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EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER IN A NOVEL HEAT SINK BY MEANS OF ALUMINA NANOFLUIDS

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In this paper, heat transfer characteristics of a miniature heat sink cooled by Al_2O_3–water nanofluids were investigated experimentally. Based on plate fin heat sinks, a new type of plate pin-finnned heat sink is developed which is composed of a plate fin heat sink and columnar pins between the plate fins. The heat sink was fabricated from aluminum and insulated by plexiglass cover plates and consisted of five pin-finned rectangular channels with a length of 42 mm. The volume fraction of the Al_2O_3–water nanofluid particles was in the range from 0.5 to 2%. Mixtures were prepared without a dispersion agent. Tests were performed while supplying a 180 W/cm^2 heat flux to the bottom of heat sink. Experimental results showed that dispersion of Al_2O_3 nanoparticles in water increased significantly the overall heat transfer coefficient, while the thermal resistance of heat sink decreased. Also, the plate pin-finned heat sink showed an increase in the heat transfer coefficient up to 20% in comparison with the conventional plate fin heat sink.

KEY WORDS: *heat transfer, experimental, Al_2O_3–water nanofluid, novel miniature heat sink, constant heat flux*

1. INTRODUCTION

During the last two decades, there has been a monumental thorough search for more efficient cooling equipment for electronics and for more compact and reliable thermal management systems. Among different available electronic cooling devices, plate fin heat sinks are the most widely used because of several major advantages such as their simple design and low manufacturing costs. But there are cases where we need to sacrifice simplicity of heat transfer devices for their higher efficiency. To attain this goal, slight but efficacious modifications should be made,

NOMENCLATURE

A	area	T_i	inlet coolant temperature
D_h	hydraulic diameter	W	width of a heat sink
H	heat transfer coefficient, W/(m^2·K)	**Greek symbols**	
H	channel height	μ	dynamic viscosity, Pa·s
k	thermal conductivity	θ	thermal resistance
L	length of a heat sink	Φ	volume fraction
$\dot{m}$	mass flow rate	**Subscripts**	
N	number of pins	Avg	average
q''	heat flux over the bottom surface	m	mean
Q	volume flow rate	nf	nanofluid
r	pin radius	w	wall
t	channel width	**Nondimensional number**	
T	temperature	Re	Reynolds number, $= \dot{m}D_h/\mu_{nf}\overline{A}$

such as adding some pins to the currently available plate fin heat sinks. This paper investigates the effect of adding circular pins to the base plate of a plate fin heat sink by using nanofluids.

In recent years, nanofluids have been proved to be an ideal candidate for enhancing heat transfer (Choi, 1995; Lee et al., 1999). Various studies have been conducted on the performance of convective heat transfer of nanofluids (Wen and Ding; Heris and Etemad, 2006; Hwang et al., 2009). They have concluded that nanofluids provide heat transfer enhancement in comparison with their corresponding base fluids. Mansour et al. (2011) studied the mixed convection of a water–Al_2O_3 mixture inside an inclined tube and concluded that higher volume fractions of particles clearly induced a decrease in the Nusselt number in the horizontal position. Ho et al. (2010) conducted experiments to investigate forced convective cooling performance of a copper plate fin microchannel heat sink with the Al_2O_3–water nanofluid as a coolant. Their results showed that the nanofluid-cooled heat sink outperforms a water-cooled one, having a significantly higher average heat transfer coefficient and thereby markedly lower thermal resistance and wall temperature. Pantzali et al. (2009) performed experimental and numerical analyses of the effect produced by using CuO–water nanofluids in a miniature plate heat exchanger (PHE) with a modulated surface. In their study, heat transfer enhancement was more pronounced at lower cooling liquid flow rates. More recently, Zhou et al. (2012) investigated experimentally the convective heat transfer and friction characteristics of a silver nanofluid in a micro-pin-finned heat sink. They stated that the volume fraction of silver nanoparticles significantly affected the convection heat transfer coefficient of the micro-pin-finned heat sink, and the thermal resistance of the nanofluid was lower than that of deionized water. Also, Duangthongsuk et al. (2012) presented an experimental study on the heat transfer and pressure drop characteristics of 1.0, 2.0, and 3.0 wt.% Al_2O_3–water nanofluids flowing through an aluminum rectangular microchannel heat sink (MCHS). The results indicated that the heat transfer performance of MCHS increased with increasing Reynolds number as well as particle concentrations. They reached a maximal 15%-increase in the heat transfer coefficient using nanofluids. In another effort, Ijam et al. (2012) studied a laminar flow of Al_2O_3–water and TiO_2–water nanofluids with different volume fractions of nanoparticles as coolants for a copper minichannel heat sink. The result showed that the adding of Al_2O_3 and TiO_2 nanoparticles to water at a volume fraction of 4% enhanced the thermal conductivity by 11.98% and 9.97%, respectively. Finally, Selvakumar and Suresh (2012) prepared CuO/water nanofluids with volume fractions of 0.1% and 0.2%. They used a thin channeled copper water block of overall dimension $55 \times 55 \times 19$ mm for their study. The convective heat transfer coefficient of the water block was found to increase with the volume flow rate and nanoparticle volume fraction, and the maximum rise (29.63%) in the convective heat transfer coefficient was observed for the 0.2% volume fraction compared to deionized water. In their work, a correlation was proposed for Nusselt number which fits the experimental Nusselt number within 7.5%.

