COMPARISON OF USING AIR, CO₂, AND HELIUM FOR THE COOLING OF SQUARE-SHAPED ELECTRONIC PARTS CFD Study with Entropy Generation Analysis

by

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Numerical simulation has been used in the current work to investigate improving the cool-down of electronic parts of cubical form involving dummy parts within a rectangular duct. Three working fluids (air, CO_2 , and helium) were used to cool 12 electrical chip arrays in the duct. The simulation investigates the effects of cooling fluid type and shifting hot element placements on whole cooling functioning at various Reynolds numbers. Also, the impact of the distance among electronic parts is researched. This is accomplished by moving the heat sources while leaving other components in their original positions as dummies to preserve the flow characteristics. The Reynolds number falls between (500-19000). The dimensionless entropy generation number reduces with the rise of the Reynolds number, while the pumping power ratio increases. It is determined that the dimensionless entropy generation computed for the case of constant viscosity of air yields slightly greater values than those obtained for the case of temperature-dependent viscosity. A high level of agreement in the experimental work is used to verify the standard k-model. Key words: entropy generation, electronic component cooling, heat dource,

CFD numerical simulation

Introduction

The high operating temperature of electronic devices such as laptops, cell phones, and digital cameras is one of the main causes that damage their components. The operating temperature increases to critical value due to the devices' limited size, design, and mechanical speed because of improved processing speed design and shrinking of such devices' sizes. As a result, more heat is generated, raising the working temperature to unsafe levels over the critical limitations. It is crucial to optimize the cooling process since the breakdown rate of these devices is absolutely related to the operating temperature. Therefore, improving the cooling

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process is essential to avoid high heat generation [1]. In literature, many researchers have dealt with improving the cooling of electronic devices [1-6]. There are three methods to improve the cooling process: enhancing the surface area of electronic components (fin or pin surface), using PCM, and improving thermal conductivity, spacing, and flow rates of working coolant [7-10]. The effectiveness of cylindrical lithium-ion battery cooling devices was quantitatively examined by Behrooz *et al.* [11]. This research concentrated on the cooling system's intake and outlet and battery distance impact. The results demonstrated that the input size and Reynolds numbers, change affected the operating temperature and pressure. Selvan and Manavalla [12] used two types of PCM materials, OM35 (50:50) and OM35 (60:40) for cooling E-machine. Three configurations were numerically created to study the thermal behavior of brackets during the cooling process. The outcomes demonstrated that high heat was transferred with increasing spacing between the brackets.

The use of nanofluid as a new coolant with a mini channel for downsizing electronic components was also presented in a review by Bahiraei and Heshmatian [13]. Bahiraei and Monavari [14] investigated the influence of irregularly shaped nanoparticles in a micro-channel heat sink at varying Reynolds numbers. The results showed the lowest heat resistance for the platelet particle-containing suspension. Greiner [15] showed that slow re-circulation in channeled regions minimizes convective heat transfer, revealing diffusion as the primary mode of heat transfer. A numerical analysis for cooling a heat sink made of triangular pins was given by Alam et al. [16]. According to the findings, raising the air velocity raised the Nusselt number, improving heat removal from the CPU. Ali et al. [17] analyzed experimental and numerical data when considering the impact of distance between two-element heat sources. They demonstrated relationships between the Nusselt number and the package spacing with Reynolds number value (2464.6 \leq Re_I \leq 16430). The features of heat transmission in grooved channels were given by Farhanieh et al. [18]. The results showed that the pace at which heat was transferred from heated components was significantly affected by the degree to which the flow was reattached and recirculated. When grooved channels were compared to flat channels, the findings revealed a 300% improvement in heat transmission. Asako and Faghri [19] simulated the laminar and turbulent flows with a localized heat flux for a heated arrangement block. The outcomes revealed that transportation in dense arrays was more effective when approaching from above.

Molki and Fagri [20] described the forced convection of air-cooled rectangular blocks in a line. Nakayama and Park [21] investigated conjugated heat transfer from an air-flow-mounted rectangular block. Laminar mixed convection heat transport in a quadrilateral channel was studied quantitatively by Kursun and Sivriolu [22]. Bahiraei *et al.* [23] numerically studied the elliptical pin-fin heat sink while employing silver-water nanofluids for cooling. Bahiraei and Mazaheri [24] have tested spiral liquid blocks with nanoplatelets made of graphene to cool electronic devices. The spiral liquid block's heat transmission was excellent and energy efficient.

