Numerical Study of Channel Pin-Fin Heatsink Heat Transfer Performance using MWCNT As A Coolant

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***Abstract*:- The development of thermal enhancement in the electronics industry is the most critical issue in today's real time and arithmetic world [47-49]. Micro channel heat sinks are significantly used to dissipate heat from miniaturized heat flux generating electronic components. This research works further enhancements in the thermal and cooling performances of a microchannel heatsink by employing Al_2O_3/water as a coolant. In this study, heat transfer characteristics, Reynolds number and pressure performances of Al_2O_3/water cooled microchannel heatsink at three inlet velocities, volume fraction and compared with a water-cooled microchannel heat sink. To investigate the performance of the microchannel heat sinks, numerical analysis, and a conjugate heat transfer model is solved in a commercial code ANSYS-CFX. The results reveal a maximum enhancement of 15.13% in average heat transfer coefficient for microchannel heat sink using Al_2O_3/water as a coolant at inlet conditions of 0.2, 0.4, and 0.6. Similarly, a temperature drop reduction of 35.79% is computed for Al_2O_3/water as a coolant at the corresponding mass flow rate. Moreover, the results suggest that replacing water with Al_2O_3 as a coolant at various volume fractions can further reduce power consumption in the channel heat sink pin by 60.65% and 62.41%, respectively.**

Keywords: Nanofluids, microchannel, heatsink, Al_2O_3/water, heat transfer coefficient, and Reynolds number

1. INTRODUCTION

Improving the heat transfer of electronic components with lower power consumption is a major challenge for engineers and researchers today, especially the use of mechanical techniques in the miniaturized electronic components is not an appropriate method. Due to the rapid advancement in the information technology industry, electronic components and devices are facing great challenges in cooling systems. In today's world of electronic devices, it is important to be more efficient when making miniaturized IC Chips. The size of electronic devices is being reduced while its performance speed and needs are increasing rapidly. Due to the vast growth in the microchip manufacturing sector has greatly reduced the size from 100 to 1 μm of the microprocessors has led to high heat flux generation [1]. While the microelectronics are operating, its operating temperature should not exceed the temperature limit specified by the company, due to this it needs a proper cooling system. It is clear from many research articles that the existing cooling system is in a situation where heat dissipation and electronic consumption are not optimally controlled due to which the efficiency and durability of electronic devices are significantly reduced. This research work was carried out with liquid cooling systems and it replaced the air cooling systems due to their limited heat dissipation capabilities and bulky design.

The Conventional heat transfer fluids such as water, oil, ethylene glycol, etc. are utilized in several kinds of research works for heat transfer systems. Due to the low thermal conductivity of conventional heat transfer fluids act as a barrier in improving the thermal performance and compactness of the heat transfer equipment [2–7]. It is also a critical constraint during the design of compact thermal systems with high efficiency.

Choi and Eastman [8] introduce a novel type of heat exchanger fluids that are suspended in the base fluid, form a colloidal solution of nanoparticles in a base fluid. They predicted that the obtained nanofluids exhibit higher thermal conductivity than the conventional fluids. Nanofluids not only solved the problems like sedimentation of large particles, clogging of flow-channels, corrosion, and erosion of pipelines but also significantly improved the heat transfer characteristics [2,5,8–14]. So this is a very interesting one because the heat transfer enhancement is further improved by the fluids and it ensures that the thermal requirements of the electronic devices are properly met.

However, the cooling mechanism is especially important as the development and performance of electronic devices are related to cooling technology. In 1981, Tuckerman and Pearse first introduced the microchannel heat sink by keeping account of the challenges in the operational capabilities of electronic devices and cooling techniques [15]. Micro/mini channel heat sink can extract a huge heat flux from a small surface area due to the large forced convective heat transfer effects in the channels. Cooling systems such as mini and microchannel heat sinks are the most suitable solutions for high heat dissipation in electronic devices with high heat flux generation when using liquid as coolants [16]. Recent decades show that numerous research efforts have been taken to improve the thermal management performance of mini and microchannel heat sinks by channel geometry optimizations [17, 18], header shape augmentations [19], and coolant nanofluids [20].

