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Study on Heat Transfer and Pressure Drop through Different Channel Heights and Widths

Raed Shakir [1] , R. SHAKIR [2]

Abstract

Numerical analyses were conducted to study the heat transfer characteristics and pressure variation in micro-scale heat sinks under consistent heat flux conditions. The study involved making predictions for Reynolds numbers ranging from 2819.3 to 5874.5 and pressure drops ranging from 0.035 to 0.55 bar. The micro-scale heat sinks with different channel heights and widths were fabricated using wire electrical discharge machining (EDM). The investigation explored the impacts of various geometric configurations and heat flux on heat transfer properties and pressure drop in the micro-scale. The geometry of the micro-scale was found to significantly influence pressure drop and heat transfer enhancement. The study aims to provide valuable insights that can guide the design of micro-scale heat exchangers, enhancing heat transfer performance for small-scale devices. The heat transfer coefficients ranged from (0.268 to 0.507) kW/m2.k at the inlet, and from (0.919 to 7.130) kW/m2.k at the outlet. Additionally, the Nusselt numbers varied between (10.43 to 19.81 at the inlet and (9.93 to 18.94) at the outlet. Furthermore, the pressure drops were within the range of (35.55 to 278.037) mbar at the inlet and (70.73 to 550.250) mbar at the outlet. These findings pertain to two micro-channel heat sinks with differing heights and widths, as detailed in the results section.

Keywords: Convection, Pressure drop, Conduction, Heat sink, Geometry configuration

Affiliation of Authors

[1, 2] University of Thi-Qar, College of Engineering, Iraq, Thi-Qar, 64001

[1] raed-sh@utq.edu.iq
[2] shraed904@gmail.com

[2] **Corresponding Author**

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دراسة انتقال الحرارة وانخفاض الضغط من خلال ارتفاعات وعروض مختلفة للقنوات

م. رعد شاكر [1] ، م. رائد شاكر [2]

المستخلص

أجريت التحليلات العديدة لدراسة خصائص انتقال الحرارة وتغير الضغط في المشتتات الحرارية ذات النطاق الصغير تحت ظروف تدفق الحرارة المشروطة. وتضمنت الدراسة وأكتشاف تنبؤات لأرقام رينولدز تتراوح من (2819.3 إلى 5874.5) وانخفاضات الضغط تتراوح من (0.035 إلى 0.55 بار). تم تصنيع المشتتات الحرارية صغيرة الحجم ذات ارتفاعات وعروض مختلفة للقنوات باستخدام ماكينة التفريغ الكهربائي السلكية (EDM). استكشف البحث تأثير التكوينات الهندسية المختلفة وتدفق الحرارة على خصائص نقل الحرارة وانخفاض الضغط على المستوى الجزئي. تم العثور على هندسة المقياس الصغير تؤثر بشكل كبير على انخفاض الضغط وتحسين نقل الحرارة. تهدف الدراسة إلى تقديم رؤى قيمة يمكن أن توجه تصميم المبادلات الحرارية صغيرة الحجم، مما يعزز أداء نقل الحرارة للأجهزة صغيرة الحجم. تراوحت معاملات انتقال الحرارة من (0.268 إلى 0.507) كيلوواط/م2.كلفن عند المدخل، ومن (0.919 إلى 7.130) كيلوواط/م2.كلفن عند المخرج. بالإضافة إلى ذلك، تراوحت أرقام نسلت بين (10.43 إلى 19.81) عند المدخل و(9.93 إلى 18.94) عند المخرج، كما تراوحت انخفاضات الضغط بين (35.55 إلى 278.037) ملي بار عند المدخل و(70.73 إلى 550.250) ملي بارعند المخرج. تتعلق هذه النتائج باثنين من أحواض الحرارة ذات القنوات الصغيرة ذات الارتفاعات والعروض المختلفة، كما هو مفصل في قسم النتائج.

الكلمات المفتاحية : الحمل الحراري، انخفاض الضغط، التوصيل، المشتت الحراري، التكوين الهندسي

انتساب الباحثين

[1، 2] جامعة ذي قار ، كلية الهندسة ، العراق، ذي قار، 64001

[1] raed-sh@utq.edu.iq
[2] shraed904@gmail.com

[2] **المؤلف المراسل**

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Introduction

The increasing demand for improved performance in electronic devices emphasizes the critical need for efficient heat dissipation at a larger scale. Traditional industries have utilized heat sinks with varied geometric designs to manage heat. J. P.