To the knowledge of the present authors, there has not been any previous work concerning the cooling capability of alumina–water nanofluids in a plate pin-finned heat sink, especially with lower particle volume fractions. Thus, the main objective of the present work was to analyze experimentally the performance of Al_2O_3–water nanofluids at small particle volume concentrations for the purpose of heat dissipation in a novel heat sink and to compare the results with the traditional plate fin heat sink.

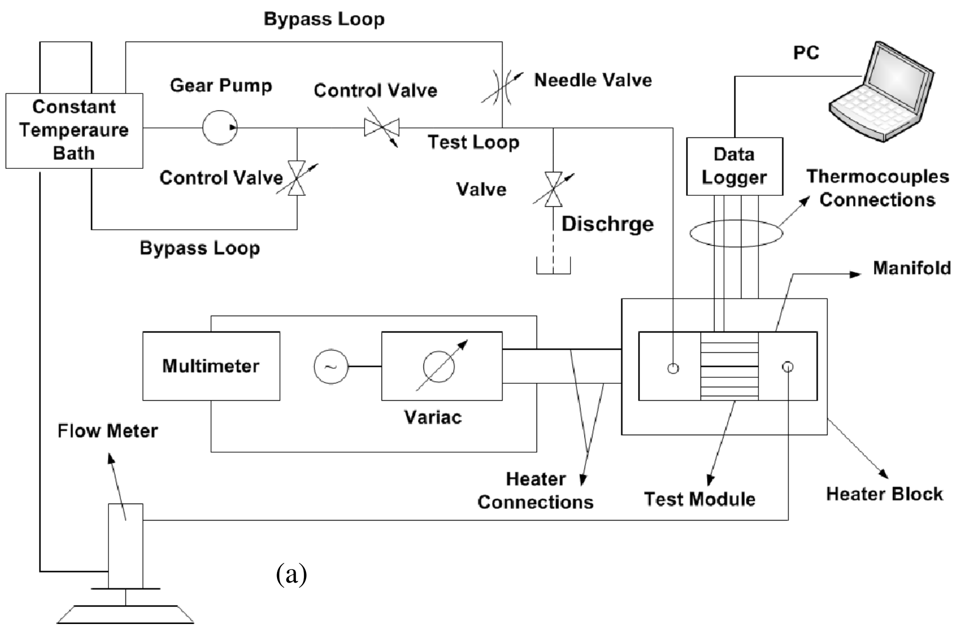
2. PREPARATION AND PROPERTIES OF NANOFLUIDS

