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EXAMINATION OF REYNOLDS NUMBER DEPENDENCE ON THE EFFECT OF NUSSELT AND PRANDTL NUMBERS ON HEAT TRANSFER CHARACTERISTICS IN A FLAT FINS HEAT SINK

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ABSTRACT

This study examines the effect of dimensionless numbers such as Reynolds, Nusselt, and Prandtl numbers and their variations on heat transfer characteristics in a heat sink of flat fins. This was done to understand the influence of Reynolds number and air temperature in the heat analysis of electronic cooling. A numerical study was conducted for the heat transfer analysis under steady and laminar flow conditions for Reynolds numbers ranged from 100 to 400 and inlet air temperature between 10 to 40 ºC. A finite volume method with a solution-independent grid was employed to solve the conjugate heat transfer problem. Reasonable agreement was found between calculated dimensionless number and that of published literature. It was observed that at the lowest limit of 10ºC, increasing the Reynolds number from100 to 200 causes a 0.52 % increase in Prandtl numbers and a drop in the increase to 0.35 % at 40ºC. Under a similar condition, the percentage increase is more noticeable for the Nusselt number, with an increase of 103.7 % dropping to about 60 % as the air temperature rises to 40 ºC. Also, increase similar to that of Nusselt number was observed for that of heat transfer coefficient. The overall results suggest that Reynolds number plays significant role on the heat transfer characteristics and this will have implication on the cooling strategies. The results obtained can be used in electronic heat sink cooling by programming the regulations of the Reynolds number as dictated by surrounding air temperature to effect suitable heat transfer occurrence. Also, the realization of the growth of the Nusselt and Prandtl numbers with heat transfer characteristic may lead to new insights for generating heat transfer correlations for future research.

Keywords: dimensionless number, heat sink, electronic cooling, air-cooling, heat transfer coefficient.

1. INTRODUCTION

The operation of electronic components such as microprocessors, transistors, integrated circuits, Graphics Processing Unit (GPU), and Central Processing Unit (CPU) in devices like computational systems generates excess heat which could lead to performance loss, damages and system failure if left unchecked. To manage the temperature rise caused by this excess heat generated, a finned heat sink employing forced convection is commonly introduced.

Air-cooling is still employed in 95 % of the computer technology industry worldwide [1] due to the economic constraint and complexity of current thermal management methods such as liquid cooling and phase change material. The importance of forced air conventional heat sink for an air-cooling system cannot be overemphasized. It should be expected that the inclusion of a cooling axial fan into an electronic system consumes a significant amount of power. There have been studies on reducing this amount of power offered by the conventional axial fan by introducing technologies such as piezoelectric fans [2,3] and multiple fan arrays [4, 5] to the heat sink in which they can provide equivalent cooling capabilities for 50 % of the power requirements offered by an axial fan [6]. However, axial fans have not been neglected in the industry due to their cheap production and compact design [1].

The flow rate of the air passing through the fins of a heat sink is a significant factor in heat sink design [7]. The heat transfer characteristics are affected by the flow rate created by the fan: a higher flow rate will result in a higher air velocity (higher Reynolds number), which will result in improved heat transfer [8].The flow rate is directly proportional to the pressure drop across the heat sink, so higher flow rates result in higher pressure drops which leads to a huge amount of energy consumption. As a result, the design approach of the heat sink seeks a compromise between lower thermal resistance and tolerable pressure drop [9] but there is a drawback in the use of the heat sink for cooling: Irrespective of the air temperature (thermal resistance) present, a stipulated flow rate will be used which could be more than the required amount needed for the cooling thereby, leading to excess power used and no energy conserved.

In this regard, the research on the examination of Reynolds number dependence on the effect of Nusselt and Prandtl numbers on heat transfer characteristics such as heat transfer coefficient in a flat fins heat sink is relevant for energy conservation. It is worth noting that dimensionless numbers such as Nusselt and Prandtl numbers are important parameters in the study of heat transfer analysis. Therefore, it is important to understand the Reynolds number (axial fan flow rate) dependence on their effects on heat transfer characteristics of heat sink at selected air temperature. The results obtained should be applied in the operation of the axial fan of heat sink to provide the right Reynolds number for cooling at the corresponding air temperature such that energy is conserved.