Holman [1], a prominent figure in the field of heat transfer and thermodynamics, is highly respected for his valuable contributions to engineering education through insightful textbooks and research. His written works are renowned for their clear elucidation of fundamental principles, extensive coverage, and practical applicability in the field of engineering. Holman's writing style excels in engaging readers, simplifying intricate topics and enhancing the learning experience. However, investigations and practical applications of miniaturized heat rejection devices remain limited. Miniaturization has long intrigued a broad spectrum of fields, sparking significant interest. R.SHAKIR [2]–[11] has presented in-depth analyses and discussions on heat sinks, providing crucial analytical and numerical predictions for various heat sink configurations such as flat channels, pin-fins channels, micro-channels, and more. These studies shed light on the impact of geometric configurations on the thermal efficiency of mini-scale heat sinks. Collectively, these research endeavors underscore the close alignment between numerical studies and experimentally reported results. The objectives of the present investigation regarding guess testing are as follows:

- Generating fresh estimated data for air applicable to miniaturized heat sink configurations.
- Hypothesizing a significant parametric pattern and exploring the potential influence on pressure drop (DP) and heat transfer features.
- Evaluating the accuracy of existing correlations for miniaturized heat sink configurations using new estimated data.
- Enhancing a novel airflow correlation to improve heat transfer predictions for miniaturized heat sink setups.

The Set-Up

The configuration comprises four primary components: a flow loop, a miniature version of the testing area, an aluminium test specimen, and the procedural condition.

Flow Loop

A schematic representation of the experimental arrangement is depicted in Figure (1). The test loop prediction involved a cooling to airflow loop and a data acquisition system. Airflow was utilized as the working fluid, and the wind tunnel's air duct had rectangular dimensions of (40 cm by 40 cm) and a length of (200 cm), constructed using acrylic material. The duct was insulated with Aero-flex compliant material (20 mm thick) on sheet metal. The air in the wind tunnel was discharged into the wind tunnel using an air compressor, flowing straight into the test section and eventually being released to the ambient environment. The airflow rate was accurately "measured" using a flow meter with a precision of (0.5%) on the full scale. The differential pressure (DP) in the metering section was measured using a differential pressure sensor with an accuracy of (0.05%) of the full scale. Pressure taps were affixed at (2) locations on the wall, one for downstream and the other for upstream on the test section. K-type thermocouples, with an accuracy of (0.5%) on a full scale, were employed to measure the air temperature. Consequently, (2) thermocouples were used for measuring the inlet and outlet temperatures of the airflow, utilizing probes with a diameter of (1mm) inserted into the channel through which the airflows passed. All thermocouple probes were pre-calibrated using a dry box temperature calibrator with an accuracy of (0.05°C). as shown in Figure (1).

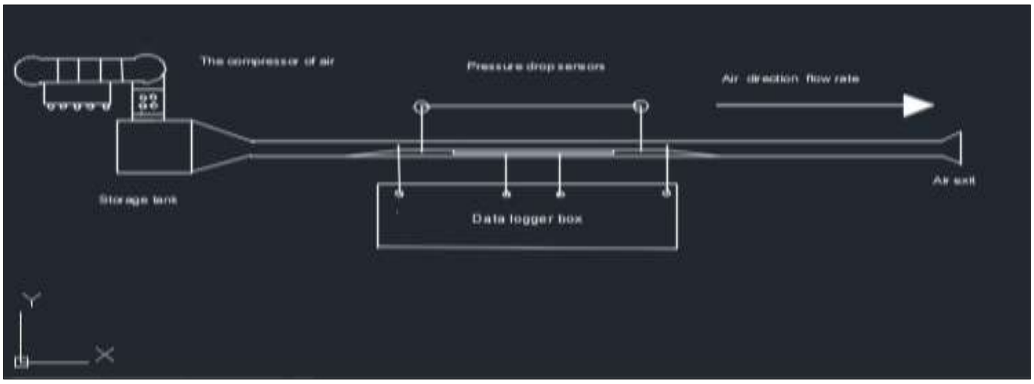

Figure (1): The Flow Circuit.[8]