A theoretical and experimental examination was carried out by Refaey *et al.* [2, 3] to enhance the heat transmission from 12 cubical heat sources within a quadrilateral duct. This research investigated how heat transmission is affected by the element spacing and the cooling air-flow rate. Only two heat source components were explored using CFD and experiments. The rest were dummy elements to maintain flow characteristics, with Re_L ranging from 4108-17115. At a Re_L value of 8538, the most distant second heat source boosted the efficiency of the first upstream heat source by 17% and 10%, respectively. When all elements in the array were heated and compared to only two heat elements, a maximum reduction of approximately 19% and 15% in average Nusselt number for an in-line and lateral position was obtained when the second heat element was located at positions 8 and 4, respectively, at Re = 17115. This was found after heating all of the array's elements. Several papers [25-29] examined the working fluid (single phase or nanofluids), entropy creation, and heat transfer enhancement.

Since the experimental work on this topic is tough for large ranges of Reynolds numbers, the current study was planned to present numerous numerical CFD simulations for an extensive range of Re_L from 500-19000. The current article examines various factors, such as the cooling fluid employed (three different working fluids are used), the distance between heat sources, and the flow velocity of the cooling fluid. As a result, a total of 165 cases in the present study involving the position of heat sources were analyzed. The cooling efficiency as a function of the heat source placement is studied. In addition, an entropy generation was analyzed for the studied cases.

Numerical analysis

The computational domain simulates the cooling air-flow through or across an array of rectangular parts demonstrating electronic components (chips 3×4) on a circuit board. The simulation examines how altering hot component placements affects cooling performance at various Reynolds numbers values and the impact of electronic component spacing. Changing positions strategy: all blocks with heat (on), one block (the specified ones) without heat that is deemed a dummy element to retain flow characteristics, and two or three components without heat in the same inline distinguished plane. Creating 11 runs for each Reynolds numbers value (five points), totaling 165 cases. Figures 1(a)-1(c) show the computational domain test section information. Chip domains are twelve rectangular metal blocks (3 columns \times 4 rows). The first row is 500 mm from the duct entry and at the bottom to provide fully established flow conditions.



Figure 1. Details description of CFD domain; (a) specifies view, (b) sectional view, (c) CFD domain, (d) mesh details, and (e) mesh zoom-in view

The computational domain generates a 3-D uniform volume mesh. The domain uses tetrahedron components, whereas the chips domain uses structured elements. A tiny mesh near the chip walls with the dimensionless y + maximum consequence of 1.0 eliminates the boundary-layer's influence on CFD results, this can be shown in figs. 1(d) and 1(e). Boundaries are fluid input velocity at the inlet according to Re_{L} values containing five points: 500, 1000, 5000, 13000, and 19000, inlet temperature 300 K, pressure outlet at the outlet, adiabatic duct walls, and chips bottom walls. The 140000 W/m³ (3.5 W) is the heat production of each heat source. Refaey *et al.* [2] recommended the standard *k*- ε turbulence model with a turbulence intensity of 5% to be the best model among other used models that describe the presently studied problem.

The calculated parameters heat flux, q, average heat transfer coefficient, h, Reynolds number, and average Nusselt number, are presented in [17]:

$$q = \frac{Q_{\text{net}}}{A_{\text{s}}} \tag{1}$$

$$h = \frac{q}{T_{\rm m} - T_{\rm i}} \tag{2}$$

$$\operatorname{Re}_{\mathrm{L}} = \frac{vL}{v} \tag{3}$$

$$Nu_{L} = \frac{hL}{k}$$
(4)

Cartesian co-ordinates are used by Ali *et al.* [17] to solve continuity, momentum, and energy equations to determine temperature distribution, flow field, and heat transfer in the duct. The existing model's solution converged when continuity, momentum, and energy residuals approached 10^{-4} , 10^{-6} , and 10^{-7} . Five sets of grids are used to verify mesh independence, including the number of cells: 320805, 413187, 558128, 771436, and 931553. Based on the average surface temperature of Chips 5-8 at Re_L = 5000, a mesh independence test was performed. It was found that, when starting to use the grid with cells number 558128 and so on the afterwise grids, a minimal effect on temperature changes was recorded with an average deviation value of 0.006%. Hence, the grid with cell numbers 558128 is utilized for all cases in this work [1]:

$$\nabla(\rho U) = 0 \tag{5}$$

$$\nabla (\rho UU) = -\nabla P + \nabla \left[\mu_t \left(\nabla U + \nabla U^T \right) \right]$$
(6)

$$\nabla \left(-k\nabla T + \rho C_p T U \right) = S_h \tag{7}$$

where S_h is the thermal source term.

