ABSTRACT
The development of advanced microchannel heat exchangers and microfluidic devices is dependent upon the understanding of the fundamental heat transfer processes that occur in these systems. There have been great advancements in our understanding of the heat transfer and fluid flow mechanisms that occur in microchannels. There is several research areas in microchannel heat transfer that promise for such applications as microprocessor cooling.

An enhanced microchannel heat exchanger (EMCHX) that uses single-phase liquid flows has been developed. This EMCHX uses flow obstructions to create a continually developing flow condition and the enhancement in heat transfer associated with that flow regime. A silicon substrate is chosen to create offset strip fins in the microchannel flow field. Experimental verification of this new method shows excellent improvement in heat transfer over plain or traditional microchannels with straight, continuous walls. However, careful attention must be paid to the added pressure drop that is created by adding these obstructions.

A new microchannel parameter called pumping power flux is developed to aid in the comparison between plain and enhanced microchannels. The pumping power flux is used in conjunction with the heat flux to calculate a coefficient of performance to demonstrate the heat transfer enhancement. The enhanced microchannels provide a much higher COP for the same flow conditions. Therefore, the improved heat transfer provided outweighs the added pressure drop caused by the enhanced microchannels.

INTRODUCTION
Microchannel heat transfer has gained importance as a method to support very high heat transfer rates. This nascent technology is still being heavily researched and is simultaneously being developed into commercial applications. An attractive application for microchannel heat transfer is microprocessor cooling. As the transistor density increases in packages, the amount of dissipated heat is also increased. There is a very active discussion about what mode of heat transfer, single-phase or two-phase flow, will be the best choice to support the increasing microprocessor heat fluxes.

A large percentage of the active research in microchannel heat transfer involves two-phase flows. Two-phase heat transfer does indeed dissipate large heat fluxes on the order of tens of MW/m². However, the two-phase flow system comes with a few more complications versus a comparable single-phase flow system. The two-phase pressure drop will be much higher than the single-phase. In addition, the two-phase flow system would also require a condensation step in the closed loop system. Microchannel condensation is also a developing technology that requires further work to understand all of the physics involved.

With all of the attention that two-phase microchannel heat transfer is getting, a great opportunity for understanding the fundamental physics that occurs in single-phase microchannel heat transfer is being missed. This is not to say that the two-phase heat transfer is not important and will not provide the technology to cool future microprocessors. The transition at hand for this industry is from advanced air-cooling to the next cooling medium. The authors believe that there is tremendous benefit to transition into single-phase microchannel heat transfer prior to implementing two-phase microchannel heat transfer.

Utilizing single-phase microchannel heat transfer for high heat flux microprocessors is a viable option for several reasons. First, the overall system complexity is reduced for a single-phase system. Secondly, the microchannels can be enhanced to provide improved overall heat transfer coefficients.

Steinke and Kandlikar (2004) reviewed and identified several possible techniques to enhance the heat transfer in microchannels. It is important to provide a simple transition from air-cooling into a different medium. The final reason is to allow the time to further develop understanding of two-phase heat transfer systems and provide an existing liquid based
infrastructure to introduce two-phase heat transfer systems into microprocessor cooling.

LITERATURE REVIEW
Understanding single-phase microchannel heat transfer is important to allow for the ultimate implementation of this technology in industry. The validity of the conventional heat transfer theories in microchannel passages is a subject of major interest. There is much debate about the validity of conventional heat transfer theories for microchannels.

Steinke and Kandlikar (2006) reviewed the available literature for single-phase liquid flow in microchannels. They attributed the reported discrepancies to improperly accounting for microchannel geometry, entry and exit losses, entrance lengths, and experimental uncertainties. When the authors properly accounted for these parameters, the conventional fluid flow theory provides strong agreement with experimental data. Therefore, the flow physics can be described with the traditional Reynolds number and friction factor.

Steinke and Kandlikar (2005) reviewed the available literature for single-phase liquid heat transfer in microchannels. Following the same procedure, they identified improved agreement with conventional heat transfer theory when the previously mentioned parameters are accounted for. Therefore, the conventional Nusselt number and heat transfer coefficients should apply to microchannel heat transfer.