The mini-Scale Of Test Section

The diagram in Figure 2 illustrates the layout of the test section. The smaller version of the heat sink measures 50 by 50 mm in width and length, and 100 by 100 mm, respectively. The miniaturization of the heat sink was achieved using Wire Electrical Discharge Machining (WEDM), with the specifics documented in Table 1. The power for the flat-type heaters was supplied by an AC power source. To insulate the rear face of the test section, a 20 mm thick heat-resistant Mica standard board, a 10 mm thick Aero flex standard board, and a 10 mm thick Acrylic standard sheet were applied. Two types of thermocouples, T-copper-constantan, with 1 mm diameter probes, were utilized to measure the wall temperature of the miniaturized heat sink. Special adhesive was used on the rear face of the wall to secure the thermocouples in place. The thermocouples were mounted on the wall by drilling holes from the rear face and affixing them securely. as shown in Figure (2).

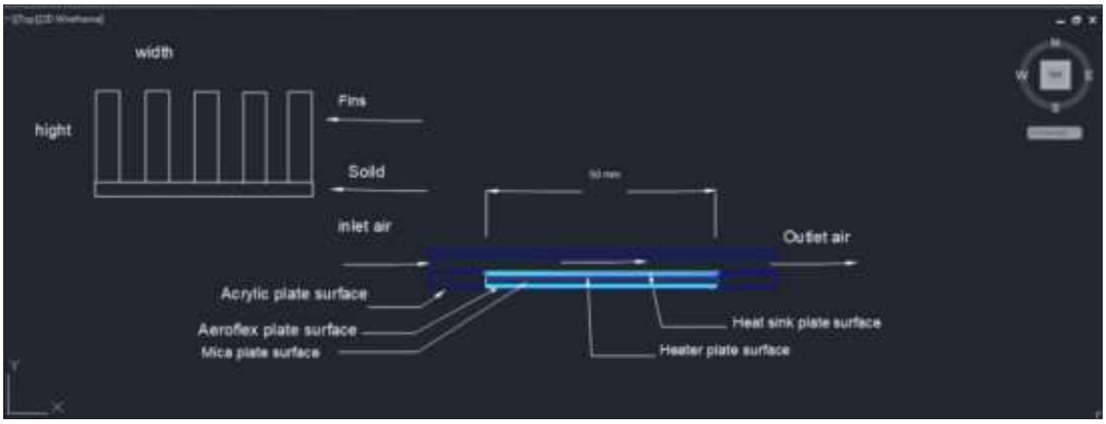

Figure (2): Experimental Segment and Test Component.[8]

The Conditions of Procedure

Estimations involved both conduction and convection through varying heat fluxes and airflow rates entering the test section. During these estimations, a slight increase in airflow rate was implemented to maintain a steady heat input to the miniaturized heat sink. Additionally, heat input to the walls of the mini-scale heat sink was regulated to achieve the desired scale by utilizing an "electrical" heater. Energy was provided by adjusting the supplied current and voltage to the heaters, and this was controlled using a "digital clamp. The attainment of a steady-state heat was determined through an energy equilibrium, considering the sensible heat via airflow. For this study, only the data conforming to the conditions of energy equilibrium were utilized, employing an iteration technique for all predicted tests. Multiple measurements were recorded for each temperature location, including the differential pressure (DP) across the test piece, and these were collected and processed using the data acquisition system. In addressing uncertainties in the estimations, the airflow meter had an uncertainty of ±0.05, the thermocouple temperature measurement had an uncertainty of ±0.05, and the pressure transducer differential had an uncertainty of ±0.05, among other factors. It can be seen in the mini-scale heat sink dimensions on the as shown in table.(1).

Table (1): Mini-scale Heat Sink Dimensions

Channel	Criteria			
	Channel height h (mm)	Base thickness and fins s (mm)	Channel width w (mm)	Size of mini-Scale of Heat Sink (mm)
1	1.0	1.0	1.0	50 × 50
2	2.0	2.0	2.0	100 × 100

Mathematics

Estimations of heat transfer influenced by airflow parameters involve analyzing factors like fluid temperature based on air-flow and wall temperatures at system pressure. This analysis employs over five hundred equations to estimate heat transfer characteristics, such as air transfer temperature. Additionally, temperatures on the allocation of heat flux can exist at the solid-air interface. Figure (3) illustrates this, showcasing a 2-D array where the major impact is along the wall line. heat sink are determined using Excel software through an iterative data analysis process.The objective is to determine that only a single The 1-D array is perpendicular to the airflow, while the 2-D array runs parallel to the airflow. The equation provided allows the determination of thermal conductivity[3]&[4].