Al_2O_3 spherical nanoparticles (purchased from Wacker, Germany) with an averaged particle size of 18 nm and 99.9% purity were dispersed in distilled water, as the base fluid, to form Al_2O_3–water nanofluids. The nanofluids were synthesized by the two-step method, without any surfactant in order not to affect the viscosity and thermal conductivity (k) of suspensions. The desired volume fractions of alumina–water nanofluids were prepared by mixing appropriate quantities of nanoparticles with the base fluid, and then sonicated in an ultrasonic bath (Hielscher UP400S, H40sonotrode) for at least 90 min. The nanofluids used in the current study stayed stable for a period of 72 h without any visible settlement. Four volume fractions of the nanofluids, Φ = 0.5, 1, 1.5, and 2 vol.%, were prepared for the experiment.

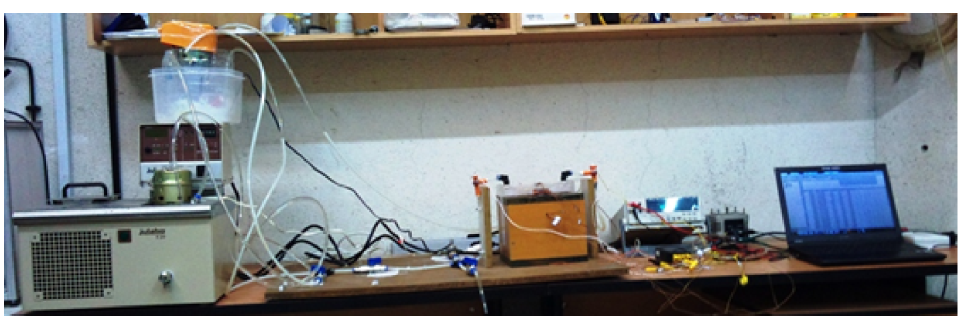
3. EXPERIMENTAL APPARATUS

Figure 1 demonstrates the flow loop and components that were designed and constructed for the present study. The main components of the test apparatus are: a closed loop for circulating the fluid, heat sink test section, and a data acquisition system. A fluid is sent into the loop from a holding tank and is continuously circulated by a gear pump which can be operated at variable rotation speeds to supply different flow rates.

A constant temperature bath (F10-Hc Julabo) was installed upstream of the gear pump to control the heat sink inlet temperature. The loop flow rate is controlled



(a)



(b)

FIG. 1: Schematic of the experimental setup (a) and photo of the experimental setup (b)

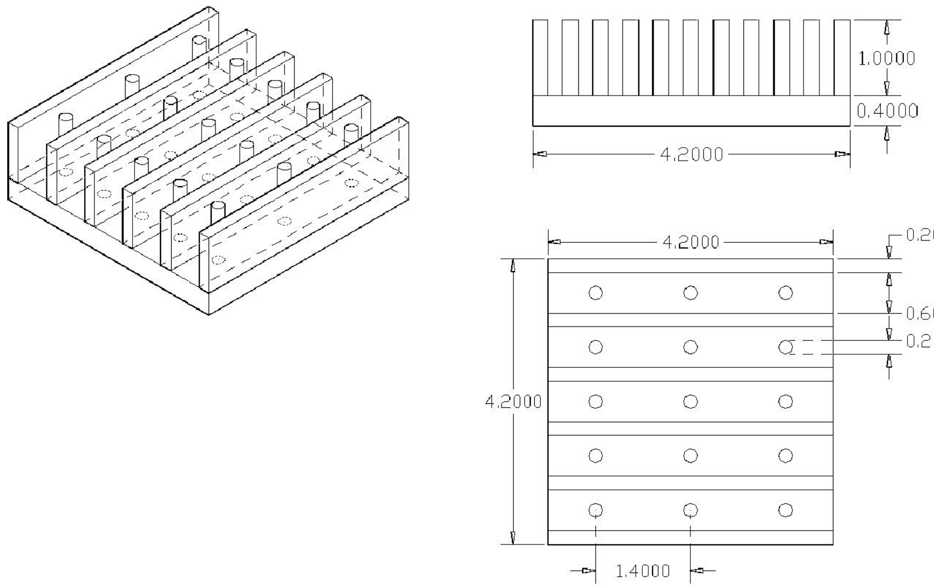


FIG. 2: Heat sink dimensions (cm)