Various studies on the heat sink as regards its heat transfer characteristics and improvement have been conducted extensively. For instance, the heat removal characteristics of a heat sink with fins of different unconventional geometries under the Reynolds number of 2000 to 11000 were studied. They found that the fin with slotted kidney shapes arranged staggered provides the highest removal rate [10].

In [11], computational and experimental studies on the effect of perforated pin fin on the heat transfer characteristics in a heat sink with different flow rates were conducted. Their results showed that there is 11 % increase in the Nusselt number for the perforated pin fin heat sinks when compared to its corresponding non-perforated pin. Interestingly, [12] carried out a study on a heat sink of pin fin arranged inline and staggered. It should be noted that they conducted a numerical study on the heat sink with fins of different perforation sizes, shapes, and numbers. It was found that the higher Nusselt number occurs as perforation size and number increases and also the heat removal rate was the highest for heat sinks with elliptical fins of elliptical perforation of 40.5 % more than the conventional ones. Moreover, in [13], the heat removal rate of a heat sink of pin fin with a hole at its base ranging from Reynolds number 6468 to 45919 was numerically investigated. They discovered that the heat sink with a base plate having a hole (D_h/D_b) less than 0.15 offered a better heat transfer performance. Similarly, through an experimental investigation conducted in [14], the highest Nusselt number and heat transfer coefficient were obtained with the heat sink of spiral pin fin under the Reynolds number ranging from 322 to 1982 when compared to that of other pin fin geometries such as cylindrical and rectangular.

The effect of fins types and configurations on the heat transfer characteristics has been a subject of attention to provide an improvement on the heat sink thermal performance. For instance, fins with perforation of different shapes but of the same surface area were investigated. It was discovered that circular perforation removes more heat from the system but the perforation shapes such as slotted and rectangular provide higher heat transfer coefficient and Nusselt number [15]. This is not surprising for [16], where numerical analysis of the effect of perforation shapes on a flat fin heat sink was carried out. It was found that about an average of 4.5 % more heat was removed by the perforated fin which is better than that of a solid fin. Similarly, it is worth noting that [17] studied the heat transfer characteristics of flat fin normal and inclined at30^{o}and 60°arranged in an inline and staggered formation at Reynolds number ranges from 4000 to 18000 and it was observed that the inline arrangement with a 30° inclination fin provides the maximum heat transfer coefficient. Moreover, [18] discovered a higher heat removal rate for wavy flat fin compared to the conventional flat fin in their numerical study carried out on a wavy flat fin heat sink to observe its effect on the heat transfer characteristics.

In [19], the advantages of adopting straight parallel-fin heat sinks with add and subtract fins through the use of numerical simulation and experimental approaches were explored. Different parameters were explored in three-dimensional numerical simulations. Computational simulations were run on two different types of straight plate fins: straight plate-subtract fin heat sinks and straight plate-add fin heat sinks. Experiments were conducted with three other forms of similar dimensions in turbulent forced-convection with Re from 6300 to 35,120 and heat flux from 1194 to 23353 W/m^2. When compared to straight than other shapes fins, the heat transfer coefficient shows a high agreement between numerical simulation and laboratory tests, increasing by around 38 and 43 % for straight with added semi-circular and straight with subtracted semi-circular fins, respectively. Furthermore, while comparing straight with added semi-circular and straight with subtracted semi-circular fins to other forms of fins, the Nusselt number increases by around 21 % and 32 %, respectively.

Based on previous studies, emphasis on the heat sink, fin design and configuration have been centred on the heat sink thermal performance improvement while few studies have been conducted to analyse the effect of important dimensionless numbers in heat transfer analysis such as Reynolds, Nusselt and Prandtl number on heat transfer characteristics of a heat sink. It is worth noting that some of these studies have focused on high Reynolds numbers and it would be of interest to see the influence of low Reynolds number on the effects Nusselt and Prandtl numbers can have on the heat transfer characteristics of a heat sink. This is necessary to have more insight that will aid improvement in the development of cooling strategies for better applications/systems. Therefore, this paper aims to study the effect of Nusselt and Prandtl number son heat transfer characteristics (such as heat transfer coefficient) at various low Reynolds numbers using numerical simulation in a non-perforated flat fin heat sink.

