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Original Article

Enhanced convection in mini channel with CNT fins

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Abstract

Thermal performance can be improved by modifying the walls of a rectangular minichannel with micro-fins of bundles of ideal single-wall carbon nanotubes (SWCNTs) (circle, square, rectangle, triangle and hexagon) and using a nanofluid. A staggered array yields better thermal performance than an inline array as either a solid or porous medium. The thermal effects with SWCNTs were better than multiwalled carbon nanotubes. Fins as porous media for SWCNTs resulted in heat transfer that was 5% (maximum) greater than solid media. A staggered triangle array yielded better thermal performance as a solid (82%) or porous medium (92%) than an unfinned array. On a minichannel, with or without nanotubes and uniform coating, nanofluid increased the thermal performance with the best performance by CuO/H_2O. Thermal enhancement of 226% was obtained using staggered triangular fins with a larger fin height of 0.75 mm, smaller fin width of 0.5 mm, spacing the fins at double the fin width, and 0.01% CuO/H_2O nanofluid. The corresponding friction factor differential was found to be 625%.

Keywords: mini-channels, nanofluid, nanotube fin, pressure drop, thermal performance

1. Introduction

Nano-, micro-, and minichannels have a higher heat transfer surface area to fluid volume ratio. Although they have excellent cooling capabilities, minichannels experience a high pressure drop as the fluid flows which can cause problems when trying to re-circulate the fluid with a pump (Nara yanaswamy, Chandratilleke, & Foong, 2008). Several surface modifications had proven to enhance thermal performance even further. One modification is adding pin fins to increase the surface area (Tullius, Tullius, & Bayazitoglu, 2012). With fins of higher thermal conductivity, like carbon nanotubes, thermal resistance is decreased and temperature decreases (Morris, 2008). Measurements on a larger number of nanotubes resulted in thermal conductivity values as low as 250 W/mK for single-wall carbon nanotube (SWCNT) samples and 20 W/mK for multiwalled carbon nanotube (MWCNT) samples (Wright *et al.,* 2007). The addition of fins on the surfaces of the channel creates small disturbances within the fluid which allow fluid to mix and enhance heat transfer (Shokouhmand, Aghvami, & Afshin, 2008). The effects of varying the thermal conductivity of the fin structures on the surface of a micro channel were already reported (Zhong *et al.*, 2007). A slight increase in heat transfer performance with little or no increase in pressure drop was obtained for a micro channel with silicon fins (Lee, Lee, & Chou, 2010). Minichannels with cylindrical fins of lower density provided the best thermal performance in the laminar flow; however, the effects of the increased pressure drop greatly outweighed the increase of heat removed (Selvarasu, Tafti, & Blackwell, 2010). With fins, about 60% of the channel height yields less resistance and improves cooling performance (Min, Jang, & Kim, 2004). The existing conventional correlations could not accurately capture the microscale interaction of the fluid and the fins by investigating the effects of pin fins for pressure drop (Koşar, Schneider, & Peles, 2011).

Assembling the carbon nanotubes (CNTs) as fins together decreases the thermal conductivity by 400 W/mK (Wright *et al.*, 2007). Optimizing the fin shapes, topology, and dimensions to achieve the maximum thermal performance has not been thoroughly executed with corresponding heat transfer and friction factor correlations. A flow-boiling analysis of CNTs with coated microchannels using water was conducted

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(Singh, Sathyamurthy, Peterson, Arendt, & Banerjee, 2010). Others investigated single-phase flow using MWCNTs on the surfaces of micro channels. Different heat rates applied to the silicon microchannel and keeping the pressure drop constant obtained a 23% higher input power and the temperature of the transistor was lower (Mo, Morjan, Anderson, Campbell, & Liu, 2005). The effects of water flowing past a fully coated carpet of MWCNTs on the surface and circular MWCNT micro-pin-fins in a staggered array was compared to a bare minichannel (Shenoy, Tullius & Bayazitoglu, 2011). With silver nanofluids suspended in a polyvinyl pyrrolidone solution through drop shaped micro-pin-fins in a micro-channel obtained an 18% increase in thermal performance with little difference in the pressure drop (Zhou, Xia, Chai, Li, & Zhou, 2012).