as the fluid leaves the pump through two by-pass lines with the use of three ball valves for coarse adjustment and a needle valve for fine adjustment of the flow rate. The volumetric flow rate in the loop was measured using a calibrated flow meter. The test module consists mainly of a miniature channel heat sink, housing, plexiglass cover plates as insulation, and a heater. The miniature channel plate pin-finned heat sink was fabricated from a square block of aluminum of size 42 mm $\times$ 42 mm $\times$ 14 mm, using a CNC machine. The channels have a rectangular shape with three pin fins of 2 mm diameter implanted in every channel. The miniature heat sink is shown in Fig. 2. The dimensions of the plate fin heat sink are the same, but there are no pins.

The heat sink assembly was placed inside a plexiglass shroud isolated from the ambient. Because of the low thermal conductivity of plexiglass, the effect of lateral heat transfer from the sides of the test section is eliminated. Also, for ensuring that the amount of heat loss is small and does not affect the results to a high extent, the surface temperature of plexiglass box sides were measured at different spots, and the results showed that the temperatures of the plexiglass sides were adequately equivalent to the ambient temperature, showing a maximum of 0.3^{o}C temperature difference occurring when the plexiglass was insulated by the lateral sides of the heat sink. All the thermocouples used in this study were calibrated. Four K-type thermocouples, coated with a compound of copper powder and thermal paste, were embedded in the heat sink for measuring the base plate temperature under the fins. The exact longitudinal distance of the thermocouples from the heat sink front, along

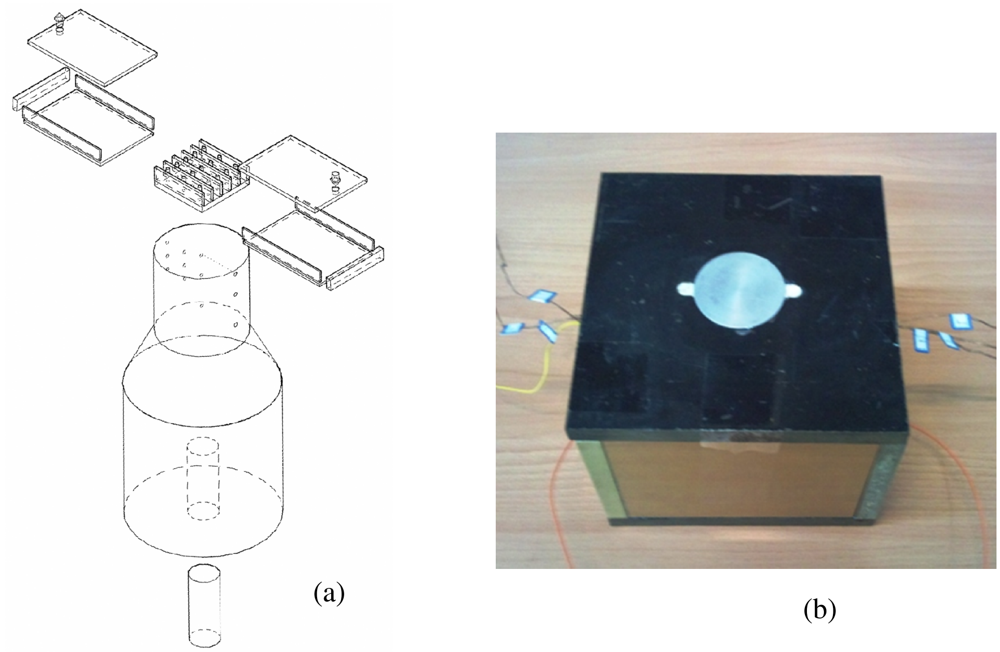


FIG. 3: Geometric configuration of the heater block (a) and a real fabricated heater block (b)

the heat sink base and under the third fin were 10 mm, 21 mm, 32 mm, and 40 mm. Also located in the inlet and outlet of the manifold were two K-type thermocouples inserted to measure the inlet and exit temperatures of the fluid.