One issue presented in those works is that all authors do not consistently report the microchannel geometry and aspect ratio. For rectangular channels, there have been several different definitions for channel aspect ratio in literature. The channel aspect ratio for the present work is given by Eq. (1).

\[ \alpha_c = \frac{a}{b} \]  

(1)

where \( a \) is the microchannel width and \( b \) is the microchannel height.

The orientation of the geometry and the applied boundary conditions are crucial for heat transfer. It would be desirable to have an \( \alpha_c < 1.0 \) to get several deep, narrow microchannels with than to have a few wide, shallow microchannels from a heat transfer perspective. The fin aspect ratio is defined as the fin thickness divided by the fin height, Eq. (2).

\[ \alpha_f = \frac{s}{b} \]  

(2)

where \( s \) is the fin thickness. This ratio is a measure of the geometry of the fin structure. It is desirable to have all, thin fins as opposed to short, wide fins. The present work presents this definition of aspect ratio in a hope to spark this discussion and provide a consistent definition to allow for future direct comparisons.

The focus of the present work is the development of an enhanced microchannel that provides improved heat transfer. Steinke and Kandlikar (2004) reviewed conventional heat transfer enhancement techniques. They identified several techniques that can be used in microchannels and microchannels to improve the heat transfer. The present work develops the concept of breaking up the developing flow profile. The result is a continually developing flow that maintains a very high heat transfer coefficient.

In addition, Colgan et al. (2005) presented an enhanced microchannel heat exchanger with a similar structure. Their main focus was developing a microchannel heat exchanger that could be directly attached to an existing microprocessor. They were able to achieved maximum heat flux of 500 kW/m².

Once fabricated, the enhanced microchannels are tested using an experimental facility developed by Steinke et al. (2006). The authors reviewed several experimental microchannel papers to develop an experimental test bed that can investigate microchannel pressure drop and heat transfer for a wide range of parameters. The facility can accurately measure flow rates, pressure drops, and heat transfer for the enhanced microchannel heat exchanger.

EXPERIMENTAL UNCERTAINTIES IN MICROCHANNELS
The experimental uncertainties can become quite large for a microchannel heat exchanger. Some of the challenges include the physical size of the system being measured and the magnitudes of the measurements. The heat transfer occurring in microchannels is very efficient. Therefore, the temperature differences between the liquid and the walls can be very small. The \( \Delta T \) can be only a few degrees or less.

Fortunately, several of the standards for experimental uncertainties still apply at the microscale. The two best standards for determining experimental uncertainties are ASME PTC 19.1 (1998) and NIST Technical Note 1297 (1994). There are many similarities between these standards and many published works. In general, the total uncertainty is comprised of two parts - systematic error and random error as given by:

\[ U = 2 \left( \frac{B}{2} \right)^2 + \left( \frac{\sigma}{\sqrt{N}} \right)^2 \]  

(3)

where \( U \) is the total uncertainty, \( B \) is the bias error, \( \sigma \) is the standard deviation, and \( N \) is the number of samples. The bias error is a measure of the systematic error and the precision error is a measure of the random errors in the system.

A complete analysis of the experimental uncertainty for fluid flow and heat transfer can be found in Steinke and Kandlikar (2006, 2005). They determined the experimental uncertainties for calculated variables such as \( \beta \text{Re} \) and \( \beta \text{Nu} \) in terms of the actual measured parameters. The present work uses the results of these works to base the reported uncertainties.

DESIGN OF ENHANCED MICROCHANNELS
The fabrication of an enhanced microchannel heat exchanger that can be fabricated with today’s technology and can be easily integrated for microprocessor cooling is presented. The most obvious choice is using passive flow obstructions to create heat transfer enhancement. The process should not include any additional process steps and it should be inline the traditional silicon micromachining.

