https://doi.org/10.33261/jaaru.2025.32.2.004 Association of Arab Universities Journal of Engineering Sciences (2025)32(2):35–41

Association of Arab Universities Journal of Engineering Sciences

مجلة اتحاد الجامعات العربية للدراسات والبحوث الهندسية

Investigation of Discrete Heating Gap Distance Effect on the Heat Transfer and Fluid Flow Characteristics in a Two Parallel Plates Channel

***Mohammed A. Nima*[1] and *Raed G. Saihood*[2]**

[1] *Department of Mechanical Engineering, University of Baghdad, 10071 Baghdad, Iraq, dralsafi@uobaghdad.edu.iq*

[2] *Department of Mechanical Engineering , University of Baghdad, 10071 Baghdad, Iraq, raed.hassan@coeng.uobaghdad.edu.iq*

**Corresponding author: Mohammed A. Nima and email: dralsafi@uobaghdad.edu.iq*

Published online: 30 June 2025

Abstract— A numerical investigation was conducted to examine the heat transfer performance of two horizontal parallel plates subjected to discrete heating from the upper plate. Water was used as a working fluid, and the gap size between the heating elements was varied. The study analyzed the effects of Reynolds' numbers, heat flux, and heating element position on heat transfer. A finite volume method and a line-by-line tri-diagonal matrix algorithm (TDMA) were employed to solve the governing equations. Results demonstrated a significant enhancement in heat transfer with an increase in gap size between the heating elements. Case 3, featuring the widest gap, exhibited the most favorable heat transfer characteristics. The findings were consistent across different Reynolds numbers and heating values. Additionally, the mean Nusselt number for the heating elements decreased downstream toward the channel exit.

Keywords— Discrete heating; Gap size; Parallel plates; Tri-diagonal matrix algorithm; Heat transfer performance.

1. Introduction

The management of the thermal performance of discrete heat transfer inside a channel received considerable interest because of its importance in many applications, especially in electronic devices. The continuous development in the electronic components specifications was always accompanied by the increase of heat generation rates during their operation which needed a proper design for the heat sink to reduce the operational temperature to an acceptable level to ensure an increase in the lifetime and efficiency.

Air cooling is the most popular method that is used by the researchers for its design and operation simplicity. Among the earliest studies is the one that was conducted by Kennedy and Zebib [8] to study the effect of local heat source configurations variation on the electronic cooling. Mohammed et al. [17] investigated numerically the effect of vortex generator on the heat transfer and fluid flow properties in a rectangular duct. It found that the rise in obstructions height will improve the heat transfer process because it increases the surface area of heat transfer and the disturbance in the flow. The problem of convective cooling was investigated by He et al. [4] in a horizontal duct supplied with discrete heated elements and the effect of different parameters was studied and analyzed. They found that radiation effect is highly increased with smaller Reynolds number and higher surface emissivities.

Pirsaaci and Sivrioglu [13] studied experimentally the heat transfer properties of air flow in a rectangular duct provided with top and bottom discrete heat sources. Their results showed that higher values of aspect ratio have a preferable effect on the secondary flow generation. Hassan et al. [6] studied experimentally the configuration effect of flat plate in the solar collectors' array on the heat transfer process. They noted that the thermal storage energy with series connection was higher than that of parallel connection. Jehhef [7] studied numerically the effect of adding two vertical obstructions with variable height on the flow characteristics. He found that increasing the height of obstructions in increases the kinetic energy of the flow .

Experimental study was conducted by Rosas et al. [14] on an electronic module of two heat sources to examine the effect of adding a curve deflector on the pressure drop and heat transfer performance. They include that effect of size variation of deflector have preferable influence on heat transfer with small vertical distance and opposite effect is determined for small horizontal distance. Miroshnichenko et al. [10] numerically investigated the problem of mixed convection in wavy channel under the condition of localized heating of micropolar fluid. They found that free convection region is highly affected by Rayleigh number, and for lower Prandtl numbers values the heat transfer is reduced for higher secondary flow intensity. Mohebbi et al. [11] conducted a numerical study with the use of second-order lattice Boltzmann method to study the forced convective heat transfer characteristics in a horizontal duct with nanofluid and its surface mounted blocks. They showed that adding nanoparticles and blocks on the walls could increase the average heat transfer coefficient up to 39.04%.