In order to provide a constant heat flux surface for simulating an electronic chip, a heater block of the shape shown in Fig. 3 (Wei, 2004; Dang, 2006), was fabricated from the same material as that of the heat sink. Six holes for thermocouples were made in the body of the heater block to measure temperatures required for calculating the heat flux supplied to the heat sink. The heater block was completely insulated by using a slag wool cover all around its body and a fiberboard box. The set of thermocouples were connected to two Testo 177-T4 data loggers through an eight channel selector.

The uncertainty of the experimental data may originate from the errors of measuring such quantities as the heat flux or temperature. The uncertainty of the heat transfer coefficient can be calculated from the following equation:

$$\frac{\Delta h_{ave}}{h_{ave}} = \frac{\Delta q''}{q''} + \frac{\Delta t + \Delta H}{t + H + \dfrac{\pi r H}{L} - \pi N r^2} + \frac{\pi\left(H\Delta r + r\Delta H - rH\Delta L\right)}{\left(t + H + \dfrac{\pi r H}{L} - \pi N r^2\right)}$$
$$- \frac{2\pi N r \Delta r}{t + H + \dfrac{\pi r H}{L} - \pi N r^2} - \frac{\Delta W}{W} - \frac{\Delta\left(T_w - T_m\right)}{T_w - T_m}, \tag{1}$$

TABLE 1: Uncertainty of measurements

Quantity	Uncertainty
Heat flux, W/m^2	±3.3%
Temperature, ^{o}C	±0.1^{o}C
Heat transfer coefficient, W/m^2·K	±6.9%

The uncertainties of measurements in the present study are listed in Table 1.

4. EXPERIMENTAL RESULTS AND DISCUSSION

The heat transfer coefficient for a plate pin-finned heat sink can be obtained from the following equation:

$$h_{ave} = \frac{2q''(t + H + \pi rH / L - \pi Nr^2) / W}{(T_w - T_m)}, \tag{2}$$

where N is the number of pins.

There are two different ways of defining the Reynolds number in pinned channels, i.e., one based on the pin diameter and the other based on the hydraulic diameter. In this work, we used the second approach:

$$\mathrm{Re} = \frac{\dot{m}\, D_\mathrm{h}}{\mu_\mathrm{nf}\bar{A}}, \tag{3}$$

where $\bar{A}$ is the ratio of the fluid volume to the channels length and D_h is the hydraulic diameter.

The heat flux supplied to the bottom of the heat sink was measured by averaging four heat fluxes obtained from measuring six temperatures in the body of the heater block:

$$q'' = \frac{q''_{12} + q''_{23} + q''_{45} + q''_{56}}{4}. \tag{4}$$

The volume fraction of particles in the nanofluid used in this study was in the range of 0.5–2%. The flow rate and inlet temperature for both pure water and nanofluids were the same. The flow rates (Q) chosen for this study were 8, 12, 16, and 20 cc/s. The inlet temperature (T_i) was fixed at 20^{o}C. The comparison between the performances of water and nanofluids as coolants is described below.

Figures 4 and 5 depict the effect of the concentration of nanoparticles on the temperature difference between two ends of two types of heat sinks base plates, when the flow rate was adjusted to 12 cc/s. As can be seen, the maximum temperature of the plate pin-finned heat sink is considerably lower than that of the

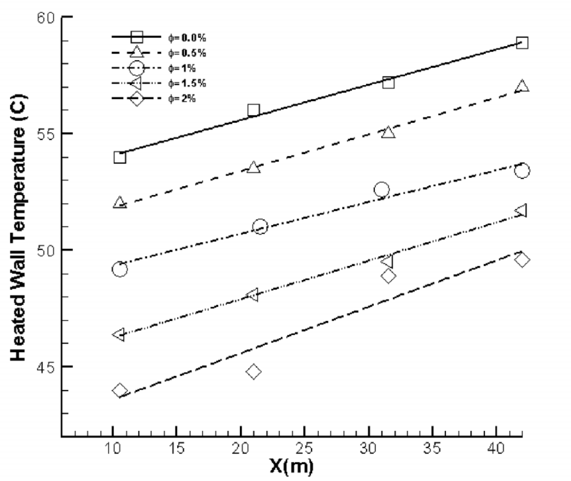


FIG. 4: Base plate temperature variation in a flat plate heat sink

plate fin heat sink. This difference can be very important for cooling highly sensitive processors where even one degree of temperature augmentation can result in their switching-off.