Since the high surface volume ratio and small surface defects affect the domain in these channels, the experimental work (Tullius & Bayazitoglu, 2013) was extended to include the SWCNT fins instead of MWCNT developed from fully carpeted rectangular mesh and laser cut to the desired fin shape, formation, and size. Deionized water was used instead of a nanofluid to easily obtain a numerical solution. Thermo-hydrodynamic performance of hydrodynamically and thermally developing single-phase flow in rectangular minichannels was experimentally investigated (Agarwal, Moharana, & Khandekar, 2010) and the Nusselt number (Nu) and dimensionless friction factors obtained were 6.5-14 and 0.441-0.258, respectively, at Reynold's numbers (Re) 500-2000 and the laminar-to-turbulent transition was found to occur at Re 1100. From the published literature, it can be seen that there is a lot of experimental and numerical data available on the use of rough wall surfaces. However, there is a scarcity of literature available for CNT fins in the turbulent flow regime inside a rectangular minichannel. This study was carried out to see whether differently shaped CNT fins can enhance heat transfer and thermal performance for turbulent nanofluid flows in a rectangular minichannel for different array geometries using computational fluid dynamics.

2. Device Geometry and Computational Modeling

Ten device geometries (in line and staggered circle, square, rectangle, triangle and hexagon) were developed and used for this simulation. A 1 mm thick rectangular mini-channel made of silicon with a size of 45x15 mm and SWCNTs with a broad diameter distribution of 10-100 nm were used. For full coverage, nanotubes with a diameter of 1 mm and 0.5 mm in height were grown at the center of a silicon wafer at the bottom in an area of 25x15 mm^2 as done with all micro-pin-fin arrangements (Figure 1).

For these simulations, the flow regime is considered to be a continuum flow. The basic governing equations for a steady-state, incompressible flow are:

Continuity equations

$$(\nabla \cdot \rho u) = 0 \tag{1}$$

Conservation of momentum

$$\rho(u \cdot \nabla u) = -\nabla p + \mu \nabla^2 u \tag{2}$$

Conservation of energy

$$\rho c_p(u \cdot \nabla T) = \nabla^2 T \tag{3}$$

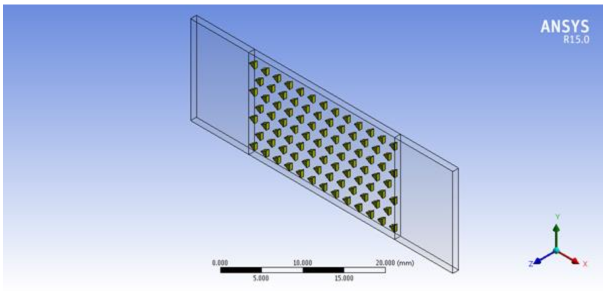

Figure 1. Minichannel model with triangular CNT pin fins in a staggered array.

The geometry is meshed based on the finite volume method. Ansys FLUENT 15.0 was used to model the flow past the pin fins in a staggered array to solve the governing equations iteratively for each control volume. The convergence criterion for the solution was residuals of less than 10^{-6} for both continuity and momentum. Each model contained a fine mesh surrounding the finned section with a less dense mesh as the channel extends to the inlet and outlet. Micro-pin-fins using multiple flow rates and heat fluxes were optimized by varying the fin height, fin width, and spacing and applying multiple material properties to the pin finned geometry. Hydrodynamic stabilized flow was developed before the fluid reaches the heated region.

The transverse and longitudinal spacing of the fins for this study was equal to double the width of the fin and the fin height was half the length of the channel height or 0.5 mm. For all geometries, the fin material was SWCNT; however, because the thermal conductivity is unknown when nanotubes are clustered together to form fins, an effective thermal conductivity of 400 W/mK was used (Zhong *et al.*, 2007). Using the various geometries and SWCNT material properties, the channel clearance was varied using fin to channel height ratios of 0.25-0.75 but the spacing was kept equal to double the fin width. The height of the fins to the channel height was kept at half the channel clearance and the number of fins along each row was varied based on the width and the spacing of the fins. Fin width varied from 0.5 to 2.5 mm. Once the optimum size of the fins was determined, the spacing was varied using the fin's width of 1.25 mm. Unlike the previous spacing that was the same as the fin width, the spacing was reduced to 2 mm and 2.25 mm while maintaining the same number of fins in the row. The same pin fin geometry similar to the first study was used with a fin width of 1 mm, spacing of 2 mm, and the fin height was half of the channel height. In addition to the CNTs, silicon, copper, and aluminum were also used as the fin material.