The validation of the current numerical model can be found in Refaey *et al.* [2]. This validation was conducted for air with the experimental data available from Refaey *et al.* [3] of heat energy produced by a single chip located in Place 5 (referred to as a single heat source or by two chips situated in Places 5 and 6, respectively for Reynolds number between 3611 and 14174. The validation results showed an average difference of 14.6% between numerical and experimental data.

Analysis of entropy generation and pumping power

Considering the array of chips as a rectangular duct, as shown below, the entropy generation during heat transfer from chips to air-flows in the duct is computed. A constant viscosity situation and a temperature-dependent air viscosity case are both explored. When doing this study, all active chips are used as the heat transfer source. An average heat transfer coefficient is used to transport heat to most of the fluids. Since air is the most prevalent fluid used in these applications, only air underwent analysis. For various Reynolds numbers, the entropy production and pumping power are evaluated.

The rate of heat transfer to the flowing fluid of the control volume may be shown:

$$\delta \dot{Q} = \dot{m}C_p dT = h \pi D (T_w - T) dx$$
(8)

The Second law applied to the duct considered gives:

$$\dot{S}_{\text{gen}} = \frac{\delta S}{\delta t} - \frac{Q}{T} + \sum_{\text{out}} \dot{m}s - \sum_{\text{in}} \dot{m}s \ge 0$$
(9)

Considering air as a pure substance, the Canonical relation may be expressed:

$$\frac{\mathrm{d}h}{\mathrm{d}x} = T\frac{\mathrm{d}h}{\mathrm{d}x} + \frac{1}{\rho}\frac{\mathrm{d}h}{\mathrm{d}x} \tag{10}$$

Using the Canonical relation, the generation of entropy rate may be expressed:

$$d\dot{S}_{gen} = \dot{m}C_p \left(\frac{T_w - T}{T_w T} dT - \frac{1}{\rho C_p T} dp\right)$$
(11)

It may further be expressed:

$$d\dot{S}_{\text{gen}} = \dot{m}C_p \left(\frac{T_{\text{w}} - T_i}{T_{\text{w}}T} e^{\frac{-4S_i x}{D}} dT + \frac{f\bar{U}^2}{2C_p dT} dx\right)$$
(12)

Integrating the aforementioned equation along the duct length, the entropy generation can be shown:

$$\dot{S}_{\text{gen}} = \dot{m}C_p \left[\ln\left(\frac{1-\alpha e^{-4S_t\phi}}{1-\alpha}\right) - \alpha \left(1-e^{-4S_t\phi}\right) + \frac{f\alpha E_c}{8S_t} \ln\left(\frac{e^{4S_t\phi}-\alpha}{1-\alpha}\right) \right]$$
(13)

It is possible to calculate the dimensionless entropy generation, which is the ratio of entropy generation per unit of heat transport:

$$N = \frac{1}{1 - e^{-4S_t\phi}} \times \left\{ \ln\left(\frac{1 - \alpha e^{-4S_t\phi}}{1 - \alpha}\right) - \alpha \left(1 - e^{-4S_t\phi}\right) + \frac{1}{8} \left(\frac{64}{\text{Re}}\right) \frac{\alpha E_c}{S_t} \ln\left(\frac{e^{4S_t\phi} - \alpha}{1 - \alpha}\right) \right\}$$
(14)

For a fully developed flow under laminar conditions and for the constant viscosity, the friction factor, f, is related to the Reynolds numbernas f = 64/Re.