Durgam et al. [2] investigate experimentally and numerically the problem of adding dummy components to discrete heaters that distributed in a vertical duct. The optimal placement of seven heat sources was investigated and arranged in a way to minimize the substrate temperature. Talukdar et al. [16] numerically investigated the problem of natural convection within parallel plates provided with symmetric and staggered discrete heat sources. Their results show that when the staggered array was adopted, the heat transfer increased on the left side while decreased on the right side. Shehab [15] studied experimentally the effect of configurations of fins array on the heat characteristics for forced flow through rectangular duct. It found that the blend of 18% notched and 9% perforated fins have the best improvement in heat transfer coefficient which reaches to 45%. Zhengyu et al. [18] conducted numerical study on partitioned thermal convection using lattice Boltzmann method. They found that the thermal boundary layer is affected by the gap length which have a combined effect with the wall thickness on the shape of the thermal boundary layer distribution. Basim et al. [1] studied numerically and experimentally the heat transfer in a flat plate collector with elliptical-shaped riser and wavy fins. It was noticed that the thermal efficiency for the new model is enhanced by 22.4% compared with conventional model.

This study examines how varying gap sizes between discrete heating elements on two horizontal parallel plates affect heat transfer performance. Unlike earlier studies that focused on fixed configurations, this work aims to identify optimal arrangements that improve heat transfer efficiency. By analyzing different setups, the study explores how convection heat transfer influences the flow field and Nusselt number on the heating elements. This approach offers new perspectives on designing effective thermal management systems.

2. Mathematical Formulation

2.1 *Problem description*

The geometry and the coordinate system of the two horizontal parallel plates subjected to a discrete heating from the upper plate are illustrated in Fig. 1. The lower plate is thermally insulated while the upper plate is supplied with five heating elements of constant heat flux (q_w = 250 and 400 w/m^2) and thickness (t=4 cm) and the gap size between the heating elements is designated as (S) with a Re (700 and 900). The channel length is (L=150 cm) and its width is (W=20 cm). Water come in the channel with constant temperature and a uniform velocity and exit as fully developed flow, the properties of water at 25 ^{O}C are listed in Table (1). Three different patterns are considered in the present work for the heating elements distribution along the upper plate, and these patterns based on fixing the position of the central heating element and changing the position of the other heating elements by increasing the gap size between them as follows:

Case 1: S=t

Case 2: S=3t

Case 3: S=6t

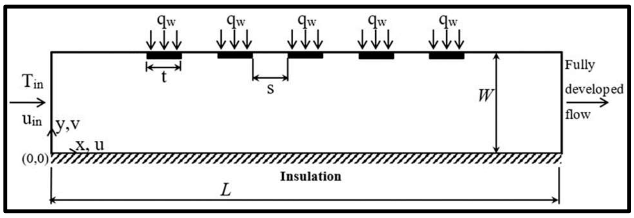


Figure 1: Schematic of the physical problem

As the gap size is increased, the heating elements are pushed towards the channel inlet and exit sections which will affect the fluid flow and heat transfer characteristic in the channel.

Table 1: Thermophysical properties of water at 25 ^{O}C and 1 atm.

Property	Symbol	Value
Density	ρ (kg/m^3)	999.9
Thermal conductivity	k (W/m.K)	0.571
Dynamic viscosity	μ (kg/m.s)	1.519×10^{-3}

2.2 *Assumptions*

The basic assumptions that rule the recent study are as follows:

1) Laminar and incompressible flow .

2) Steady state.

3) Boussinesq approximation is invoked.

4) The flow is two-dimensional.

5) Heat generation and viscous dissipation are neglected.