A Related Work

One of the most critical issues in the electronic industry is the thermal management of electronic devices, especially meeting the limitations on the maximum operating temperature and ensuring temperature uniformity across the devices. Increased integration and tiny electronics have created issues for thermal design, development, packaging, and evaluation of novel high-performance cooling systems. There have been recent attempts to embed cooling devices into the electronics as integrated parts of the overall design [21-23]. Recently, a number of different cooling solutions, including two-phase flow, nucleate boiling, microchannel heat sinks, nanofluids, and jet impingement, have been examined by a number of different researchers [24–26]. The microchannel heatsink has been shown to be a high-performance cooling method among the cooling methods.

Using nanofluids Al_2O_3/H_2O and TiO_2/H_2O, Ijam et al. [27] found that the thermal performance of a mini channel heat sink was improved by 17% and 16%, respectively. Ho et al. [28] observed that using alumina nanofluids as a coolant in a copper minichannel heat sink causes the frictional factor to increase by a little amount. Vinoth et al. [29] carried out an experimental investigation into the influence of the cross-section of the channel on the heat transfer performance of an oblique finned channel. Alumina/water nanofluids were employed in three various forms of ribbed microchannels, which were constructed and used in three different ways: square, semicircle, and trapezoidal.

An experiment on the zigzag microchannel heat sink was carried out by Duangthongsuk and Wongwises [30]. They studied the influence of single cross-cutting of the flow channel on the Nusselt number and pressure drop characteristics, as well as on Reynolds number and particle concentration. In a series of studies, Maganti et al. [31] used parallel microchannel heat sink topologies for microprocessor cooling, and the results were promising. Their coolant was comprised of a different class of nanofluids than ours: Aluminum oxide, copper oxide, carbon nanotubes (CNT), graphene, and silicon oxide nanoparticles distributed in the water. A group of researchers, led by Sohel et al.[32], has solved the governing equations of nanofluids (copper and alumina distributed in water and ethylene glycol) flowing through a circular microchannel and minichannel heat sinks to produce concentrated entropy. They came to the conclusion that nanofluids provided the fastest lowering rates of fluid friction entropy generation rate, which were 38 percent and 35 percent, respectively, at a 6 percent volume fraction of nanoparticles in the fluid.

Coolants containing suspended metallic nanoparticles were examined numerically by Palm et al. [33] to determine their heat transfer enhancement capabilities when used inside standard radial flow cooling systems. Their findings clearly demonstrate that the utilization of these fluid/solid particle combinations can result in significant heat transfer improvements. Nazififard et al. [34] quantitatively examined the properties of Al2O3/water nanofluids for potential use in water-cooled research reactors. Their findings indicated that utilizing nanofluids with a volume fraction of 1% increases heat transmission by around 4% and that the pressure drop of nanofluids is only about 3% greater than that of the base fluid.

Many studies have been conducted on the modification of the thermal properties of base fluids by the addition of suitable nanoparticles, i.e. the formation of nanofluids. These studies have taken into consideration nanomaterials such as carbon-based nanoparticles, including single-walled (SWCNTs) and multiwalled carbon nanotubes (MWCNTs) [35], graphite [36], graphene [37,38], and graphene oxide [39,40]. Because SWCNTs and MWCNTs have a lot of thermal conductivity, they can be used in applications that deal with heat. This means that their thermal performance is better than that of base fluids [41]. Kumaresan et al. [42] investigated the thermophysical characteristics of MWCNT nanofluids based on a water–ethylene glycol combination and found that the thermal conductivity was increased by a maximum of 19.73 percent at an MWCNT mass fraction of 0.45 percent.