The idea is simply to replace the traditional continuous microchannel walls with short fins that also alternate in position. This is very similar to the concept of offset strip fins in compact plate heat exchangers. Figure 1 shows the concept of using microchannel offset strip fins. The fluid enters a flow passage and travels in the flow direction a short distance. Next, the fluid encounters a flow obstruction in the form of another fin. The fluid must separate and move around the next fin. It then travels the length of the second fin and encounters another
fin obstruction. The entire process is repeated as the fluid moves down the entire flow length. As a result, the fluid never enters into a fully developed state and must split and move around the next fin. This device will make use of the enhancement found in developing flows. In addition, the resulting mixing will force the fluid to have more interaction with the heated walls and also provide heat transfer enhancement.

Figure 1: Concept of Using Offset Strip Fin Geometry for Microchannel Enhancement.

Unfortunately, this heat transfer enhancement will come at the price of added pressure drop for the entire microchannel heat exchanger. The added heat transfer performance must be carefully weighed with the added pressure penalty.

Due to the continually developing flow in the enhanced microchannels, the pressure drop and heat transfer performance will be difficult to compare to conventional theory. This device should be treated as a compact heat exchanger. For a conventional compact heat exchanger, the friction factor and the Colburn $j$ factor are used as the main metrics of performance. These parameters are used to describe the overall performance of the compact heat exchanger. They are used due to the wide variety of complex geometries that occur in current compact heat exchangers. It should be noted that the friction factor for compact heat exchangers is referring to the apparent friction factor and not the purely frictional flow losses friction factor in conventional theory. For the present work, they will be utilized to describe the overall performance of the microchannel heat exchangers and provide a basis of comparison.

A new expression for the hydraulic diameter for the offset strip fin is needed to compare the enhanced microchannels to the plain microchannels. Figure 2 shows the nomenclature for the enhanced microchannels. A unit cell is define in the flow geometry and seen by the dashed lines.

The flow passage width is $a$, the fin thickness is $s$, $L_f$ is the fin length, and $g$ is the gap between successive fins. The unit cell length is the length of the fin plus the gap distance and the unit cell width is the flow passage width plus the fin thickness. The hydraulic diameter can be derived based upon the compact heat exchanger formulation given in Eq. (6)

$$D_h = \frac{4 A_{\text{Free Flow}} L_{\text{Unit Cell}}}{A_{HT}}$$

where $L_{\text{Unit Cell}}$ is the unit cell length and $A_{\text{Free Flow}}$ is the minimum free flow area. The minimum free flow area is given by Eq. (7) and the heat transfer area is given by Eq. (8).

$$A_{\text{flow}} = b a$$

$$A_{HT} = 2 b s + g s + a g + a L_f + 2 b L_f$$

The resulting expression for the offset strip enhanced microchannel geometry is given by Eq. (9).

$$D_h = \frac{4 A_{\text{Free Flow}} L_{\text{Unit Cell}}}{A_{HT}} = \frac{2 a b (L_f + g)}{s \left( b + \frac{g}{2} \right) + \frac{a g}{2} + \frac{L_f}{2} (a + 2 b)}$$

In this formulation, the length of the fin and gap dimensions plays a role in the resulting hydraulic diameter. For the present work, there is no gap used in the reported geometries.

Figure 3: SEM Micrograph of Enhanced Microchannels.

The ends of the fin structures are tapered to form a tip. The reason for this detail is to assist the flow in separating as it moves around the obstruction. The flow should be in continually developing state. However, the fluid could separate from the fin and the enhancement would be lost if the ends are too blunt. An 80 degree included angle is used to perform this function. The geometries in the present work do not include a
gap between the successive rows of fins. The sharpened ends give enough free area in order to allow the fluid to pass the fin. Figure 3 shows an SEM micrograph of the enhanced microchannel geometry. This image is taken at the inlet header and the sidewall flow obstructions are also present.

There are four different enhanced microchannel nominal geometries seen in Table 1. The pitch is fixed at 100 μm for all of the geometries. The flow passage width is either 50 or 60 μm and the fin thickness is either 50 or 40 μm, respectively. The fin lengths are also varied to achieve different levels of flow development on the fin surfaces. The equivalent hydraulic diameters range from 76 to 98 μm.

Table 1: Enhanced Microchannel Nominal Geometries.