For the case of constant viscosity, dimensionless entropy generation may be obtained:

$$N = \frac{1}{1 - e^{-4S_t\phi}} \times \left\{ \ln\left(\frac{1 - \alpha e^{-4S_t\phi}}{1 - \alpha}\right) - \alpha \left(1 - e^{-4S_t\phi}\right) + 8\alpha \frac{E_c}{\operatorname{Re}S_t} \ln\left(\frac{e^{4S_t\phi} - \alpha}{1 - \alpha}\right) \right\}$$
(15)

The following relation may be employed to investigate the case of temperature-dependent viscosity of air for the computation of dimensionless entropy generation and pumping power ratio:

$$\frac{\mu}{\mu_0} = \left(\frac{T}{T_0}\right)^{3/4} \tag{16}$$

where $\mu_0 = 1.92 \cdot 10^{-5}$ Pa·s at $T_0 = 291$ K:

$$dp = -\frac{32\overline{U}}{D^2} \left[\mu_0 \left(\frac{T}{T_0} \right)^{3/4} \right] dx$$
(17)

$$\dot{S}_{gen} = \dot{m}C_{p} \left\{ \left[\ln \left(\frac{1 - \alpha e^{-4\psi_{1}}}{1 - \alpha} \right) - \alpha \left(1 - e^{-4\psi_{1}} \right) \right] + \frac{32}{3} \frac{\overline{U}\mu_{0}}{\rho C_{p} DS_{t} T_{0}^{3/2}} \cdot \left[\left(1 - \alpha e^{4\psi_{1}} \right)^{3/4} - \left(1 - \alpha \right)^{3/4} + 3\psi_{1} \right] \right\}$$

$$N = \frac{1}{1 - e^{-4\psi_{1}}} \left\{ \left[\ln \left(\frac{1 - \alpha e^{-4\psi_{1}}}{1 - \alpha} \right) - \alpha \left(1 - e^{-4\psi_{1}} \right) \right] + \frac{32}{3} \frac{\lambda}{S_{t}} \left[\left(1 - \alpha e^{4\psi_{1}} \right)^{3/4} - \left(1 - \alpha \right)^{3/4} + 3\psi_{1} \right] \right\}$$

$$(18)$$

$$(18)$$

$$N = \frac{1}{1 - e^{-4\psi_{1}}} \left\{ \left[\ln \left(\frac{1 - \alpha e^{-4\psi_{1}}}{1 - \alpha} \right) - \alpha \left(1 - e^{-4\psi_{1}} \right) \right] + \frac{32}{3} \frac{\lambda}{S_{t}} \left[\left(1 - \alpha e^{4\psi_{1}} \right)^{3/4} - \left(1 - \alpha \right)^{3/4} + 3\psi_{1} \right] \right\}$$

$$(19)$$

where

$$\lambda = \frac{\overline{U}\mu_{0}}{\rho C_{p} D T_{0}^{3/2}}$$

$$P_{r} = \frac{32\phi}{\rho C_{p} D \left(T_{w} - T_{i}\right) \left(1 - e^{-4\psi_{1}}\right)} \left[\mu_{0} \left[\frac{T_{w} \left(1 - e^{-4\psi_{1}}\right)}{T_{0}} \right]^{3/4} \right]$$

$$P_{r} = \frac{32E_{c}\psi_{1}}{\left(1 - e^{-4\psi_{1}}\right)} \xi \left[\frac{T_{w} \left(1 - e^{4\psi_{1}}\right)}{T_{0}} \right]^{3/4}$$
(20)
(21)

(21)

where

$$\xi = \frac{\mu_0 D}{3.66 k \overline{U}}$$

Further details of the aforementioned equations can be found in [30].

Results and discussions

One chip is a dummy element

Figure 2 shows the Nusselt number of Chips 5 and 8 when one of the rest chips in its column (column 2) is a dummy and the rest are heat sources. It is possible to deduce from fig. 2(a) that transferring the dummy block in the identical column from Chip 5 has no effect on the Nusselt number for the three working fluids at all studied Reynolds numbers. On the other hand, regarding the Nusselt number number for Chip 8, the Nusselt number slightly increases as the dummy element is transferred toward Chip 8 at the same Reynolds number. But this effect increases as the operating fluid changes and extreme enhancement is obtained using CO₂. This is attributed to two products, the first one is that as the dummy element is near Chip 8, the chip's temperature decreases: consequently, Nusselt increases. The maximum growth in Chip 5 Nusselt number reaches 10% at Re_L 5000 when using CO₂ instead of air. While for Chip 8, the maximum increase in Nusselt number reaches 20% at Re_L 5000 when using CO₂ instead of air. This is mostly due to an increase in spacing between the focus on chip (Chip 8) and the preceding heat source, which influences the cooling rate. In other words, when the dummy element is far away, Chip 8 achieves its lowest Nusselt number for all Reynolds numbers. This indicates that there should be a gap between the two viewpoint chips for a high cooling rate.



