EXPERIMENTAL ANALYSIS

4.1 Experimental Setup

The setup consists of a compressor from which a cooling refrigerant is circulated (Figs. 14 and 15). The compressor is used for compressing the refrigerant up to a sufficiently high pressure and temperature, with the compression being done on vapor. Then there is condenser where the refrigerant is cooled at a constant pressure by rejecting heat to the condenser. The liquid from the condenser goes to an expansion valve, where the temperature and pressure reduce, and is converted into vapor. Finally, the low-pressure, low-temperature flow passes to the manifold where it absorbs...
latent heat at a constant pressure and converts it into a vapor state. Then again it is supplied to the compressor and thus the cycle is completed. The accumulator stores the energy for the compressor. A heater is used for heating the manifold. The flow is measured by a rotameter. Piping is for the cooling refrigerant circulation. The manifold has the inlet and outlet and a copper microchannel in it.
4.2 Experimental Results on Heat Input vs. Temperature Difference

Now for experimental results we took only the temperature readings, and represented the temperature difference vs. the heat input. The graphs of experimental results for temperature difference across the microchannels yield the values 10 to 20% less than the theoretical results for different heat inputs, flow rates for the 30-mm-long pipe (Figs. 16–19). As the heat input increases, the configuration with a higher hydraulic diameter shows a decrease in the temperature difference of the refrigerant. The temperature rise of the refrigerant R-22 ranges from 0.89 to 28.95 with various configurations for different heat inputs.

![Graph showing heat input vs. temperature difference for a flow rate of 0.016 kg/s](image1)

**FIG. 16:** Heat input vs. temperature difference for a flow rate of 0.016 kg/s

![Graph showing heat input vs. temperature difference for a flow rate of 0.033 kg/s](image2)

**FIG. 17:** Heat input vs. temperature difference for a flow rate of 0.033 kg/s
5. CONCLUSIONS

It has been found that the heat transfer coefficient for a hydraulic diameter of 260 μm is equal to 12,000 W/m²·K. As compared to other configurations, the heat transfer coefficient is higher for the 260-μm-diameter microchannel, so it is more optimistic as compared to other configurations of the microchannel. Also, as the heat input increases, the configuration with a higher hydraulic diameter shows a smaller pressure drop as compared to a hydraulic diameter of 260 μm. The pressure drop for a microchannel is 0.026 kPa. For a hydraulic diameter of 370 μm the pressure drop is minimum and
equal to 0.01 kPa. The experimental setup has been developed to test the temperature difference across the microchannel for a heat input from 25 W to 150 W for a 30-mm-long microchannel. Experimental results show that for the 30-mm-long microchannel of a hydraulic diameter of 260 μm the temperature rise of the refrigerant R22 lies in the range from 2°C to 18°C for a single-phase refrigerant flow. The graphs show that the experimental results for the temperature difference across the microchannels is 10 to 20% less as compared to theoretical results for different heat inputs and flow rates for a 30-mm-long channel. The range of applicability allows comparison of the refrigerant distribution in different designs of microchannel heat exchangers. This method also has the advantage of being both noninvasive and low cost.

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REFERENCES


A Rectangular Microchannel Using the R-22 Refrigerant