The aim of the study is to provide the extent of influence of Reynolds number increase on heat transfer parameters so as to provide understanding on efficient electronic cooling strategies inflat fins heat sink usage, and to understand the heat transfer characteristics such as heat transfer coefficient against dimensionless numbers for future research on correlations regarding heat transfer.

To achieve this aim, the following objectives are accomplished:

- to perform a mesh sensitivity study on the model developed and determine the validity of the numerical model by comparing its result to that obtained from experiment or previously published work;
- to extract the relevant thermophysical results from the simulated numerical model and analyse the influence of Reynold number change on the simulated Nusselt, Prandtl number, and simulated heat transfer coefficient
- to observe the heat transfer coefficient variation with

the Prandtl number and Nusselt number at different Reynolds numbers.

2. MATERIALS AND METHOD

The heat sink (Figure-1) used for the study has its flat fins in a staggered arrangement with a transverse and longitudinal pitch of 0.75 mm. The fin is assumed to be made of Aluminium and its dimensions are measured as 3.8 mm in length, 0.75 mm in thickness and 5 mm in height. Figure-2 shows the physical and computational domain which depicts a symmetry section cut from the heat sink to simplify the numerical study to be considered. The characteristic length used in the calculations of Reynolds and Nusselt numbers is the fin's length. The inlet plane of the computational domain was assumed 6 times the characteristics length from the heat sink while the outlet plane was assumed 15 times the characteristics length to avoid backflow effects. A laminar airflow under a steady-state condition was assumed. The study was conducted for low Reynolds numbers ranging from 100 to 400.

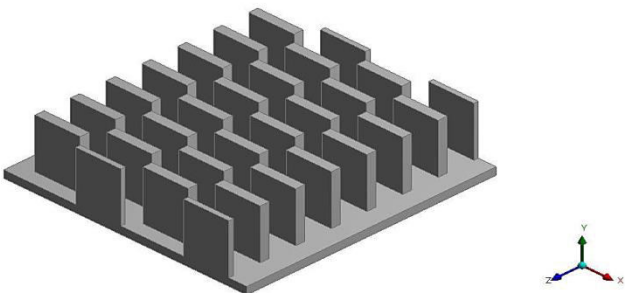
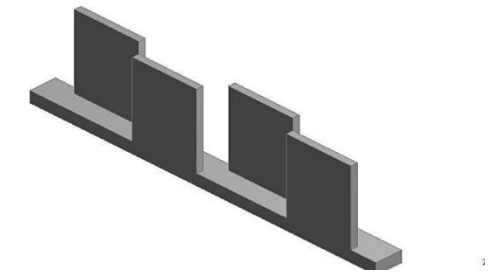
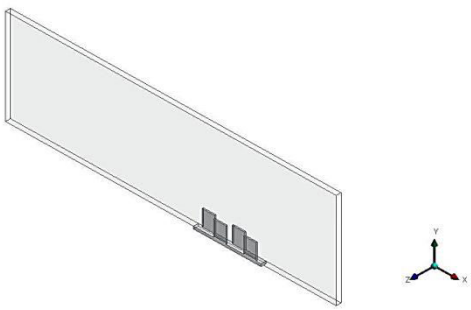


Figure-1. The flat fin heat sink.



(a)



(b)

Figure-2. The symmetrical cut of the heat sink: *a* – the physical domain; *b*– the computational domain.

As shown in Figure-2b, the YZ plane at the positive direction of the domain represents the velocity inlet boundary condition, while that at the negative direction represents the pressure outlet boundary condition. The two XY planes of the domain represent the symmetry boundary condition, the top and bottom directions of the XZ plane were considered adiabatic wall. The free stream velocity and temperature were assumed to be the inlet velocity and temperature, respectively. Varied inlet air temperatures (10, 20, 30 and 40ºC) were considered to take into account the variation of Prandtl numbers in the study. The ranges of inlet air velocities (1) used in the study were derived from the Reynolds numbers 100 to 400 and on the air properties as stated by the temperature of the inlet air where ρ_T is the density of air at the inlet air temperature, v is the inlet air velocity, L is the characteristics length, and μ_T is the air dynamic viscosity at the inlet air temperature. Therefore, four velocities were considered for each Reynolds number. Furthermore, sixteen conditions were observed and simulated based on the studied inlet air temperature and Reynolds number.