$$\delta^2 T/\delta y^2 + \delta^2\, T/\delta z^2 = 0 \qquad (1)$$

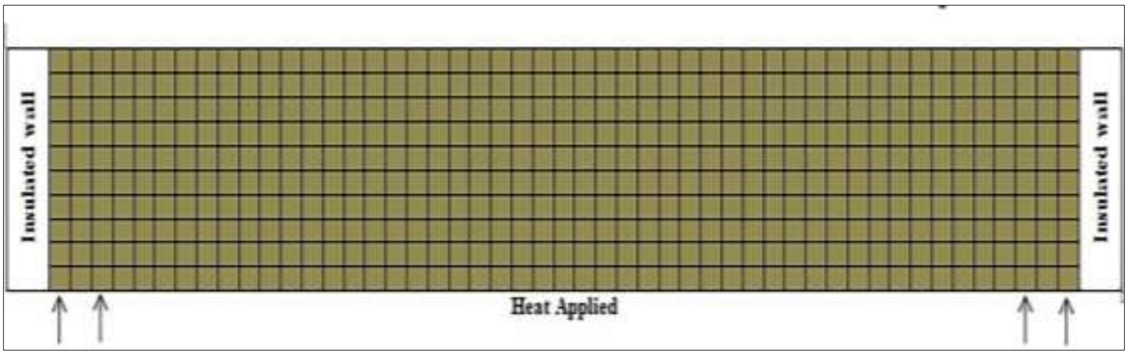

Figure (3) :Thermal Conduction through the Aluminum Test Piece Wall [3]&[4]

In the context where (y) is perpendicular to the airflow axis and (T) represents the temperature on the aluminum wall, the heat transfer in formula (1) is calculated by dividing it by the square cell area of dimensions (1mm) and (2 mm), alongside (δz) and (δy) which denote the cell sizes. Equation (2) is designed to adjust based on boundary conditions and is solved iteratively until the temperature in each cell matches the previously determined estimated value, with the error kept at (0.001) through the iterative process. [3]&[4].

$$T_{i,j} = \delta y^2\left(T_{i+1,j} + T_{i-1,j}\right) + \delta z^2\left(T_{i,j+1} + T_{i,j_1}\right)/2\left(\delta y^2 + \delta z^2\right) \quad (2)$$

As illustrated in Figure 3, each cell must attain energy balance. [1]

$$q_h W_{cell} = \alpha_h (T_w - T_a)\left[W_c + 2\eta_{fin} H_c\right] \quad (3)$$

So, (Wcell) represents the foundational region of a unit cell,

$$W_{cell} = W_C + W_S \quad (4)$$

Whereas (ηfin) denotes the efficiency of the fin.[1]

$$\eta_{fin} = tanh\left(\beta_{fin} H_C\right)/\beta_{fin} H_C \quad (5)$$

In the context where (βfin) represents the parameter related to the fin. [1]

$$\beta_{fin} = \sqrt{2\,\alpha_h / K_A W_S} \quad (6)$$

The value of (Tw) at small scales, both mini and micro, can be obtained as. [1]

$$T_W = T_{th} - \frac{q_h L_{th}}{K_A} \quad (7)$$

The value for (Ta) at small scales, both mini and micro, is acquired through. [1]

$$T_a = T_{in} + \frac{q_h W Z}{M_a C_{pa}} \quad (8)$$

The Nusselt number calculation can be obtained by. [1]

$$NU = 0.0214\left[1 - (D_h/L)^{2/3}\right] * \left[Re^{0.8} - 100\right] Pr^{0.4} \quad (9)$$