Grag et al. [43] used MWCNTs water-based nanofluids to investigate the laminar convective heat transfer in a circular tube. The heat transfer coefficient for 1.0 mass percent MWCNTs/water nanofluid was increased by around 32%. Mukesh and chandrasekar [44] revealed that the friction factor and convective heat transfer rate are calculated using a double helically coiled tube heat exchanger and multiwall carbon nanotube (MWCNT)/water-based nanofluids as a cooling medium. Heat transfer is also thought to increase with an increasing volume concentration of MWCNT/water-based nanofluids. The objective of the present work is to numerically evaluate the heat transfer enhancement behavior of MWCNT/water-based nanofluids in microchannel heatsink and compared them with conventional coolants. In the Microchannel heatsink numerical analysis was carried out for different Reynolds numbers and volume concentrations of nanofluids. In order to examine the conjugate fluid flow and laminar forced convective heat transfer properties of MWNCT/water nanofluids flowing in a Microchannel heatsink, a three-dimensional computational fluid dynamics model was created using the commercial software programme ANSYS FLUENT.

II PROBLEM STATEMENT AND GOVERNING EQUATIONS

A heatsink enclosed within a channel filled with nanofluid is used to cool an electronic chip, as shown in Figure.1.

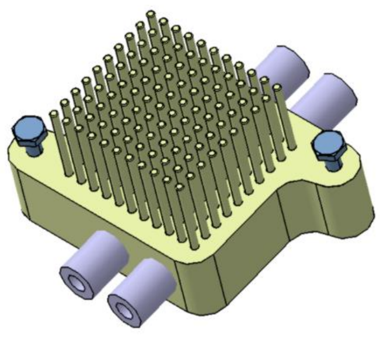

Fig.1 Pin-Fin Heatsink with microchannel

In this study, the nanofluid flowed into the channel from the one side of the channel and left it from the other side after meeting and cooling the electronic chip (see Fig. 1). Based on Fig. 1, the heat sink was installed on an electronic component with fixed heat flux. The heat sink and electronic components were perfectly insulated. This heatsink has several pin fins. The fluid passes through the pin fins and cools them. The electronic chip is heated due to its operation and generates a constant heat flux of 100 W/m. The dimensions of the heatsink and its other geometry details are shown in Fig. 1a. The nanofluid stream enters the channel with uniform velocity and temperature. The fully developed fluid flow exits the channel with a constant pressure boundary condition. Also, all parts except the area that has been subjected to heat flux are well insulated. Besides, a no-slip boundary condition is imposed on the walls.

A Governing Equations

Turbulent and laminar flows were studied. The following assumptions were made [44-46]:

- A single-phase, incompressible, and laminar/turbulent flow was assumed,
- The gravity, natural convection, and radiation were ignored, and
- The viscosity-induced heat generation was ignored.

Based on the above assumptions, Eqs.(1)-(5) are written. The three-dimensional continuity, momentum, and energy equations are given as

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}+\frac{\partial W}{\partial Z}=0 \quad (1)$$

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}+W\frac{\partial U}{\partial Z}=-\frac{1}{\rho_r}\frac{\partial P}{\partial X}+\frac{\mu_r}{\rho_r}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}+\frac{\partial^2 U}{\partial Z^2}\right) \quad (2)$$

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}+W\frac{\partial V}{\partial Z}=-\frac{1}{\rho_r}\frac{\partial P}{\partial Y}+\frac{\mu_r}{\rho_r}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}+\frac{\partial^2 V}{\partial Z^2}\right) \quad (3)$$

$$U\frac{\partial W}{\partial X}+V\frac{\partial W}{\partial Y}+W\frac{\partial W}{\partial Z}=-\frac{1}{\rho_r}\frac{\partial P}{\partial Z}+\frac{\mu_r}{\rho_r}\left(\frac{\partial^2 W}{\partial X^2}+\frac{\partial^2 W}{\partial Y^2}+\frac{\partial^2 W}{\partial Z^2}\right) \quad (4)$$

$$U\frac{\partial T_t}{\partial X}+V\frac{\partial T_t}{\partial Y}+W\frac{\partial T_t}{\partial Z}=-\frac{1}{\alpha_t}\left(\frac{\partial^2 T_t}{\partial X^2}+\frac{\partial^2 T_t}{\partial Y^2}+\frac{\partial^2 T_t}{\partial Z^2}\right) \quad (5)$$