$$\mathrm{Re}=\frac{\rho_T v L}{\mu_T}. \tag{1}$$

The base surface of the heat sink assumed a constant heat flux [20] of 4500W/m^2. The continuity (2), momentum (3) and energy (4) governing equations used are:

$$\frac{\partial u_i}{\partial X_i}=0, \tag{2}$$

$$u_j\frac{\partial u_i}{\partial X_i}=-\frac{1}{\rho}\frac{\partial P}{\partial X_i}+\frac{\partial^2 u_i}{\partial X_i^{\,2}}, \tag{3}$$

$$u_j\frac{\partial T}{\partial X_j}=\alpha\frac{\partial^2 T}{\partial X_i \partial X_i}. \tag{4}$$

The difference in the upstream and downstream pressure is denoted by *P* while T represents fin surface temperature and u_i, u_j denotes the velocity components in the x- and y-directions. A Computational Fluid Dynamics (CFD) software called ANSYSFluent was used to solve the Navier-Stokes and energy equations in a three dimensional incompressible, steady form simultaneously.

A suitable mesh setting was used for the numerical model which was selected by conducting a mesh sensitivity study. There are insufficient experimental researches for the current model of staggered arranged flat fins. The numerical results of [20] for solid flat fins were used to validate the current numerical model. [20] showed that in the numerical modelling, the distance between the inlet and the block obstruction must be big enough to eliminate the impacts of the boundary. The computational

plate's leading edge was 15D (D = fin thickness) from the inlet boundary. To compute the findings for validation, it was conducted between Reynold numbers of 100 to 350 and the free stream temperature was kept at 22ºC.

The simulations were conducted under Reynolds numbers 100 to 400 and at free stream temperatures of 10 to 40 ºC. Figure-3 shows the step-by-step approach employed for the simulations. The average wall temperature (T_w) of the heat sink and the free stream temperature (T_o) is used to obtain the film temperature (T_f=(T_w+T_o)/2) which is used to estimate the properties of air such as specific heat, kinematic viscosity, and thermal conductivity needed to obtain the Prandtl number, Pr (5) at the film temperature.

$$P_r = \left(\frac{\mu . c_p}{k} \right)_{T_f}, \quad (5)$$

$$h = \frac{q}{A(T_w - T_o)}, \quad (6)$$

$$Nu = \frac{q.L}{kA(T_w - T_o)}. \quad (7)$$

The Fluent solver aids in the estimation of the convection heat transfer coefficient field around the heat sink from which its area average is estimated, h(6) and also the Nusselt number (7). T_w is the wall temperature, and T_o is the free stream temperature. L is the characteristics length, q is the heat removal rate, A = 121.68mm^2 is the total surface area of the fin surfaces and base surface of the heat sink

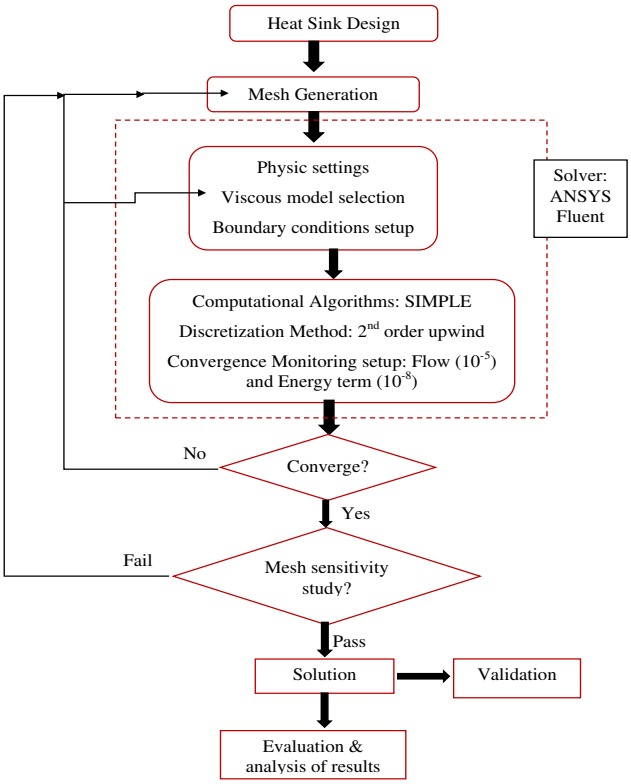


Figure-3. Flowchart of Simulation Methodology.