2.3 *Governing equations*

The governing equations of mass, momentum and energy can be reduced based on the above assumptions to the following: DeWitt [5]

Conservation of mass

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \quad (1)$$

<u>*Conservation of mass*</u>

x-Momentum Equation

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \quad (2)$$

y-Momentum Equation

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+\rho g\beta(T-T_{in}) \quad (3)$$

<u>*Conservation of energy*</u>

$$\rho C_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \quad (4)$$

2.4 *Boundary Conditions*

For the recent problem, the boundary conditions are given as:

<u>At x = 0 (inlet)</u>

$$u=u_{in} \quad ,v=0 \quad (5)$$

$$T=T_{in} \quad (6)$$

<u>At x = L (exit)</u>

$$\frac{\partial u}{\partial x}=0 \quad ,v=0 \quad ,\frac{\partial T}{\partial x}=0 \quad (7)$$

<u>At y = 0 (adiabatic wall)</u>

$$u=v=0 \quad (8)$$

$$\frac{\partial T}{\partial y}=0 \quad (9)$$

<u>At y = W (heated wall)</u>

$$u=v=0 \quad (10)$$

$$\frac{\partial T}{\partial y}=\begin{cases}-\frac{q_w}{k} & under\ the\ heated\ elements\\ 0 & elsewhere\end{cases} \quad (11)$$

2.5 *Further Calculations*

The Reynolds number can be defined as:

$$Re=\frac{\rho u_{in} W}{\mu} \quad (12)$$

The local Nusselt number under each heating element can be defined as:

$$h_x=q_w/\left(T_{w,x}-T_{b,x}\right) \quad (13)$$

$$Nu_x=\frac{h_x W}{k}=\frac{q_w W}{k\left(T_{w,x}-T_{b,x}\right)} \quad (14)$$

The mean Nusselt number at each heated element is calculated as follows:

$$Nu_{mi}=\frac{1}{t}\int_{x_i}^{x_i+t} Nu\, dx \quad (15)$$

where x_i is the location of the heating element i from the channel inlet.

The global Nusselt number can be written as:

$$Nu_g=\frac{\sum_{i=1}^{i=N} Nu_{mi}}{N} \quad (16)$$

where N is the number of heating elements in the duct.

2.6 *Numerical Procedure*

In the recent study, equations (8) and (9) are connected using the SIMPLE algorithm of Patankar [10] and are solved using the finite volume method with line-by-line tri-diagonal matrix algorithm TDMA. Fully implicit control volume based in finite difference formulation is used to solve the energy equation (4) in which the combined convective and conductive terms are discretized with the use of the power law scheme of Patankar [12].

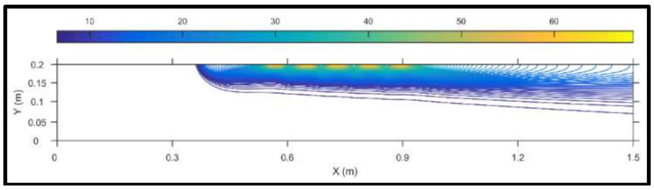
A relative error in velocity and temperature is less than 10^{-5} between successive iterations to obtain a good convergence, while it is less than 10^{-7} to achieve convergence the pressure field. Ferziger & Peric [3]

$$\bar{R}=\frac{\sum\left|a_e\phi_e+a_w\phi_w+a_n\phi_n+a_s\phi_s+b-a_p\phi_p\right|}{\sum\left|a_p\phi_p\right|}\leq 10^{-7} \quad (17)$$

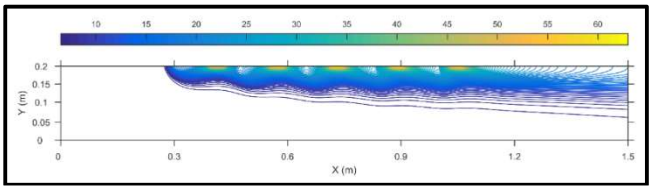
3. Results and Discussion

3.1 *Temperature Contours*

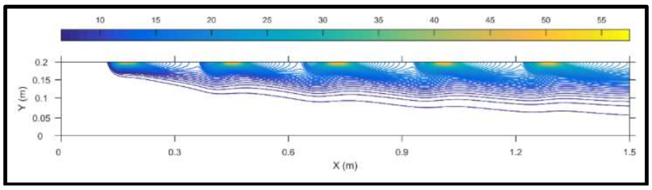
Fig. 2 shows how changing the spacing size between heating elements impacts temperature distribution in the duct when qw=400 W/m² and Re=700. A wider gap expands the thermal boundary layer, yet fluid temperature drops due to higher buoyancy. Heat is transmitted to the fluid by conduction between the heating elements and plates, and increased thermal conductivity increases efficiency. The temperature gradient, flow velocity, and gap size all have an impact on forced convection, which transfers heat from the plate to the fluid. Larger gaps improve heat transfer by improving flow distribution and raising the Nusselt number, whereas a smaller thermal boundary layer accelerates heat transfer.