Initial inlet temperature and outlet static pressure values applied to the model were assumed for all simulations to be 25 °C and 0 Pa, respectively. To monitor the heat transfer coefficient and the heat transfer rate, the outer walls of the channel were set to be adiabatic. No-slip boundary conditions and no interfacial resistance were assumed. These simulations were in the single phase regime and the fluid properties were kept constant throughout the simulations. Water that flowed past the pin fins were subjected to heat by the bottom surface. A constant heat flux of 100 W/m^2 was applied to a 15x25 mm^2 area at the bottom of the channel. With constant heat flux, different geometries of nanotube bundles and forced convection, the temperatures across the surface of the microchip and bulk temperature of the fluid varied.

To accurately obtain the heat transfer coefficient (h) across the microchip region for simulations, an average h was obtained. As the scales start moving toward micro- or nano-regime, conventional continuum calculations can no longer be used. Also, resistance created at the interface of CNTs and the working fluid is still a major issue. The SWCNT fins were modeled as a solid emerging out from the surface rather than bundles of nanotubes. SWCNT micro-pin-fins were made up of many nanotubes where the fluid penetrates through small gaps (nanochannels) allowing for an increase in convection within the system. As shown in (Shenoy *et al.,* 2011), SWCNTs absorb the fluid at high temperatures creating a porous like material which increases the heat transfer due to a hastened nucleate boiling onset initiating phase change. This assumption in the model may underestimate the thermal performance of SWCNT solid fins. Hence, the best performance case is simulated using a porous medium. SWCNT and MWCNT materials were also used in this context for comparison of thermal performance. The influence of Al_2O_3/H_2O and CuO/H_2O nanofluids instead of water on the thermal performance in comparison with cases of no nanotubes and with nanotube fins were also investigated.

The averages of Re and Nu were collected for the different inputted heat fluxes and flow rates. The average axial velocity was obtained as the fluid flowed across the fin bank.

$$Re = \frac{\rho u D_c}{\mu} \tag{4}$$

Nu is proportional to the average heat transfer coefficient (h) and the hydraulic diameter of the channel and non-proportional to the thermal conductivity (k).

$$Nu = \frac{\boldsymbol{h} D_c}{k} \tag{5}$$

The average heat transfer coefficient is obtained by the amount of surface area of the fins and the base that the fluid interacts with. The expression is given as

$$\boldsymbol{h} = \frac{\left(\boldsymbol{h}_{fin} A_{fin} + \boldsymbol{h}_b A_b\right)}{\left(A_{fin} + A_b\right)} \tag{6}$$

Where h_{fin} and h_b are the heat transfer coefficients of the fin and the base of the heated region only, and A_{fin} and A_b are the surface areas in which the fluid touches the fin and the heated base region, respectively. The heat transfer coefficient of the fin and the base at the fluid interface is obtained by

$$\boldsymbol{h} = \frac{q}{T_w - T_{nw}} \tag{7}$$

Where q is the wall heat flux, T_w is the temperature of the wall, and T_{nw} is the near wall temperature. The friction factor is determined across the pin fins only to avoid any entrance and exit effects of the fluid flow using a maximum axial velocity.

$$f = \frac{2\Delta p \mathrm{D_c}}{\rho l u^2} \tag{8}$$

3. Results and Discussion

3.1. Different shaped micro-fins

The results of different finned shaped microstructures of inline configuration on the surfaces of minichannels were compared to a smooth, unfinned channel. With increasing Re, the Nu values increased for all minichannel simulations. The Nu for the triangle and square shaped fins had similar values with a maximum average Nu difference of 1.4% for higher Re. For this data, triangular shaped fins showed better thermal performance. The circle shaped portrayed a thermal performance which was less effective than the triangle shaped fins by 10%, and 3% and 4% less effective than the hexagon and rectangle shaped fins, respectively. However, they were still more efficient than the unfinned channel. For lower Re, the Nu increase was very high compared to the unfinned channel and at higher Re the gap decreased to have a 59% performance enhancement for the circular shaped fins. Figure 2 reflects the results of this study.