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Figure 2. Variation of one chip Nusselt number with Re_L on the same columns; (a) Chip 5 and (b) Chip 8

Figure 3(a) shows temperature contours in isometric view and flow wise direction view for the case of only Chip 5 is dummy when using CO₂ at Re_L = 19000. While fig. 3(b) shows the temperature change of Chip 8 for the three working fluids when the fake element is moved through its column. One may infer that the temperature of Chip 8 decreases at the same Reynolds number when the fake element is moved from Chip 5 to Chip 8 in the same column. As an example, Chip 8 temperature at Re_L 5000 reached 331.57 K when Chip 5 was a dummy and 307.1 K when chip 8 it self was a dummy while utilizing air. When Chip 5 is a dummy, Chip 8's temperature for CO₂ is 349 K, and when Chip 8 is a dummy, it is 311 K. This is mainly caused by an increase in the distance between the target chip (Chip 8), which impacts the cooling rate, and the prior heat source. In other words, Chip 8 reached its highest temperature for all Reynolds number values when the fake element was distant from it. This indicates that a space between the two viewpoint chips is necessary for a high cooling rate. The velocity contours for the case of Chip 5 is dummy is depicted in fig. 3(c) at Re_L = 19000 when using CO₂.



Figure 3. Sample of temperature results for one chip is dummy; (a) temperature contours, Chip 5 is dummy, Re = 19000 for CO_2 , (b) variation of the Chip 8 temperature with the three working fluids, and (c) velocity contours, Chip 5 is dummy, Re = 19000 for CO_2

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Two chips are dummy elements

Figure (4) illustrates part of the findings as two chips are dummies, and the rest are used as heat sources. It depicts the relationship between Chip 8 and Chip 7 Nusselt number and Re_L. The figure revealed that as the flow inlet velocity increases, the air-flow and re-circulation flow side combination improve, and the Nusselt number number increases linearly for the three working fluids. Furthermore, CO_2 gives a higher value for the two chips. This is attributed to the cooling effect of CO_2 being higher than air. When the input flow velocity rises, the temperatures of the two heat sources drop. At a higher input velocity, fig. (4) shows flow impingement on the second fake element (Chip 6). Figure (5) shows temperature contours from sample cases for two chips are dummies, and the rest chips are heat sources, for the case of using CO_2 at Re = 19000.



Figure 5. Temperature contours sample cases of two chips are dummies, and the rest chips are heat sources, for CO_2 at Re = 19000; (a) Chips 5 and 6 and (b) Chips 6 and 7

Case of one row is a dummy

Figure 6 illustrates the findings when one row is a dummy, and all other chips are heat sources. The figure represents the Nusselt number values for Chip 7 and Chip 8 for the three working fluids at all Re_L. According to fig. 6, the outcomes demonstrate that the trends are the same for all results whenever the dummy row position changes. Moreover, the Nusselt number increases rapidly as the dummy row position is near the target chip. This is attributed to the reduction in chip temperature as the dummy row goes toward the chip. In addition, when the flow input velocity increases, the side blend of air-flow and re-circulation flow improves after the first row. The outcomes revealed an enhancement of 35% for Chip 6 Nusselt number when using CO₂ instead of air when the second row is a dummy. This is attributed to the thermal conductivity of CO₂, which is lower than that of air. Furthermore, the final row temperatures rise because it stays a source term, and the flow transfers heat from prior rows and freezes a little quantity of warmness from it. The increase in distance between the chips in the final row and the preceding heat source (due to forming row 3 as a dummy) influences the cooling rate. Figure 7 shows temperature contours from sample cases of one row is dummy, and the rest chips are heat sources, for CO₂ at Re = 19000.