<table>
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<tr>
<th>Number</th>
<th>a</th>
<th>s</th>
<th>Lf</th>
<th>Dn</th>
<th>n</th>
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<td>500</td>
<td>98</td>
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<td>10000</td>
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</tbody>
</table>

Figure 4 provides a closer look at the fin geometry. As previously stated the ends of each fin are sharpened in order to reduce the pressure loss from blunt end and to allow fluid to pass through each successive layer. Notice that the fin shape is clearly defined and the edges are very sharp.

The surface roughness was also measured for the enhanced microchannels. Figure 6 shows a SEM micrograph of the inlet to the microchannels as seen from the manifold area. The side wall roughness is the same as the plain silicon microchannels previously discussed and has an e/Dh ratio of 0.01. The walls of the fins have a rougher texture with an e/Dh ratio of 0.03. The bottom wall seems to have a slightly rougher surface than the plain microchannels with an e/Dh ratio of 0.007. Once again, it would be difficult to predict exactly which surface roughness values dominates the flow behavior without experimental data. However, one could speculate that the surface with the most fluid contact and area would tend to dominate over the others. Therefore, it is believed that the dissipated by the fin must move through that minimum area. Also, notice the slight sculpting of the fin surface due to ruminates of the mask.

The surface roughness of microchannel may take on a more dominate role. Although this will have little to no effect upon laminar flow, it will become critical to determine the surface roughness of the flow passages for a turbulent flow case.
EXPERIMENTAL RESULTS FOR ENHANCED SILICON MICROCHANNELS

The experimental results for the enhanced microchannel heat exchanger will be presented. The enhanced microchannels will be compared to plain or traditional straight, continuous walled microchannels. These plain microchannels have not been described in detail. However, the plain microchannels are fabricated from the same silicon substrates and are compared to the enhanced microchannels using similar hydraulic diameters.

The thermal performance is evaluated under the scope of the microprocessor cooling application. Therefore, the results will be presented in terms of the thermal resistances. One must carefully evaluate the added pressure drop the enhancement features create and will be seen using the apparent pressure drop across the microchannel test section.

Finally, a new parameter called the pumping flux will be introduced. It is used to provide a direct comparison between plain and enhanced microchannels. Using this parameter, a coefficient of performance is introduced concerning a microchannel heat exchanger.

Microchannel Heat Exchanger Thermal Resistance

The main parameter utilized in microprocessor cooling is the thermal resistance. It is simply a measure of the resistance of the heat exchanger to dissipate the input power. An advantage of using this parameter is that it reduces the complication of local behavior and provides an overall metric of the performance. The unit thermal resistance is given in Eq. (10).

\[ R = \frac{(T_j - T_i)}{q} \]

where \( R \) is the unit thermal resistance, \( T_j \) is the junction temperature, and \( T_i \) is the inlet fluid temperature. The heat transfer area for a unit cell is given by Eq. (8). The total heat transfer area is calculated counting the number of unit cells that occur in the microchannel heat exchanger and is used as the total heat transfer area.

To investigate the effect of the offset strip fin enhancement, several experiments are conducted with the plain geometries and the enhanced geometries. The volumetric flow rate into the test section is fixed at three different levels to create three flow cases. The three flow rates are 39 mL/min, 53 mL/min, and 69 mL/min. This causes the mass flow rates to be equivalent for comparison purposes. The actual Reynolds numbers may very slightly due to geometry and temperature dependent property variations. However, the comparison can still be made on a quantitative basis.

The plain microchannel geometries of G01-001, G01-003, and G01-005, as seen in Table 1, will be used for a base performance case. These microchannel geometries are chosen because they all have a fin thickness of 40 μm which is the same or close to the fin thickness of 40 μm and 50 μm in the enhanced microchannels. Finally, a microchannel depth of 200 μm for both the plain and the enhanced microchannels is used.

Figures 7, 8 and 9 shows the thermal resistance versus Reynolds number for the flow rates of 39, 53, and 69 mL/min, respectively. The plain microchannel geometries are represented by open points and the enhanced microchannel geometries are represented by closed points. The prefix G01 refers to the plain microchannels and G02 refers to the enhanced microchannels. The microchannel depth is 200 μm for each of the microchannel geometries. The inlet water temperature is set to 20 °C for all of the experiments. The uncertainty bars are included for completeness, although they are difficult to see due to the large range represented.
°Cm²/W. However, the enhanced microchannels seem to reach a thermal resistance of approximately 0.001 °Cm²/W, irrespective of microchannel geometry. This change can be attributed to the flow never becoming fully developed, promoting mixing, and increased fluid surface interaction which lead to an increase in the conduction at the walls.