The Prandtl number (Pr) can be obtained by. [1]

$$P_r = \frac{C_p\, \mu_a}{k_a} \tag{10}$$

The Reynolds number can be obtained by.[1]

$$R_e = \rho_a V D_h / \mu_a \tag{11}$$

In the context where (Dh) is observed.[1]

$$D_h = A_h / P_h \tag{12}$$

Boundary conditions for the mini-scale have been computed based on theorized (f_f). [1]

$$f_f = 0.0791\, Re^{-0.025} \tag{13}$$

The pressure drop can be determined by.[1]

$$\Delta P_a = 2\, f_f\, \rho_f V^2 \frac{L}{D_h} \tag{14}$$

Results and Discussion

Figure 4 illustrates changes in the average heat transfer coefficient concerning the air Reynolds number. As anticipated, the heat transfer coefficient increases with higher heat flux due to the greater temperature difference between (T-air) and (T-heat sink). Lower heat flux results in a heat transfer coefficient lower than that of higher heat flux. Additionally, Figure 4 displays variations in the average heat transfer coefficient predicted against the air Reynolds number for different geometrical configurations. It is evident from Figure 4 that the heat transfer coefficient significantly increases with increasing Re-air. The impact of channel height and width on enhancing the heat transfer coefficient is also depicted. This is because the heat transfer coefficient is influenced by the heat transfer rate, larger heat transfer area, and greater roughness of the heat sink surface. For a given Re-air, the heat transfer coefficient at (w=2.0 mm, h=2.0 mm) is higher than that at (w=1.0 mm, h=1.0 mm). Furthermore, the results for (3) mass flux and (4) heat flux via all experimental tests are shown in Figure (4).

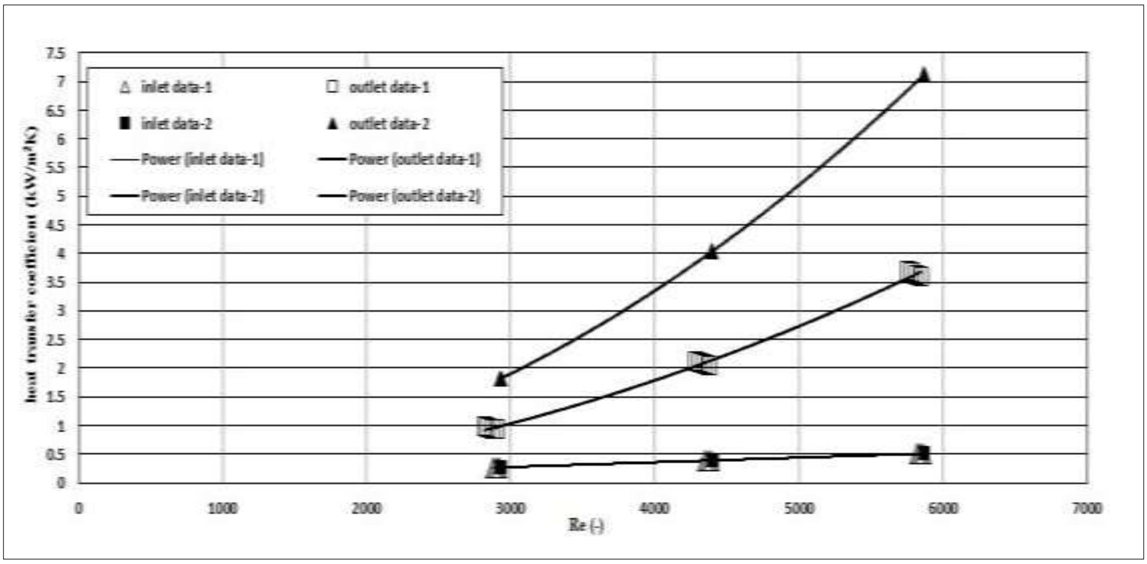

Figure (4): Heat Transfer Coefficient versus Reynolds Number

Figure (5) depicts the fluctuation of pressure drop (DP) with Re-air for various channel widths and heights. As the air mass flow rate increases, leading to a higher heat transfer rate, the increase in (DP) is greater than that of the air mass flow rate. Consequently, (DP) tends to increase as the air mass flow rate rises. Greater surface area and surface roughness result in an elevated heat transfer rate, causing the (DP) of a channel with (w=2.0 mm & h=2.0 mm) to surpass those of (w=1.0 mm & h=1.0 mm) observed at the inlet and outlet locations.