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Figure 7. Temperature contours from sample cases of one row is dummy, and the rest chips are heat sources, for CO_2 at Re = 19000; (a) first row, (b) second row, and (c) third row

Entropy generation results

Figure 8(a) depicts the impact of the variation of Reynolds number on entropy generation during heat transfer to constant viscosity air-flows in the rectangular duct. A dimensionless parameter is termed an entropy generation number. Since the friction factor, f, reflects the reciprocal of the Reynolds number, the dimensionless entropy generation number, N, defined as the ratio of entropy created to the total heat transfer rate to the duct, is observed to decrease with an increase in Reynolds number. The pumping power to heat transfer rate ratio, defined in fig. 8(b) as the relationship between the parameter Prandtl number and the fluctuation in the Reynolds number, is shown to increase as the Reynolds number rises. This is because the promotion of Reynolds number leads to an increase in the product of flow velocity and equivalent diameter of the rectangular duct, which appears in the numerator of expression obtained to compute the Prandtl number for constant viscosity air-flow in the duct.

Figure 9(a) demonstrates the impact of the change of Reynolds number on entropy generation during heat transfer to temperature-dependent viscosity air-flows in the rectangular duct. Since the viscosity of air is minimal, the quantities of N entropy generation number show decreasing behavior like the constant viscosity case of fig. 8(a). Furthermore, it is shown that the dimensionless entropy generation N estimated for constant viscosity gives somewhat greater values than those found for temperature-dependent viscosity. Figure 9(b) depicts the effect of Reynolds number change on the pumping power to heat transfer rate ratio Prandtl number for the temperature-dependent viscosity example of air-flow. A significant rise in the values of Prandtl number is observed with the more extensive promotion of the Reynolds number, which is found due to an increase in bulk temperature and viscosity. The temperature-dependent viscosity assumption values for Prandtl number are lower than those of Prandtl number obtained for the constant viscosity case.



Figure 8. (a) Impact of the variation of Reynolds number on dimensionless entropy generation rate for constant viscosity air-flow and (b) impact of the variation of Reynolds number on pumping power to heat transfer for constant viscosity air-flow



Figure 9. (a) Impact of the variation of Reynolds number on dimensionless entropy generation for temperature dependent viscosity air-flow and (b) impact of the variation of Reynolds number on pumping power to heat transfer rate temperature dependent viscosity air-flow

Conclusions

This work displays a sizable numerical simulation that was run to look at ways to increase the cooling of false and cubic electronics pieces within a rectangular duct. Furthermore, because of difficulties in executing experiments to obtain low values of Reynolds number, the current study provides CFD simulations with minimal Reynolds number of 500.12 electrical chips (3×4 array) are tested in an air duct across a wide range of Re_L, from 500-19000. The study focuses on the effect of adjusting the cooling air-flow rate and chip spacing on cooling improvement. To do this, the following points are detected while the heat sources are shifted, and the remaining components are used as dummies to retain the flow characteristics.

- Increasing the inflow flow rate lowers the temperatures of the chips for all working fluids.
- The impact of Re_L on the resultant temperature is significantly more significant when $Re_L < 500$, reduced when $5000 < Re_L < 10000$, and minor when $Re_L > 10000$.
- The elements farthest away from the duct inlets had the most excellent temperatures when the dummy element was far from it.
- The cooling rate is adjusted by increasing the space between components in the fluid-flow direction.

- Designers should run such electronic board systems at optimized higher Re_L levels and with CO₂ as an operating fluid.
- An enhancement of 35% for chip 6 Nusselt number when CO₂ is used instead of air while the second row is a dummy.
- Increasing the Reynolds number resulted in decreasing the dimensionless entropy generation number.
- The pumping power ratio is observed to be increasing with the promotion of the Reyniolds number.
- The calculated findings show that dimensionless entropy generation computed for the case of constant viscosity of air provides somewhat higher values than dimensionless entropy generation computed for the case of temperature-dependent viscosity. The analysis carried out using the equations formulated for dimensionless entropy generation number and pumping power ratio is limited to the conditions of laminar flow of fluid through a duct of constant surface temperature.

Nomenclature

- $A_{\rm s}$ heat element surface area, [m²]
- h convection heat transfer
- coefficient, [Wm⁻²K⁻¹]
- k thermal conductivity, [Wm⁻¹K⁻¹]
- L heat source length, [m]
- Q heat transfer rate, [W]
- q heat flux, [W/m²]
- Nu average Nusselt number (= hL/k), [–]
- Re Reynolds number (= vL/v), [–]
- T temperature, [K]
- v mean velocity, [ms⁻¹]

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Greek letter

- v kinematic viscosity, [m²s⁻¹]
- Subscripts
- L based on chip length
- i inlet
- m mean
- net net amount of heat

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