Figure 9: Thermal Resistance vs. Reynolds Number. For $Q = 69$ mL/min; Plain = G01-001, G01-003, and G01-005; Enhanced = G02-033, G02-034, G02-35, and G02-036; $b = 200$ μm; $T_i = 20$ °C.

As mentioned, the Reynolds numbers vary for each of the geometries. One reason is the variation of the fluid properties based upon the temperature changes occurring in the microchannel. The water can rise as much as 30 °C for the higher heat flux cases causing significant variation in viscosity, which has been accounted for. After seeing the very large magnitude change in thermal resistance demonstrated by the enhanced microchannels, the small differences in the Reynolds number would not account for the change and therefore is not of concern.

To understand the amount of heat that is being dissipated, the heat flux versus the Reynolds number for a flow rate of 38 mL/min is shown in Fig. 10. In this figure the uncertainty bars are more readily seen and once again, cannot account for differences seen. As expected, the enhanced microchannels are dissipating substantially more heat than the plain microchannels. In order to match this performance, the plain microchannels would need to operate at much higher flow rates. The design point of the present work is 200 W/cm². Current microprocessors are operating around 200 W/cm² and would easily support this heat flux. The enhanced microchannels are dissipating around 1,000 W/cm² and become competitive with the heat flux of a two-phase microchannel system. However, the enhanced microchannels come with added pressure drop of only a few tens of kilopascals versus the plain microchannels.

**Apparent Pressure Drop**

It is important to investigate the added pressure drop in the enhanced microchannels. The added pressure penalty must be carefully considered when designing an entire liquid cooling system. It is critical to measure the overall pressure drop in order to properly size a fluid pump. For the remainder of this discussion, the pressure drop referred to is actually the apparent pressure drop. The measured pressure drop in the inlet and outlet plenums includes some inlet and exit losses. Those losses are subtracted from the measured pressure drop, Steinke and Kandlikar (2006), and the remaining pressure drop is the apparent pressure drop.

Figure 10: Heat Flux vs. Reynolds Number. For $Q = 69$ mL/min; Plain = G01-001, G01-003, and G01-005; Enhanced = G02-033, G02-034, G02-35, and G02-036; $b = 200$ μm; $T_i = 20$ °C.

Figure 11: Apparent Pressure Drop vs. Reynolds Number. For $Q = 39$ mL/min; Plain = G01-001, G01-003, and G01-005; Enhanced = G02-033, G02-034, G02-35, and G02-036; $b = 200$ μm; $T_i = 20$ °C.

For the enhanced microchannels, it is very difficult to separate the pressure drop due to frictional losses from those due to developing flows since the flow never reaches the fully developed condition. Keeping the pressure drop in the apparent form is also in agreement with traditional compact heat exchanger analysis. Figures 11, 12, and 13 show the apparent pressure drop for the three different flow rates. In each case,
the enhanced microchannels have a substantially higher pressure drop than the plain microchannels. This is an expected result due to the added flow blockage presented by the inclusion of the fin structures.

![Figure 12: Apparent Pressure Drop vs. Reynolds Number. For Q = 53 mL/min; Plain = G01-001, G01-003, and G01-005; Enhanced = G02-033, G02-034, G02-35, and G02-036; b = 200 μm; Ti = 20 °C.]

![Figure 13: Apparent Pressure Drop vs. Reynolds Number. For Q = 69 mL/min; Plain = G01-001, G01-003, and G01-005; Enhanced = G02-033, G02-034, G02-35, and G02-036; b = 200 μm; Ti = 20 °C.]