Figures 6 and 7 show the convective heat transfer coefficient for both types of heat sink as a function of the Reynolds number for pure water and Al_2O_3–water nanofluids at different volume factions of particles in a laminar flow. In the plate fin heat sink, as the Reynolds number varied from 200 to 1800, the heat transfer coefficients of the nanofluids with the volume fractions of 0.5%, 1%, 1.5%, and

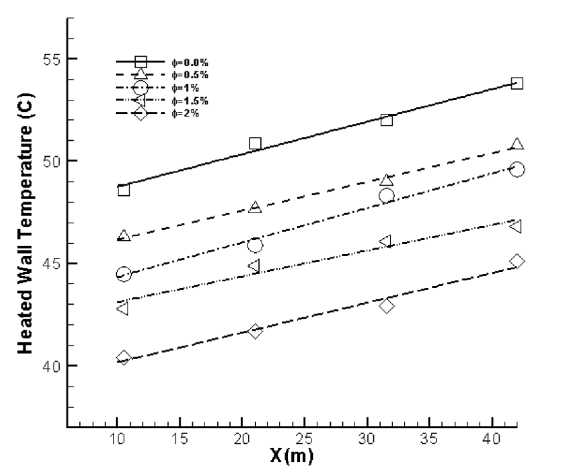


FIG. 5: Base plate temperature variation in a plate pin-finned heat sink

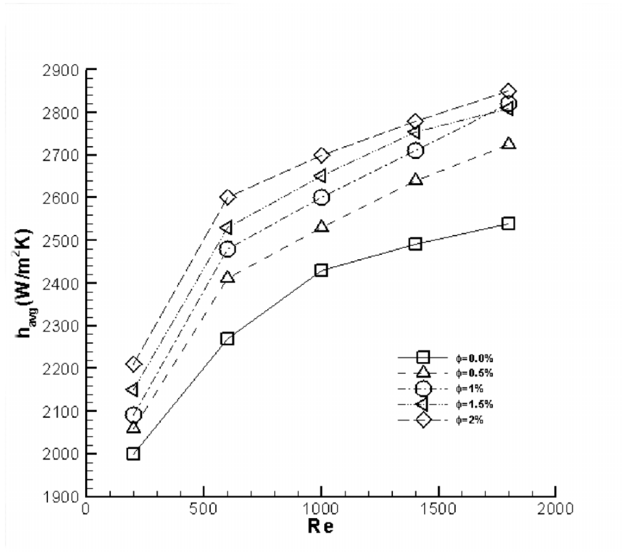


FIG. 6: Heat transfer coefficient enhancement in a plate fin heat sink with the use of the Al_2O_3–water nanofluid

2% increased up to 16% as compared with that of pure water, respectively. Also, when using a plate pin-finned heat sink, the heat transfer coefficients were up to 20% higher than those for a plate fin heat sink in similar conditions. One more important point is the thermal entry length. At the Reynolds number of about 600, there is a slight change in the slope of the heat transfer coefficient curve. This change can be attributed to the disappearance of a thermodynamically fully devel-