(a) Case 1



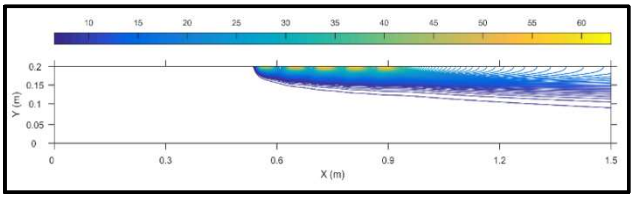
(b) Case 2



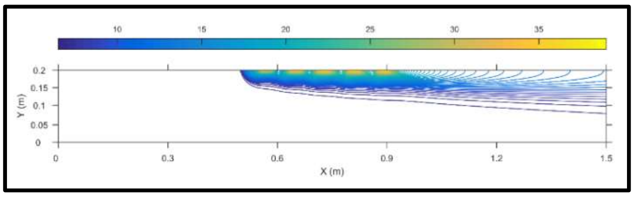
(c) Case 3

Figure 2: Temperature contours in °C for qw=400 W/m^2, Re=700

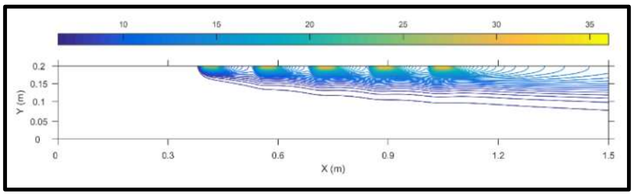
Figs. 3 and 4 show the influence of gap size variation between the heating elements on the temperature contours along the channel for different heat flux and Reynolds numbers. The same be haviour can be detected as that in Fig. 2.



(a) Case 1

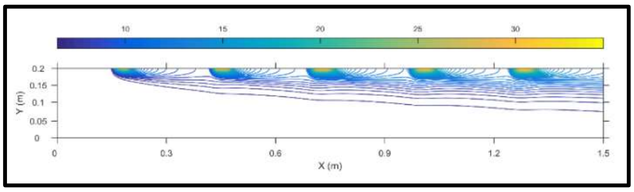


(b) Case 2

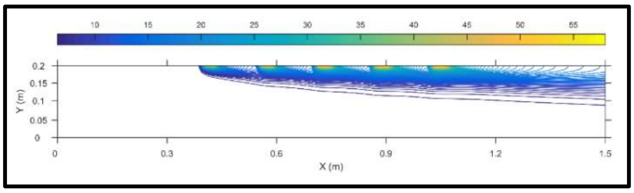


(c) Case 3

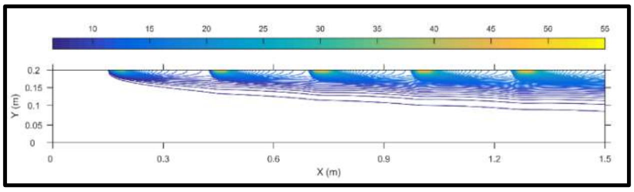
Figure 3: Temperature contours in °C for q_w=250 W/m^2, Re=700



(a) Case 1



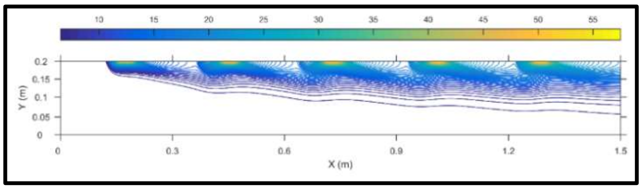
(b) Case 2



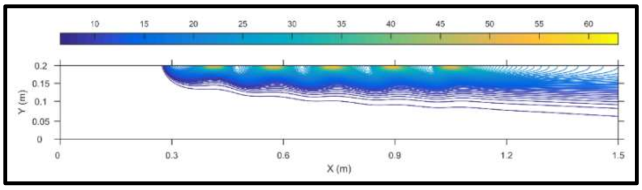
(c) Case 3

Figure 4: Temperature contours in °C for q_w=250 W/m^2, Re=900

Fig. 5 shows the impact of heat flux change on temperature distribution along the duct for Case 2, Re=700. It is clear that as the heat flux is rised, the buoyancy force improves and causes a rapid development in the thermal boundary layer accompanied by higher fluid temperature values.