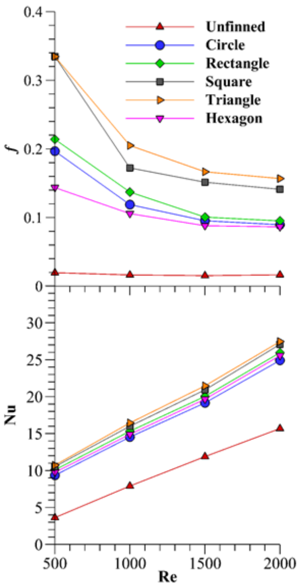

Figure 2. Fin geometry: Nu and friction factor vs. Re.

With the addition of fins, there was an increase in the friction factor. A value that is too low can create problems for the pumps to flow fluid through the minichannel. As the Re increased from Re 500 to 2000, the friction factor decreased for all models up to 113%. The triangle pin fin had the highest friction factor with a value of 0.335 followed by the next shape which was the square shape. This increase was much higher than the friction factor (0.02) of the unfinned channel and the lowest recorded hexagon and circle shapes yielded a maximum friction factor of 0.144 for the same Re. The shapes that were more aerodynamic revealed a lower friction factor because there was less separation of the fluid from the solid body. The opening between the fins disrupts the momentum and the trailing edge of the thermal boundary layer of each oblique fin. This causes the leading edge to re-develop allowing the flow to remain in the developing state. The addition of fins showed a big impact on the performance of the heat exchanger.

3.2 Different micro-fin height

Figure 3 displays the results for different channel clearances. With increasing fin height, the thermal performance increased with increasing Re. From the lowest fin height at 0.25 mm, compared to the unfinned channel, the Nu increase was very high at a low Re and the gap decreased to 30% for a high Re. From the largest channel clearance of 0.75 mm to the smallest of 0.25 mm, the Nu increase was 23% (Figure 3).

As the fin heights increased, the friction factor drastically increased 33-fold. The friction factor from the largest channel clearance was 0.68 and the smallest channel clearance decreased to 0.209. Since there was a large friction factor difference for the increased height and the small Nu relative increase, the suggested fin height was dependent on the available pump parameters used within the cooling apparatus.

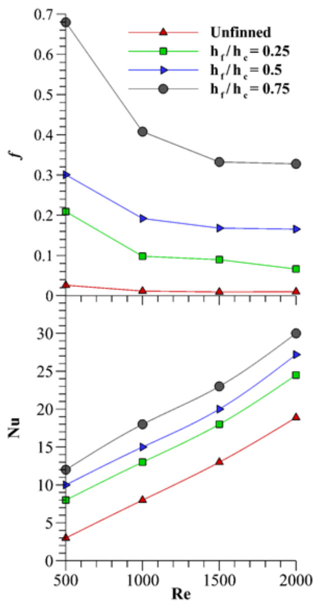

Figure 3. Ratios of fin height to channel height: Nu and friction factor vs. Re.

3.3 Different micro-fin width and spacing

With smaller fin width and spacing, i.e. higher number of fins, the Nu value was greater than the other fins. Fins with a width of 0.5 mm resulted in a 97% higher Nu value than the unfinned channel and a 40% higher Nu value than the lowest curve of fin width and spacing at 2.5 mm. Fins with a width of 2.5 mm still yielded a higher Nu value of 41% greater than the unfinned channel with a higher Re (Figure 4). While the smaller fin width and spacing provided higher performance, the friction factor across the fins was sacrificed. As the fin dimensions reduced, the friction factor increased. Fins with a width of 0.5 mm and spacing of 1 mm had a friction factor that was 316% greater compared to the bigger fin width and spacing of 2.5 mm and 5 mm which had a friction factor that was 41% greater than the unfinned channel (Figure 4). The bigger fins resulted in a smaller friction factor across the channel. This is probably due to fewer disturbances in the fluid within the channel. It is interesting to note that for both the friction factor and the Nu, the fins with a width of 0.5 mm and 0.75 mm were very similar. The optimum fin width was between these two widths. Similar to the other conclusions, specification of the pump parameters should be taken into account on the number of fins the system can have before the friction factor is too high. As the spacing was closer to equal the fin width, the performance was greater (Figure 4). From the smallest spacing to the largest, i.e. 2 mm to 2.5 mm, there was a 3% thermal enhancement for larger fin spacing. The friction factor increased as the fin spacing decreased. There was a 25% increase in the friction factor from the 2.5 mm spacing to 2 mm. From the results, it can be concluded that the decrease in width/spacing creates a higher thermal performance with a sacrifice in friction factor. By only changing the spacing, the maximum spacing yielded a higher thermal performance and provided a minimal friction factor.