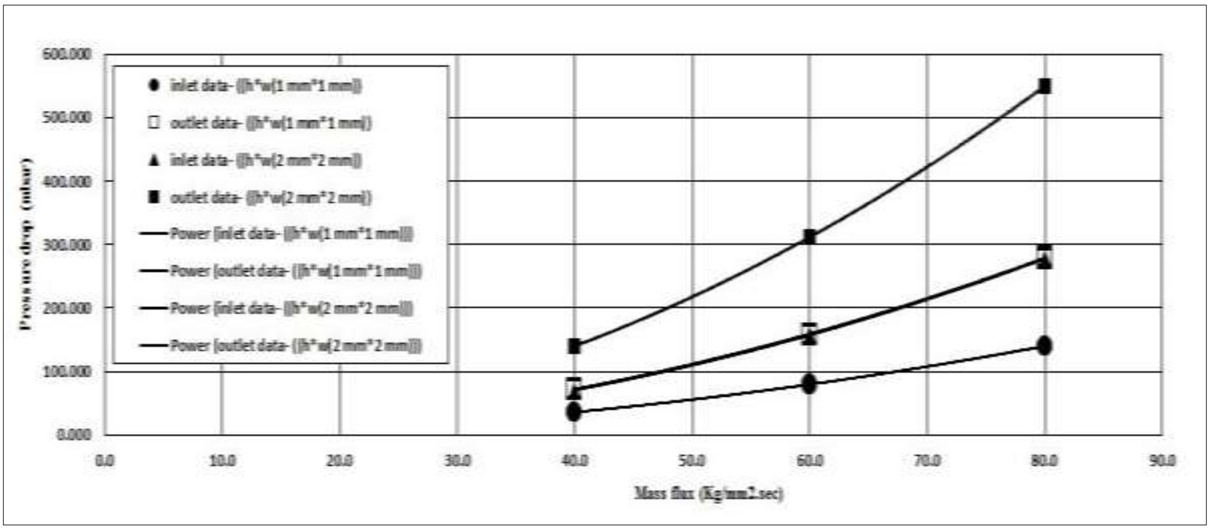

Figure (5): Relationship between Pressure Drop and Reynolds Number

Figure 6 illustrates the changes in pressure drop (DP) in millibars for different geometrical configurations. The airflow characteristics within the micro-channel are considerably intricate and enhanced compared to traditional scales. This complexity is attributed to the greater surface roughness and increased surface forces at the miniaturized scale. It is evident from Figure 6 that the (DP) consistently increases with the air Reynolds number. Moreover, the size and shape of surface roughness irregularities on the miniaturized scale significantly influence the variations in (DP), as depicted in Figure 6.