**Coefficient of Performance Using Pumping Flux**

It can become very difficult to compare different microchannel heat exchanger geometries. There is a wide variety of geometries and boundary conditions employed in these heat exchangers. A parameter that can measure the overall performance of the microchannel heat exchanger seems to be missing. The thermal resistance is excellent but it provides no information regarding the pressure penalty involved with the heat exchanger. A parameter that would account for both the heat flux being dissipated and the pressure drop required would be desired. Therefore, a novel parameter called the pumping power flux has been developed.

The pumping power flux is a new parameter proposed here to use in microchannel heat exchangers. It is based upon the idea of pumping power and is seen in Eq. (11).

\[
P = \frac{m \Delta p}{\rho}
\]

(11)

where \( P \) is the pumping power. The pumping power can be manipulated to transform it into a flux term by dividing by the free flow area of the flow. Equation (12) shows the formula for the pumping power flux, or the pumping flux for short.

\[
P'' = \frac{m \Delta p}{\rho A_{free}}
\]

(12)

where \( P'' \) is the pumping power flux and \( A_{free} \) is the free flow area. This parameter now has the same units and meaning as the heat flux. The pumping power flux is the amount of pumping power per unit free flow area in the microchannel. Another useful form of the pumping power flux is found in Eq. (13) where the mass flow rate is replaced with the mass flux.

\[
P'' = \frac{G \Delta p}{\rho}
\]

(13)

The heat flux and pumping power flux can be related in a simple manner by considering the coefficient of performance (COP) in thermodynamics. In its simplest terms, the COP is the desired output divided by the required input as in a refrigeration system. It is the heat removed by the cycle divided by the work done by the cycle. It would be desirable to have as high a COP as possible.

For the present work, the COP will be the desired heat flux to be dissipated divided by the required pumping power flux, Eq. (14).

\[
COP = \frac{q''}{P''}
\]

(14)

Substituting in known parameters gives Eq. (15) and rearranging gives Eq. (16).

\[
COP = \frac{\rho q}{m \Delta p A_{free}}
\]

(15)

\[
COP = \frac{\rho q \Delta p}{m A_{HT}}
\]

(16)

The COP terms out to be a ratio of the heat transfer area to the free flow area. If one could significantly increase the heat transfer area with all others remaining unchanged, the COP would increase. This is the goal of the heat transfer enhancements discussed in the present work. The free flow is reduced slightly but the heat transfer area is significantly increased, as a result the COP is increased. The COPs for the three flow rates are given in Fig. 14, 15, and 16.

It can be seen from the figures that the enhanced microchannels have a much higher COP than the plain microchannels. The plain microchannels have a seemingly linear relationship in a log-log plot. That relationship is also seen in the enhanced microchannels. However, the slope is much larger and allows the COPs for those cases to be several
times larger than the plain microchannels at the same Reynolds numbers. With very high COPs, the enhanced microchannels actually cost less to dissipate more amount of heat even with the increase in pressure drop over the plain microchannels.

The evaluation of the enhanced microchannels is challenging. The thermal performance is increasing at the same time the pressure drop is increasing. A careful evaluation of the net benefit gained by the microchannel enhancement is required. The COP defined in, based upon the heat flux and the pumping power flux, this work is intended to provide a metric used in such an evaluation.

An enhanced microchannel heat exchanger has been designed, fabricated and tested for thermal performance and pressure drop penalty. There is a marked improvement for heat that can be dissipated for the same footprint area. There is an increase in pressure drop for the enhanced microchannel heat exchanger. However, the coefficient of performance states that the benefits in heat transfer performance out weigh the increase in pressure drop. This enhancement technique is viable for high heat flux applications and further study is recommended to optimize the geometry.

Figure 14: Coefficient of Performance vs. Reynolds Number. For Q = 39 mL/min; Plain = G01-001, G01-003, and G01-005; Enhanced = G02-033, G02-034, G02-35, and G02-036; b = 200 μm; Ti = 20 °C.

Figure 15: Coefficient of Performance vs. Reynolds Number. For Q = 39 mL/min; Plain = G01-001, G01-003, and G01-005; Enhanced = G02-033, G02-034, G02-35, and G02-036; b = 200 μm; Ti = 20 °C.