The approach shown in Figure-3 was performed for each of the sixteen conditions that were simulated based on their Reynold numbers and inlet air temperature. From each simulation, results such as the dimensionless numbers and heat transfer coefficient were extracted.

3. RESULTS AND DISCUSSIONS

To establish that the solutions obtained are insensitive to the model grid, various grid settings were studied to establish that the obtained solutions were independent of the mesh by conducting a mesh sensitivity study.

Table-1 displays the mesh independence test for the simulation at Reynolds number 100 and inlet temperature 10 ºC, from which the minimum grid size of 549,000 is enough for obtaining sufficient results for the current simulation.

Table-1. Mesh independence test for Reynolds number 100 at inlet temperature 10 ºC.

Number of element	Nusselt number	Friction coefficient
260,000	1.70	0.0661
320,000	1.69	0.0658
412,000	1.69	0.0662
549,000	1.68	0.0658
761,000	1.68	0.0659

The numerical data of [20] were considered in validating this model. For the validation, the inlet air and free stream temperature were set to 22 ºC under Reynolds number ranging from 100 to 350 as shown in Figure-4.

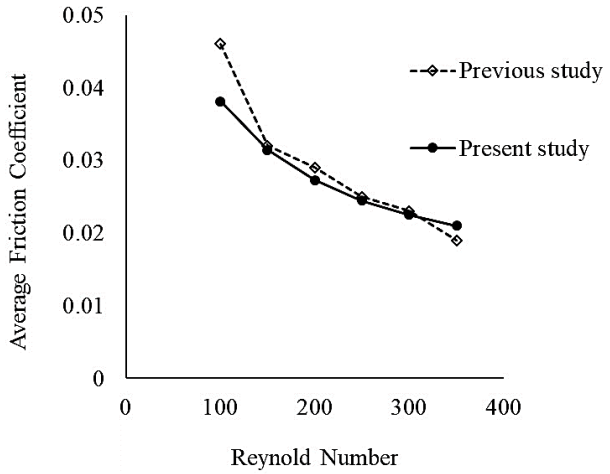


Figure-4. Comparison between results of [20] and that of current numerical study: Friction drag against Reynold number.

Figure-4 presents the average friction coefficient comparison on the heat sink and that of [20] and a good agreement can be found between both studies. After the numerical simulation and convergence has being reached, the calculated Prandtl number, Nusselt number, and heat transfer coefficient were extracted and their variations with Reynolds number under the studied inlet air temperature were displayed in Figure-5 to Figure-7.

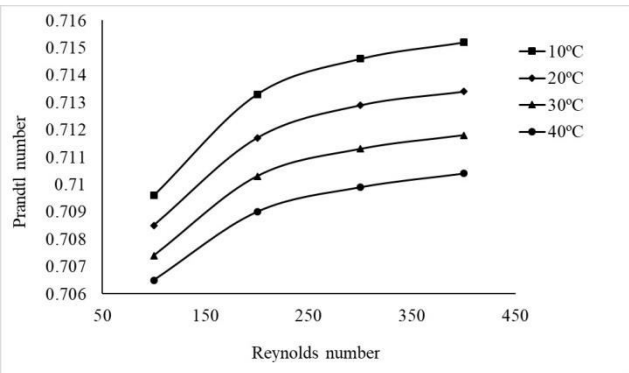


Figure-5. Prandtl number variation with different Reynolds numbers for different inlet air temperature.

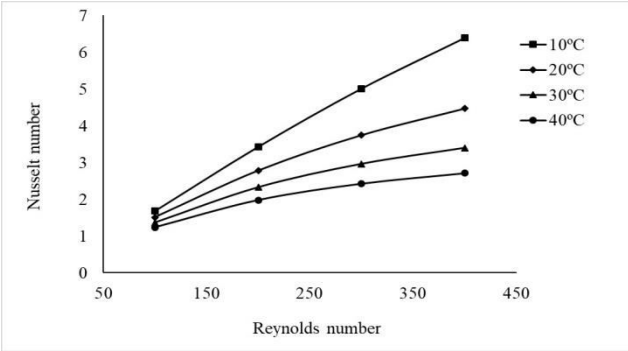


Figure-6. Variation of Nusselt numbers with different Reynolds numbers for different inlet air temperature.