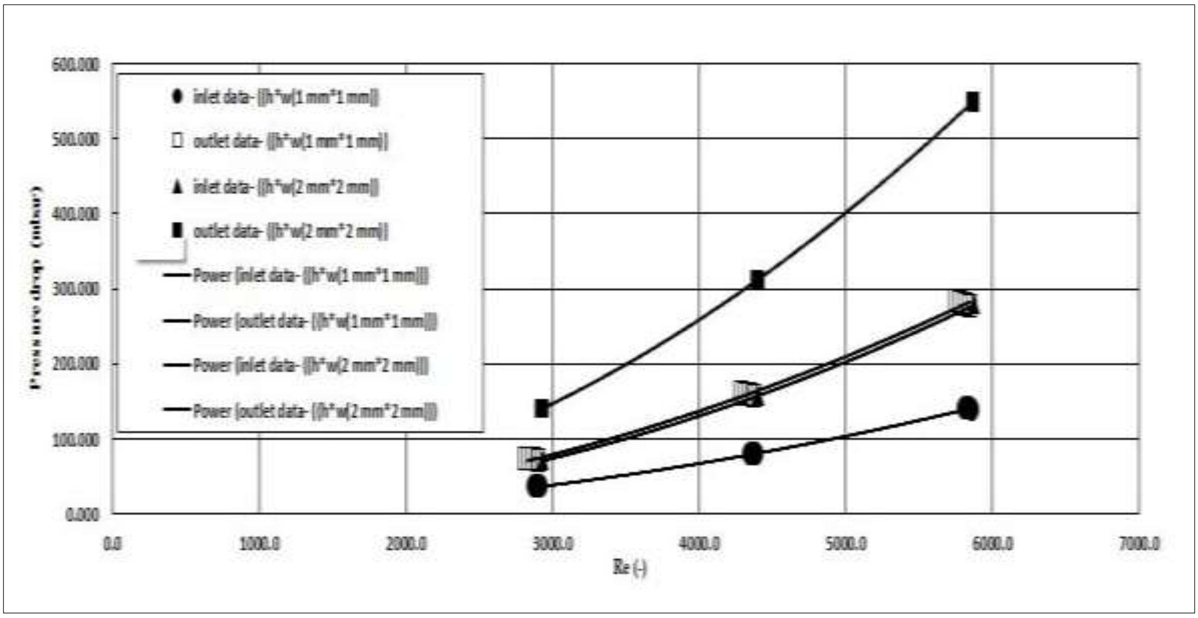

Figure (6): DP against Re-air

The impact of (h-channel) and (w-channel) on the enhancement of Nusselt number (NU) is depicted in Figure 7. Due to a larger heat transfer area and increased surface roughness, the Nusselt number

(NU) of the heat sink with (w=2.0 mm & h=2.0 mm) is greater than that with (w=1.0 mm & h=1.0 mm). Furthermore, the results can be visualized in an alternative representation, as shown in Figure 7, with the same interpretation as in Figure 7.

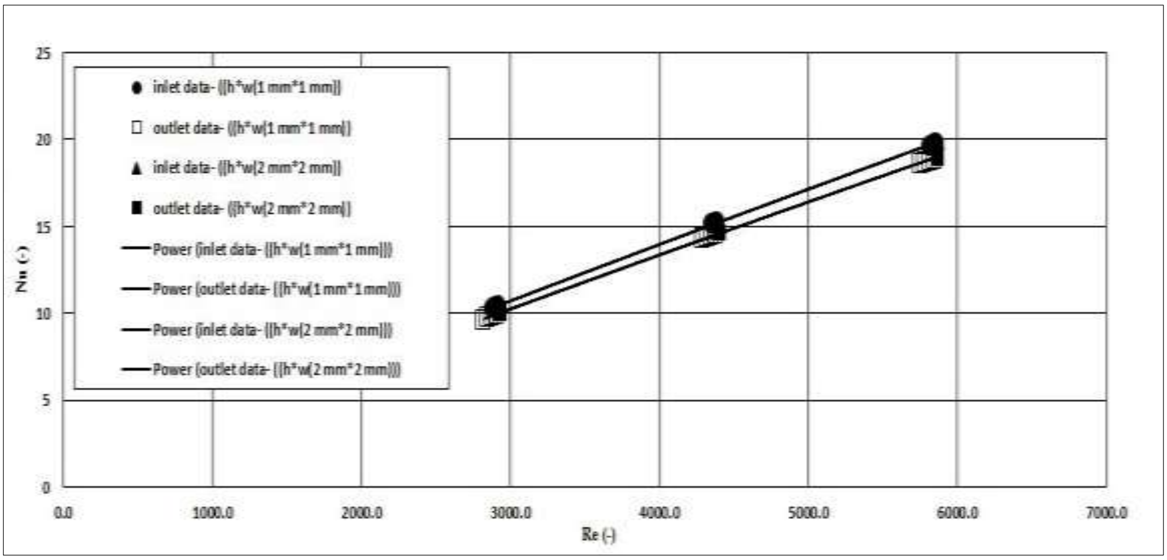

Figure (7): Nusselt Number for Air versus Reynolds Number for Air

Conclusions

The primary focus of the study was to examine the potential advantages of heat transfer in the small-scale context of a heat sink. The conclusions were heavily based on iterative approaches and heat transfer equations. The significant finding pertained to subsequent procedural steps. Turbulent and transition flows were observed across to tested scenarios, displaying development in each case. Notably, the heat transfer coefficient at the outlet position surpassed that at the inlet position. In all test instances, the pressure drop (DP) at the outlet was greater due to elevated wall and liquid temperatures. The study revealed that the rate of heat transfer depended on the airflow's cooling capability. Consequently, the heat sink's temperature decreased as airflow re-air increased. The study also highlighted the substantial impact of surface roughness size and configuration at the small-scale on enhancing heat transfer and affecting alterations in pressure drop (DP). The coefficient of heat transfer (NU) and (DP) in the results referred to the micro-channel. Upon comparing the results, it was evident that test pieces with larger height and width in the figures of the results section exhibited higher values. This was attributed to differences in hydraulic diameter and heat flux variation between the two test pieces.

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