Figure 16: Coefficient of Performance vs. Reynolds Number. For Q = 39 mL/min; Plain = G01-001, G01-003, and G01-005; Enhanced = G02-033, G02-034, G02-35, and G02-036; b = 200 μm; Ti = 20 °C.

Figure 17: Coefficient of Performance vs. Reynolds Number. Open square points - Previous Data; Open Diamonds - Present Work Plain Microchannels; and Closed Circles - Present Work Enhanced Microchannels.

Finally, the COP for the plain and enhanced microchannels can be compared to the COP for the previous literature. Only the previous literature that reports the heat flux and pressure drop can be used for the comparison. Figure 17 shows the comparison of the present work to the available data. The open squares represent the previous literature data, the open diamond
points represent the present work’s plain microchannels, and the closed circles represent the present work’s enhanced microchannels.

It can be seen that the plain microchannels for the present work fall into the same order of magnitude COP as the previous literature data. This validates the present work plain microchannels. It can be seen that the enhanced microchannels achieve a higher COP for a give flow rate than the plain microchannels. An also achieve that higher COP at a lower Reynolds number as compared to the previous literature. As a result, the enhanced microchannels definitely provide an improvement over plain microchannels.

CONCLUSIONS

The following conclusions are drawn from the present work. There are over 150 papers specifically addressing the topic of fluid flow and heat transfer in microchannels. The reported discrepancies from conventional fluid flow and heat transfer theories can be explained by not properly accounting for the entrance and exit losses, the developing flow, and the experimental uncertainties. Therefore, conventional fluid flow and heat transfer theory apply for single-phase liquid microchannel fluid flow and heat transfer.

An offset strip fin enhanced microchannel heat exchanger is fabricated and tested. The results show a marked improvement in the overall thermal resistance of the enhanced microchannel heat exchanger compared to one using plain microchannels. The unit thermal resistance of the plain microchannels is approximately 0.1 °C m² / W. The unit thermal resistance of the enhanced microchannel heat exchanger at the same flow and heat transfer conditions is 0.001 °C m² / W. There is a clear decrease in the thermal resistance and this would result in better thermal performance.

Finally, a new parameter called pumping power flux is developed to aid in the evaluation and comparison of enhanced microchannel heat exchangers. Using the new COP parameter based upon the heat flux and the pumping power flux, it can be concluded that the COP and pumping power flux are an appropriate metrics for comparing plain and enhanced microchannel heat exchangers. The maximum COP for the plain microchannels is 11 where the maximum for the enhanced microchannels is 290. Therefore, the enhanced microchannel heat exchanger is substantially more effective at dissipating the same heat flux despite the increase in pressure drop.

There is a great deal of research needed to bring these proposed techniques into fruition. In some cases, the technology might not be available. However, the present work is expected to serve as a road map for microchannel, minichannel, and microdevice heat transfer enhancement.

NOMENCLATURE

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<th>Symbol</th>
<th>Definition</th>
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<td>a</td>
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<td>A</td>
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<td>b</td>
<td>channel height, m</td>
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<td>B</td>
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<td>D_h</td>
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<td>uncertainty of parameter i</td>
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<tr>
<td>U</td>
<td>uncertainty</td>
</tr>
<tr>
<td>V</td>
<td>voltage, V</td>
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</table>

Greek

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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</thead>
<tbody>
<tr>
<td>α_c</td>
<td>channel aspect ratio = a / b</td>
</tr>
<tr>
<td>α_f</td>
<td>fin aspect ratio = s / b</td>
</tr>
<tr>
<td>Δ</td>
<td>change in</td>
</tr>
<tr>
<td>η</td>
<td>fin efficiency</td>
</tr>
<tr>
<td>ρ</td>
<td>density, kg m⁻³</td>
</tr>
<tr>
<td>σ</td>
<td>standard deviation</td>
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</table>

Subscripts

<table>
<thead>
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<th>Definition</th>
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</thead>
<tbody>
<tr>
<td>app</td>
<td>apparent</td>
</tr>
<tr>
<td>c</td>
<td>cross section</td>
</tr>
<tr>
<td>f</td>
<td>fin, fluid</td>
</tr>
</tbody>
</table>

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REFERENCES