The energy equation of the solid area is written as Eq.(6)

$$0=k_s\left(\frac{\partial^2 T_s}{\partial X^2}+\frac{\partial^2 T_s}{\partial Y^2}+\frac{\partial^2 T_s}{\partial Z^2}\right) \quad (6)$$

Therefore, the standard k ε method was employed in this study for simplification and computation reduction. As a result, the transfer equations are written as

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial(U\rho k)}{\partial X}=\frac{\partial}{\partial X}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial X}\right]+\mu_t\left(\frac{\partial V}{\partial X}+\frac{\partial U}{\partial Y}\right)\frac{\partial V}{\partial X}-\rho\varepsilon \quad (7)$$

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\frac{\partial(U\rho\varepsilon)}{\partial X}=\frac{\partial}{\partial X}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\epsilon}{\partial X}\right]+C_1\frac{\varepsilon}{k_f}\mu_t\left(\frac{\partial V}{\partial X}+\frac{\partial U}{\partial Y}\right)\frac{\partial V}{\partial X}-\rho C_2\frac{\varepsilon^2}{k_f} \quad (8)$$

where ρ is the density (kg/m^3), μ and μ_t are the dynamic viscosities of laminar and turbulent flows flow (kg/m.s), respectively, k is the turbulence kinetic energy, ε is the turbulent dissipation rate, C_1 is a constant of 1.44, C_2 is a constant of 1.92, and σ_ε and σ_k are the Prandtl numbers of turbulent flow. In addition, ε and k are equal to 1.3 and 1.0, respectively

B Nano fluid properties

In order to calculate the properties of nanofluids including density and specific heat capacity, the following equations have been used.

$$\rho_{eff} = \varphi\rho_{np} + (1-\varphi)\rho_f \quad (9)$$

$$c_{p,eff} \; \frac{(1-\varphi)(\rho c_p)_f + \varphi(\rho c_p)_{np}}{\rho_{eff}} \quad (10)$$

Where, the np index corresponds to the nanoparticles and the f index corresponds to the base fluid.

$$\frac{\mu_{eff}}{\mu_f} = 1 + 23.09\emptyset + 1525.3\emptyset^2 \quad (11)$$

The following equation has also been used to calculate the thermal conductivity according to the article [35]:
In the above relation, the temperature is measured in degrees Celsius. d_p represents the diameter of the nanoparticles and ϕ is the volume percentage of the nanoparticles that is equal to 1.

C Data acquisition
The hydraulic performance and thermal performance are two essential factors in the pin-fin structural investigation. The absorbed heat by the fluid flowing on the heatsink is

$$q = \dot{m}C_{pf}(T_{out} - T_{in}) \quad (12)$$

Where, m is the mass flow rate (kg/s), C_{pf} is the heat capacity (J/kg.K), and T_{in} and T_{out} are the input and output temperatures (K), respectively. Moreover, the convective HT coefficient h is written as in Eqs.(14)

$$h = \frac{Q}{A_s\left[T_w - \frac{T_{out}+T_{in}}{2}\right]} \quad (13)$$

Where, A_S is the area of the base plane, and T_w is the mean pin-fin temperature (K). The pressure is obtained as Eqs.(15)

$$\Delta P = \bar{P}inlet - \bar{P}Outlet \quad (14)$$

Where, P_{Inlet} and P_{Outlet} denote the inlet and outlet pressures, respectively.
The Nusselt number is calculated by Eqs.(16) .

$$= 1.0 + 5\emptyset\left(\frac{47}{d_p(nm)}\right) - 0.0248\emptyset\left(\frac{k_p}{0.613}\right) \quad (15)$$

$$Nu = \frac{hl}{k} \quad (16)$$

Where, l is is the characteristic length and k is the thermal conductivity of the fluid.
Thermal efficiency is calculated by using the Eqs.(17) [[30].

$$\eta = \dot{m}C_{pf}\frac{(T_{out}-T_{in})}{Q} \quad (17)$$

Where, Q is equal to the amount of heat applied to the heatsink.
The thermal resistance, as one of the important parameters, is used to evaluate the performance of cooling modules. The thermal resistance of the liquid-cooled are calculated using the following equations, respectively [31,32].