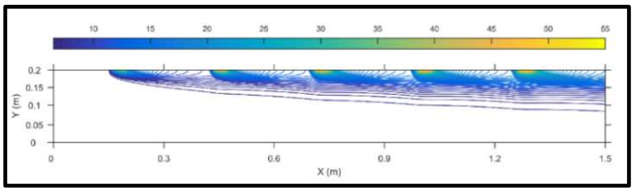
(a) q_w = 250 W/m^2



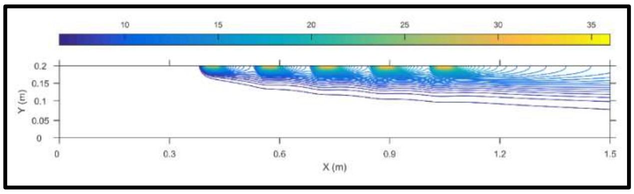
(b) q_w = 400 W/m^2

Figure 5: Temperature contours in °C for Case 2, Re=700

Fig. 6 illustrates the influence of Reynolds numbers change on temperature distribution along the duct for Case 3, q_w=400 W/m^2. It is clear that as the Reynolds numbers is raised, the thermal boundary layer decline throughout the channel and fluid temperature is reduced.



(a) Re=700



(b) Re=900

Figure 6: Temperature contours in °C for Case 3, q_w=400W/m^2

3.2 *Temperature Distrbution*

Fig. 7 shows the influence of gap size variation between the heating elements on the upper plate wall temperature for q_w=400 W/m^2, Re=900. It obvious that the wall temperature values start to increase earlier in Case3 compared with that in Case 1 as the heating elements positions are changed towards inlet and exit sections. Another observation can be made regarding the maximum wall temperature value for the two cases, the wall temperature for Case1 is higher than that of Case 3 by 10 ^{O}C. this can be attributed to the fact that when the gap size between the heating elements is small the thermal boundary layer thickness is higher and caused higher temperature values.

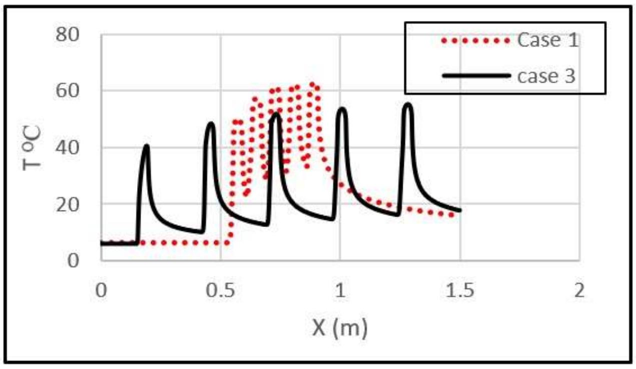


Figure 7: Wall Temperature in °C for q_w=400 W/m^2, Re=900

Fig. 8 shows the influence of gap size variation between the heating elements on the bulk temperature for q_w=400 W/m^2, Re=900. It can be seen that for Case 1 there is a rapid increase in the bulk temperature values in the heating region and a smooth increased for Case 3.