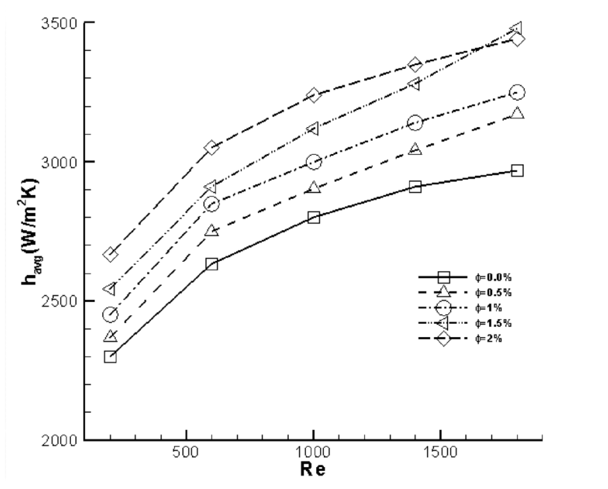


FIG. 7: Heat transfer coefficient enhancement in a plate pin-finned heat sink with the use of the Al_2O_3–water nanofluid

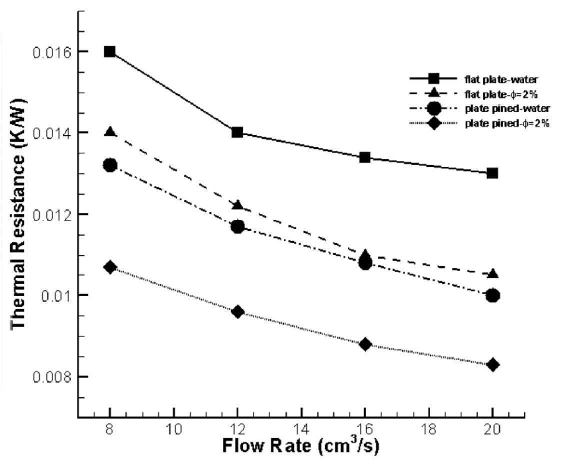


FIG. 8: Thermal resistance as a function of flow rate

oped (TFD) flow. At Reynolds numbers below 600, a TFD flow occurs inside the heat sink; as we increase the flow rate and proportionally the Reynolds number, the occurrence of the TFD point is delayed inside the heat sink itself. As a consequence, the flow undergoes a high change in the thermal boundary layer, and therefore a sharp slope exists; but when the Reynolds number exceeds 600, the flow does not become fully developed in the heat sink, and with increase in the flow rate, there occurs only a slight change in the thermal boundary layer and heat transfer coefficient of the heat sink.

The cooling performance of a heat sink with nanofluids can be determined with the thermal resistance defined as

$$\theta = \frac{(T_{\max} - T_{\text{in}}}{q''}\,, \tag{5}$$

where q'', T_{in}, and $T_{\max}$ are the heat flux, inlet coolant temperature, and the maximum temperature of the heat sink base plate, respectively.

The thermal resistances of pure water and of a 2%-volume fraction nanofluid for a 180-W/cm^2 heat flux input and different flow rates are illustrated in Fig. 8. It shows that in the investigated range of flow rates the thermal resistance defined by Eq. (5) is reduced when a nanofluid is applied as a coolant. Moreover, the plate pin-finned heat sink reduces the thermal resistance to 23% in comparison with a traditional plate fin heat sink.

5. CONCLUSIONS

In this study, the heat transfer characteristics of a novel mini-channel heat sink cooled by alumina–water nanofluids were investigated experimentally. Tests were performed for volume fractions in the range from 0.5 to 2%. The temperature distributions obtained are then used to evaluate the thermal resistance that characterizes the heat sink performance. The key findings of the study are as follows:

- Al_2O_3 nanoparticles dispersed in water increase the heat transfer coefficient of the both heat sinks significantly. This surpassing performance can be mainly attributed to the higher thermal conductivity of the nanofluids and Brownian motion of particles.
- The heat transfer coefficient increased with increasing particle concentrations and the increased amount of heat transferred did not decrease with increase in the Reynolds number.
- The thermal resistance of miniature heat sinks decreased to as low as 0.0083 K/W for the plate pin-finned heat sink and 0.105 K/W for the plate fin heat sink. Also, the thermal resistance of the plate pin-finned heat sink was by 23% lower than that of the plate fin heat sink. This shows that using circular pins for increasing the area of the heat sink surface can be regarded as promising way for replacingt conventional plate fin heat sinks.

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