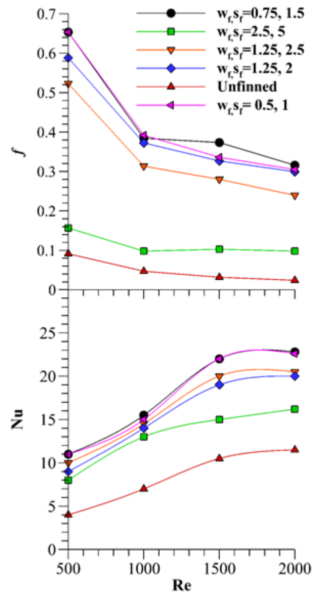

Figure 4. Fin width and fin spacing: Nu and friction factor vs. Re.

3.4 Different micro-fin material properties

For single phase flows, changing the material properties showed little variation within the Nu values, yet, they were still larger than the unfinned channel (Figure 5). These findings showed that the chosen fin material had little effect on the Nu values. In reality, the fins had a porous characteristic that had small gaps where the fluid could penetrate. The nanotubes could also initiate nucleation sites initiating boiling and therefore enhance heat transfer. Because of this, the results for the CNT fins were underestimated using this program. From this conclusion, the CNTs should yield a greater thermal performance than the other fins. The friction factors were not different between the fin materials because the geometry remained the same in this part of the study. The friction factor in this study was shown to be critical in optimizing the fin geometry and topology. For a single-phase, laminar flow through a minichannel, Nu values for pin fins did show improvement compared to the unfinned channel; however, various shapes, heights, widths, spacing, and thermal properties showed only slight deviations.

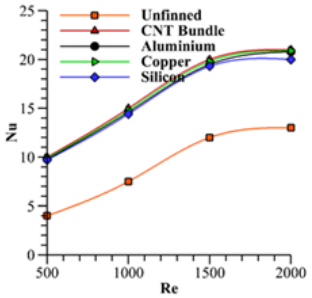

Figure 5. Nu vs. Re for selected fin materials.

3.5 Effects of different arrays, media, walls of CNT, uniform coating, and nanofluids

For the same Re, devices with the MWNTs performed better than a device without MWNT. Similar to the results by Shenoy *et al.*, (2011), for a finned device and a fully covered device, the surface area was much higher than a device without MWNTs; therefore, more heat was removed. The device without MWNTs had a thermal increase of 37% using a 0.01% volume concentration of Al_2O_3 nanofluid with respect to the same device using deionized water (Figure 6). The fully covered and finned MWNT devices had increases of 45% and 135%, respectively, at the same base temperature compared to the channel without MWNTs using Al_2O_3 nanofluid (Figure 6). These channels already had surface defects in the form of MWNTs; therefore, only a slight increase was observed probably due to the presence of the Brownian motion and nanoconvection in which the nano-particles take part.

There was a 3% increase in thermal performance with a channel containing SWCNTs than MWCNTs and a 2% increase in thermal performance when a porous medium is used instead of a solid medium in channels with triangular inline fins. Similarly, there was a 4% increase in thermal performance for staggered triangular fins in place of inline configuration and a 1% increase when CuO nanofluid was used in place of Al_2O_3 nanofluid. The friction factor obtained using the nanofluid for each channel showed higher values with the addition of nanoparticles. Staggered and SWCNTs showed a similar pattern in comparison to inline and MWCNTs, respectively. Similar to the results for water, a fully covered MWCNT device caused higher friction factors compared to MWCNTs and MWCNT without finned devices. This is due to the difference in hydraulic diameters created by the protruding MWCNTs on the surface. Lower friction factors were also noticed for CuO nanofluid and porous medium in comparison to Al_2O_3 nanofluid and solid medium respectively. This study verifies that the major enhancement when using nanofluids to cool heated surfaces is the surface defects that are deposited on the surface. As stated by other researchers (Ahn & Kim, 2012), these particles create imperfections on the surface causing increased wettability. The channels which already contain engineered structures increase the surface area and wettability and the thermal performance is not significantly improved.