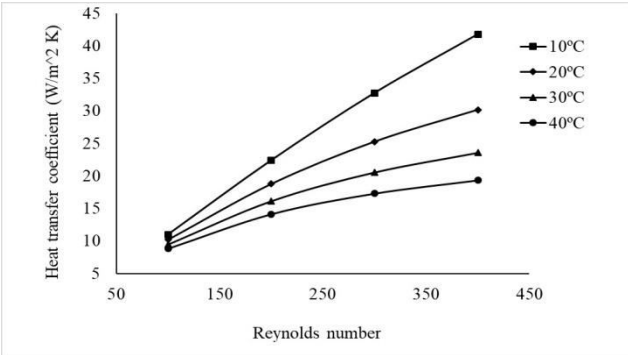


Figure-7. Variation of heat transfer coefficient with various Reynolds numbers for different inlet air temperature.

Tables 2, 3 show the influence of Reynolds number increase on Prandtl numbers and Nusselt numbers, respectively within the inlet air temperature range of 10 to 40 ºC.

Table-2. Percentage increase in Prandtl numbers under various conditions.

Inlet air Temperature ºC	Percentage increase %		
	100 to 200	200 to300	300 to400
10	0.52	0.18	0.08
20	0.45	0.17	0.07
30	0.41	0.14	0.07
40	0.35	0.13	0.07

Table-3. Percentage increase in Nusselt numbers under various conditions.

Inlet air Temperature ºC	Percentage increase %		
	100 to 200	200 to300	300 to 400
10	103.73	46.16	27.52
20	84.02	34.65	19.37
30	70.35	27.38	14.72
40	59.98	22.44	11.75

The percentage change in heat transfer coefficient was similar to that of Nusselt number as observed in Figure-6 and Figure-7 that their variation with Reynolds number acted similarly. However, as the Reynolds number changes under a particular air temperature, the degree of change in the dimensionless numbers were presented. Percentage increase occurs for the analysis of the dimensionless number (Prandtl and Nusselt number) and also the heat transfer coefficient.

Figure-5 and Figure-6 display the heat transfer coefficient variation with Prandtl and Nusselt numbers, respectively at different Reynolds numbers. It can be observed that as the Prandtl and Reynolds number increases, the heat transfer coefficient also increases

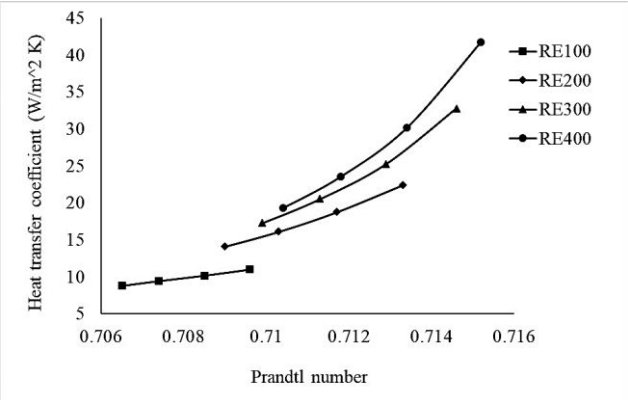


Figure-8. Heat transfer coefficient variation with Prandtl number at different Reynolds numbers.

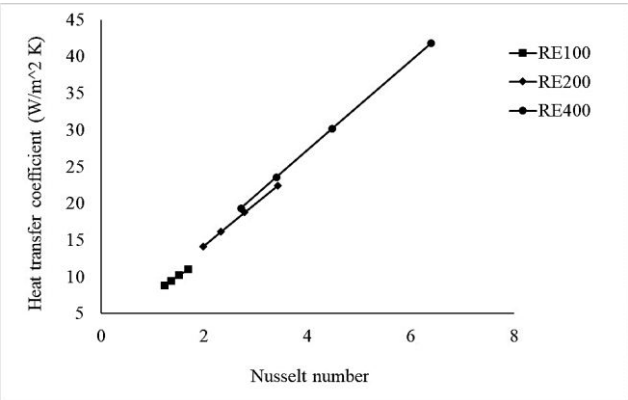


Figure-9. Heat transfer coefficient variation with Nusselt number at different Reynolds numbers.