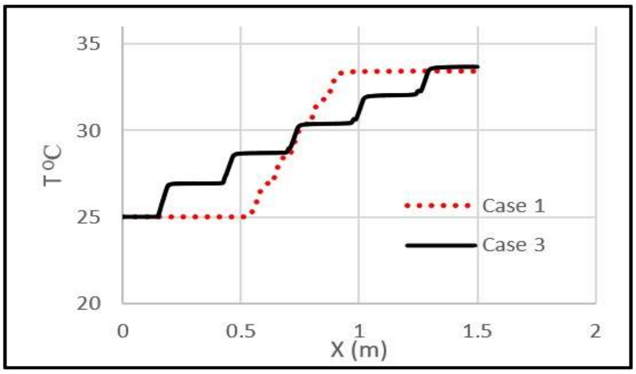


Figure 8: Bulk Temperature in °C for q_w=400 W/m^2, Re=900

3.3 *Mean and global Nusselt Number*

Fig. 9 shows the influence of gap size variation between the heating elements on the mean Nusselt Number values for q_w=400 W/m^2, Re=900. A general behavior can be detected for the mean Nusselt Number values for the three cases, the maximum value of the mean Nusselt Number for each case is at the inlet section and decreases towards the exit section. This behavior indicates that the heat transfer decreased towards the exit section as decrease the difference between the temperature of the warm surface and the water temperature towards the exit section. Fig. 9 also indicates that the values of mean Nusselt Number are higher when the gap size increased from Case1 to Case 3 and this gives evidence that distribution of the discrete

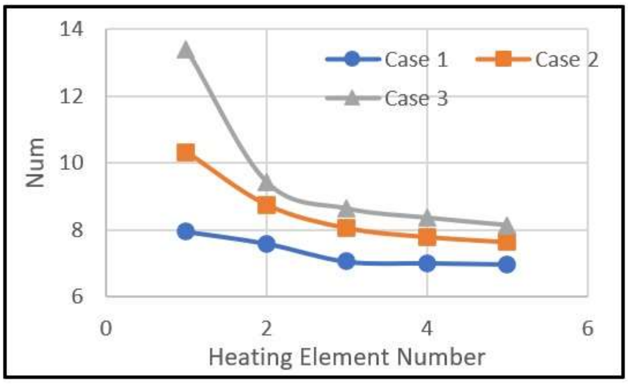


Figure 9: Mean Nusselt Number for q_w=400 W/m^2, Re=900

Fig. 10 shows the influence of gap size variation between the heating elements on the global Nusselt Number values for q_w=400 W/m^2, Re=900. In agreement with the results obtained in Fig. 9, the global Nusselt Number values are higher near the inlet section and decreased towards the exit section.

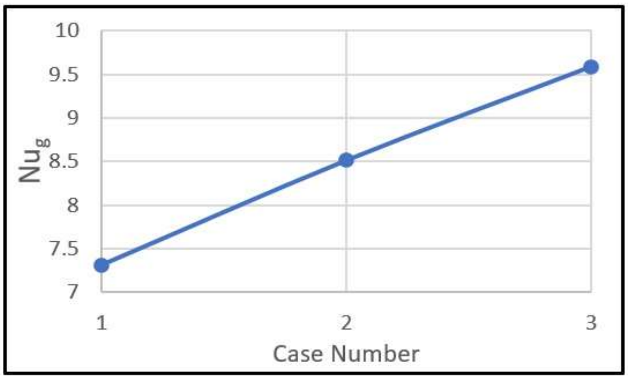


Figure 10: Global Nusselt Number for q_w=400 W/m^2, Re=900

4. *Comparison with Previous work*

From the literature, there is no such close case to compare with. However, a qualitative comparison with Maxime et al. [9] can be done. As shown in Fig. 11, similar behavior can be detected as the thermal boundary layer is spread through the domain based on the heating element position.

5. Conclusions

The current study investigated the influence of gap size between discrete heating components on heat transfer performance in a channel formed by two parallel plates. The main conclusions can be summarized as follows:

- The findings show that as the gap size reduces, the thermal boundary layer thickness increases, resulting in greater wall temperatures. Specifically, Case 1, with the shortest gap size, showed a rapid rise in bulk temperature values, roughly 20% greater than Case 3, which had the biggest gap size.

- The mean Nusselt number values were found to be highest in the inlet part and decreased towards the exit section in all three cases. Notably, the average Nusselt number for Case 3 was almost 15% higher than that of Case 1, suggesting that wider gap sizes greatly improve heat transmission performance.

- The increase in bulk temperature values is notably sharp for Case 1, which features the smallest gap size. Specifically, this case shows an increase of approximately 25% in bulk temperature compared to the larger gap sizes.