III RESULTS AND DISCUSSION

In this numerical study, the thermal and hydraulic performance of a channel heat sink pin is evaluated by employing Al_2O_3/water as a coolant. The heat transfer rate of a channel heat sink and hydrodynamic parameters of Al_2O_3/water nanofluids with different volume fractions such as 0.25%, 0.5%, and 0.75% were numerically studied and analyzed by CFD simulations.

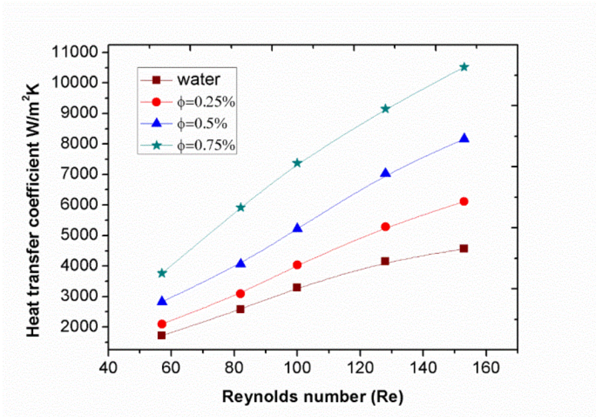

Fig.2Heat transfer coefficient versus Reynolds number for different volume fraction

Fig.2 shows the variation of the average heat transfer coefficient as a function of Reynolds number for various volume fractions of nanofluid. It can be seen in Fig.2, the heat transfer coefficient increases with an increase in Reynolds number. This means that a high inlet velocity is able to improve the cooling performance of the heat sinks. Also, it can be noticed in this figure the heat sink with nanofluid has the higher heat transfer coefficient in comparison with pure water.
It can be seen that an increase in the inlet velocity has increased the value of the Nusselt number. The reason is the increase in heat transfer coefficient by increasing the velocity. Besides, increasing the velocity reduces the amount of wasted exergy in the channel. Increasing the velocity reduces the amount of wasted exergy due to the reduction in the temperature at the outlet. Thermal resistance is the primary parameter to show channel heat sink pin performance. Thermal resistance for both distilled water and Al_2O_3/water is shown in Fig.3, which shows the decrement of thermal resistance with the increase of Reynolds number for both fluids.

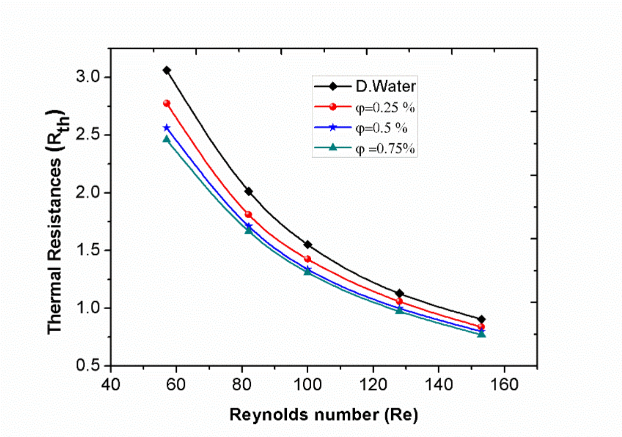

Fig.3 Thermal Resistance versus Reynolds number.

IV CONCLUSIONS

This study numerically investigated a channel heatsink with pin configurations and coolants. It was numerically investigated how the cooling capability and efficiency of channel Pin-Fin Heatsink could be improved. The performance of the channel Pin-Fin Heatsink with Al_2O_3/water as a working fluid, the thermal and hydrodynamic characteristics including heat sink base temperature, heat transfer coefficient, and thermal resistance for various volume fractions of nanoparticles are presented and discussed. The results reveal a maximum enhancement of 15.13% in average heat transfer coefficient for microchannel heat sink using Al_2O_3/water as a coolant at inlet conditions of 0.2, 0.4, and 0.6. Similarly, a temperature drop reduction of 35.79% is computed for Al_2O_3/water as a coolant at the corresponding mass flow rate.

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