Investigation at Reynolds number of 300 was excluded to show a clearer chart in Figure-6 for the remaining Reynolds numbers.

3.1 Discussion of the Results from the Simulation of the Studied Conditions

As presented in Tables-2 andTable-3, at an inlet air temperature of 10 ºC, increasing the Reynolds number from 100 to 200 causes a percentage increase of 0.52 % in Prandtl numbers, the increase drops further to 0.35 % as the inlet air temperature increases to 40 ºC. These show a better Prandtl number improvement under lower air temperature conditions. The percentage increase is more pronounced for the Nusselt number with an increase of 103.7 % dropping further to about 60 % as the air temperature increases to 40 ºC. It is worth noting that similar variation occurs in heat transfer coefficient as shown in Figure-5 and Figure-6 as it is known from (7) that a relationship between Nusselt number and heat transfer coefficient as shown in (8) can be observed which indicates an increase in Nusselt numbers dictate a corresponding increase in the heat transfer coefficient,

$$Nu = \frac{hL}{k}. \tag{8}$$

It can be observed that as the Prandtl and Reynolds number increases, the heat transfer coefficient also increases as reviewed in Figure-8 and Figure-9. This reflects as a result of the increase in momentum diffusivity and strong convective influence of the air. A progressive transition from linear to exponential growth could be observed between the relationship of heat transfer coefficient and Prandtl number as the Reynolds number increases. Meanwhile, for every Reynolds number, linear growth is observed between heat transfer coefficient and Nusselt number. This confirmed the direct proportionality between Nusselt number and heat transfer coefficient as shown in (8) where the gradient is the ratio of the characteristic length to the thermal conductivity of the fluid. It was also observed that there was an increase in variation of heat transfer coefficient with Nusselt number as the Reynolds number increases.

The scope of the study was limited to steady and laminar flow of air within the Reynolds numbers of 100 to 400 using a heat sink of flat fins. Future work may be considered on the transient and turbulent flow of air at higher turbulent numbers. Also, implementing this study on other preferred heat sink types.

In the development of thermal management of electronic components using heat sink of flat fins,as presented in Table-2 and Table-3, the knowledge of the percentage increase in the dimensionless number and heat transfer coefficient as the Reynolds number changes at a particular air temperature can be programmed to the circuitry of the cooling fan in regulating the airflow velocity (i.e. the Reynolds number) going to the electronic component thereby dictating the rate of heat removal occurring.

The trend, response, and growth of the heat transfer characteristics with change in Prandtl and Nusselt numbers at different Reynolds numbers could offer insights leading to new ways to create heat transfer correlations in the future.

4. CONCLUSIONS

The examination of Reynolds number dependent on the effect Nusselt and Prandtl numbers can have on the heat characteristics in a flat pins heat sink was numerically carried out and a solution-independent grid size of 549,000

was selected through the mesh sensitivity study. A reasonable agreement was found from the validation when comparing the simulated friction coefficient of the model to that of published literature.

From the simulated results, it shows that there was a better improvement in the Prandtl number under lower air temperature conditions when increasing the Reynolds number. In the case of increasing the Reynolds number from 100 to 200, there was an increase of 0.52 % which dropped further to 0.35 % as the inlet air temperature increased. The Nusselt number experience increase under similar conditions as that of the Prandtl number but with a more pronounced increase of 103.7 % and a drop to 60 % at 40 ºC. This should be useful in influencing efficient heat transfer for energy conservation purposes.

The heat transfer coefficient tends to exhibit exponential growth in relationship with the Prandtl number as the Reynolds number increases but display a linear growth for that of Nusselt numbers in the studied Reynolds numbers. It was observed that the variation of the heat transfer characteristics with Nusselt numbers increases for every increase in Reynolds number. These insights could lead to new ways to create heat transfer correlations in the future.

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