- The global Nusselt number is found to be highest in Case 3, which has the largest gap size, and lowest in Case 1, which has the smallest gap. Specifically, Case 3 has a global Nusselt value that is around 30% greater than Case 1.

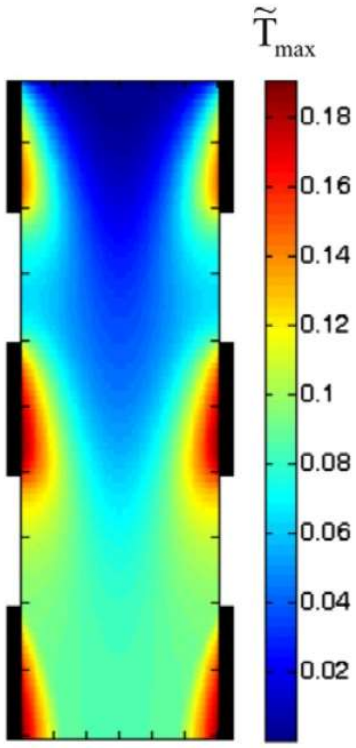


Figure 11: Temperature contours, Maxime et al. [9]

References

[1] Basim Freegah, Mohammed Hussien Alkhafaji, Mohammed Hamed Alhamdo, "Study the thermal response of a solar flat-plate collector under transient solar radiation experimentally and numerically", Journal of Engineering Research, (2024), https://doi.org/10.1016/j.jer.2024.03.004.

[2] Durgam S., Venkateshan S.P., Sundararajan T., A novel concept of discrete heat source array with dummy components cooled by forced convection in a vertical channel. Applied Thermal Engineering. 129, pp. 979–994, (2018).

[3] Ferziger, Joel H.; Perić, Milovan: Computational methods for fluid dynamics. Berlin: Springer (2002).

[4] He J., Liu L., Jacobi A.M., Conjugate Thermal Analysis of Air-Cooled Discrete Flush-Mounted Heat Sources in a Horizontal Channel. J. Electron. Packag. 133, 041001 (8 pages), (2011).

[5] Incropera, F.P., DeWitt, D.P. Fundamentals of heat and mass transfer, Seventh edition. Chichester: John Wiley and Sons (2011).

[6] Jafar Mehdi Hassan, Qussai Jihad Abdul-Ghafour, and Mohammed Fowzi Mohammed, "Experimental Study of the Thermal Performance of Flat Plate Solar Collectors Array by Different Connection Configurations", Journal of Engineering, Vol. 21, No. 5, p.p. 55-71, (2015).

[7] Kadhum Audaa Jehhef, "Numerical Simulation of Flow in Rectangular Duct with Different Obstruction Heights", Journal of Engineering, Vol. 21, No. 11, p.p. 94-110, (2015).

[8] Kennedy K.J., Zebib A., Combined free and forced convection between horizontal parallel planes: some case studies. Int. J. Heat Mass Transfer 26, pp. 471–474, (1983).

[9] Maxime Tye-Gingras, Louis Gosselin, Alexandre K. da Silva, "Synchronizing time-varying discrete heating elements in forced convection for minimal thermal resistance", International Journal of Heat and Mass Transfer, Vol. 53, p.p. 4917- 4926, (2010).

[10] Miroshnichenko I.V., Sheremet M.A., Pop I., Ishak A., Convective heat transfer of micropolar fluid in a horizontal wavy channel under the local heating. Int. J. of Mechanical Sciences. 128-129, pp. 541–549, (2017).

[11] Mohebbi R., Lakzayi H., Sidik N.A.C., Japar W.M.A.A., Lattice Boltzmann method based study of the heat transfer augmentation associated with Cu/water nanofluid in a channel with surface mounted blocks. Int. J. Heat Mass Transfer. 117, pp. 425-435, (2018).

[12] Patankar SV., "Numerical heat transfer and fluid flow", New York: McGraw-Hill; (1980).

[13] Pirasaci T., Sivrioglu M., Experimental investigation of laminar mixed convection heat transfer from arrays of protruded heat sources. Energy Conversion and Management 52, pp. 2056–2063, (2011).