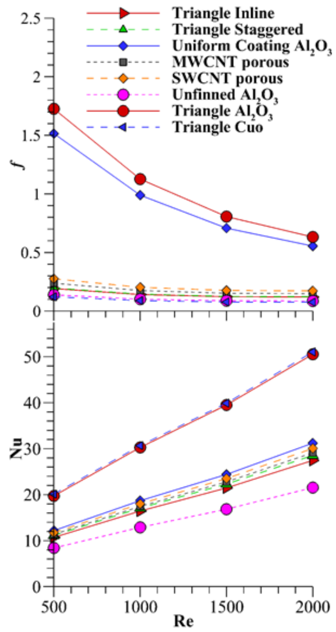

Figure 6. Effects of different properties, materials, and configurations.

4. Validation

Experimental results were taken from the work of Liu, Liu, Xu, & Chen (2011) for a heat sink containing 625 square micro-pin-fins of 445x445 μm^2 in a staggered array. Water was used as the working fluid flowing with Re 60–800. The fins and channel had a height of 3 mm. The longitudinal and transverse spacing of the fins was 565.7 µm. The ambient fluid temperature was initially at room temperature and heat was applied to the bottom surface. The Nusselt number and friction factor across the finned structure were calculated and compared to the experimental data and were found to be very similar to the values obtained in the experimental work for Nu values and friction factor (Figure 7) except for Re 100 and 500 (for friction factor alone) within a 21% thermal differential. This proved that the present study can accurately predict the heat transfer through the channel with micro-pin-fins on the surface. The nanofluid volumetric concentration of the particles in this case was 0.01.

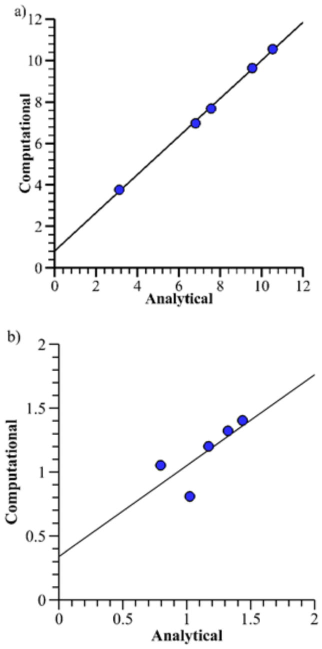

Figure 7. Validation of simulated data of staggered square pin fins with experimental data at different Re: a) Nu, b) friction factor.

5. Conclusions

The use of microstructures on the surfaces of minichannels was effective with increased heat removal from a heated surface. It was observed that the presence of SWCNTs and MWCNTs resulted in enhanced heat removal for a given volumetric flow rate. Nu values of fin shapes were within 10% of each other at high Re and the triangle fins registered the highest and the circle fins the lowest; however, the friction factor values were also greatly affected in the same fashion. The lowest Nu value of the circle fin was larger than 59% compared to the unfinned channel. With increased fin height the Nu increased; however, the friction factor also increased drastically. From the lowest tested fin height to channel ratio to the highest, the difference in Nu was 23% and the friction factor differential at a higher Re was 0.262. As the fin width and spacing decreased the Nu increased; however, the friction factor increased significantly. For a fin width of 0.5 mm, there was an increase in heat transfer performance that was 97% higher than the unfinned channel and the Nu values were 40% higher than the larger fin widths and spacings. But, the friction factor increased significantly (611%) compared with the unfinned channel. When the spacing was made almost equal to the fin width, the thermal performance was better by 14% with a minimal friction factor of 12%. Fin material showed little effect on fin performance. Fins with SWCNTs, when modeled as a porous media, increased the heat transfer performance significantly with the highest value of 5% for SWCNTs compared with a solid medium. Staggered triangle array yielded better thermal performance both as a solid (82%) and porous medium (92%), compared with the unfinned channel and there was a 99% increase in the amount of heat transfer when fully covered nanotubes were used. It was observed that on a minichannel surface with nanotube fins, nanofluids increased the thermal performance in the channels with nanotube fins with the best observed by CuO/H_2O nanofluid (226%) with respect to no nanotubes. The friction factor increased significantly with a 625% increase to the unfinned channel.

Nomenclature

A_b, A_{fin}	Area of base, area of fin
D_c	Hydraulic diameter of channel
f	Friction factor
h, h_b, h_{fin}	Heat Transfer Coefficient, HTC of base, HTC of fin
h_c, h_f	Height of channel, height of fin
l	Length of channel
S_f	Spacing of fins
W_f	Width of fins

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