[14] Rosas A.S., Ali R.K., Abdel-Aziz A.A., Elshazly K.M., An experimental investigation of convective heat transfer enhancement in electronic module using curved deflector. Heat and Mass Transfer. 53, pp. 985–994, (2017).

[15] Saad Najeeb Shehab, "Experimental Study of Forced- Convection from Horizontal Rectangular Fins Array into Air Duct", Al-Khwarizmi

Engineering Journal, Vol. 15, No. 1, p.p. 35- 45, (2019).

[16] Talukdar D., Li C.-G., Tsubokura M., Investigation of compressible laminar natural-convection for a staggered and symmetric arrangement of discrete heat sources in an open-ended vertical channel. Numerical Heat Transfer, Part A. 76, pp. 115–138, (2019).

[17] Waheed S. Mohammed, Sabah Tarik Ahmed, and Laith J.H., “The Effects of Vortex Generator Types on Heat Transfer and Flow Structure in a Rectangular Duct Flows”, Al-Khwarizmi Engineering Journal, Vol.4 , No.1 , pp 27-47, (2008).

[18] Zhengyu Wang, Huilin Tong, Zhengdao Wang, Hui Yang, Yikun Wei, Yuehong Qian, Effect of Gap Length and Partition Thickness on Thermal Boundary Layer in Thermal Convection. Entropy. 25(2), 386, (2023).

Nomenclature

Cp	Specific heat capacity, J/kg.K
g	Gravitional acceleration, m/s^2
h	Heat transfer coefficient, W/m^2.K
k	Thermal Conductivity, W/m.K
L	Channel length, m
Nu	Nusselt number
p	Pressure, Pa
q	Constant heat flux, W/m^2
Re	Reynolds number
S	Gap size between the heating elements, m
t	Thickness of heating elements, m
T	Temperature, ^{O}C
u,v	Velocity in x and y direction, m/s
W	Channel width, m

Greek symbols

β	Volumetric Coefficient of Thermal Expansion, 1/K
μ	Viscocity, Pa.s
ρ	Density, kg/m^3

Subscripts

g	Global
in	Inlet
w	Wall

دراسة تأثير مسافة الفجوة بين عناصر التسخين المنفصلة على انتقال الحرارة وخصائص تدفق السوائل في قناة ذات لوحين متوازيين

محمد عبد الرؤوف نعمة[1] ، رائد كاطع صيهود[2]

[1] قسم الهندسة الميكانيكية ، جامعة بغداد ، 10071 بغداد ، العراق ، *dralsafi@uobaghdad.edu.iq*
[2] قسم الهندسة الميكانيكية ، جامعة بغداد ، 10071 بغداد ، العراق ، *raed.hassan@coeng.uobaghdad.edu.iq*
الباحث الممثل : محمد عبد الرؤوف نعمة ، dralsafi@uobaghdad.edu.iq
نشر في : 30 حزيران 2025

الخلاصة - تم إجراء دراسة عددية لفحص أداء انتقال الحرارة بين لوحين أفقيين متوازيين يتعرضان لتسخين متقطع من اللوح العلوي. تم استخدام الماء كسائل عامل، وتم تغيير حجم الفجوة بين عناصر التسخين. تناولت الدراسة تأثير أرقام رينولدز وتدفق الحرارة وموقع عنصر التسخين على انتقال الحرارة. تم استخدام طريقة الحجم المحدود وخوارزمية خط بخط للمصفوفة الثلاثية القطرية (TDMA) لحل المعادلات الحاكمة. أظهرت النتائج تحسناً كبيراً في انتقال الحرارة مع زيادة حجم الفجوة بين عناصر التسخين. وأظهرت الحالة الثالثة، التي تتميز بأوسع فجوة، أفضل خصائص انتقال للحرارة. كانت النتائج متسقة عبر مختلف أرقام رينولدز وقيم التسخين. بالإضافة إلى ذلك، انخفض متوسط رقم نوسلت لعناصر التسخين كلما اتجهنا نحو مخرج القناة.
الكلمات الرئيسية – التسخين المنفصل ، حجم الفجوة ، لوحات متوازية