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# Numerical study and geometrical investigation of the position of alternated fins mounted in channels subjected to forced convective flows

## Estudo numérico e investigação geométrica da posição de aletas alternadas inseridas em canais sujeitos a escoamentos com convecção forçada

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## ABSTRACT

The present numerical study aims to investigate the geometry of an arrangement of isothermal rectangular fins, mounted in alternating form in the lower and upper surfaces of microchannels, subjected to laminar, incompressible, and forced convective flows in a two-dimensional domain. The main purpose is maximizing the efficiency and the heat transfer rate among the fins and the fresh flow in the channel, and minimizing the pressure drop in the finned channel. It is investigated here one degree of freedom ($L_1/L$) that represents the distance between the first and second fins of the arrangement. The remaining degrees of freedom are kept constant ($L_2/L$ and $L_3/L$). All numerical simulations were performed considering air as a working fluid for Reynolds and Prandtl numbers of $Re_H$ = 100 and $Pr$ = 0.71. The best thermal performance of the arrangement was achieved for the extreme ratios of $L_1/L$, i.e., when the second fin was mounted near the first or third fins. A small advantage is noticed for the lowest magnitude of $L_1/L$ in comparison with the other extreme position. The difference between the best and worst thermal performance, measured by the heat transfer rate, was nearly 23%. For the pressure drop, an intermediate ratio $L_1/L$ = 0.4 was conducted to get the best fluid dynamic performance.

**Keywords**: Forced convection; Finned channels; Geometrical investigation; Computational modeling

## RESUMO

O presente estudo numérico busca investigar a geometria de um arranjo de aletas aquecidas, retangulares, com configuração alternada, montadas em microcanais bidimensionais sujeitos a escoamentos refrigerados de ar com convecção forçada, incompressíveis e laminares. O propósito do problema é maximizar a eficiência e a taxa de transferência de calor entre as aletas e o escoamento refrigerado e minimizar a perda de pressão no canal com aletas. No presente estudo é investigado um grau de liberdade ($L_1/L$) que representa a distância entre a primeira e segunda aletas do arranjo. Os demais graus de liberdade são mantidos constantes ($L_2/L$ e $L_3/L$ ). Todas as simulações numéricas foram realizadas para um escoamento de ar com números de Reynolds e Prandtl constantes, $Re_H$ = 100 e $Pr$ = 0,71. Os melhores desempenhos térmicos para o arranjo foram obtidos para as razões extremas de $L_1/L$, ou seja, quando a segunda aleta foi inserida próxima a primeira ou terceira aletas, com uma pequena vantagem para a menor razão de $L_1/L$ em comparação com o extremo superior. A diferença na performance térmica entre a melhor e pior configurações foi de aproximadamente 23% para a taxa de transferência de calor. Para a perda de pressão, a razão intermediária $L_1/L$ = 0,4 conduziu ao melhor desempenho fluidodinâmico.

**Palavras-chave**: Convecção forçada; Canais aletados; Investigação geométrica; Modelagem computacional

# 1 INTRODUCTION

Microchannel heat exchangers (MHE) are commonly used to obtain high magnitudes of heat transfer per unit volume. Recently, the manufacturing of the heat exchangers has been improved, conducting to the reduction of weight and size of the devices. This aspect associated with the decrease of the manufacturing costs are leading to the use of MHE in several applications as cooling of electronic packaging, cooling of automotive, aerospace, and naval engines, and commercial refrigeration (Chen & Ding, 2015; Naqiuddin et al., 2018).

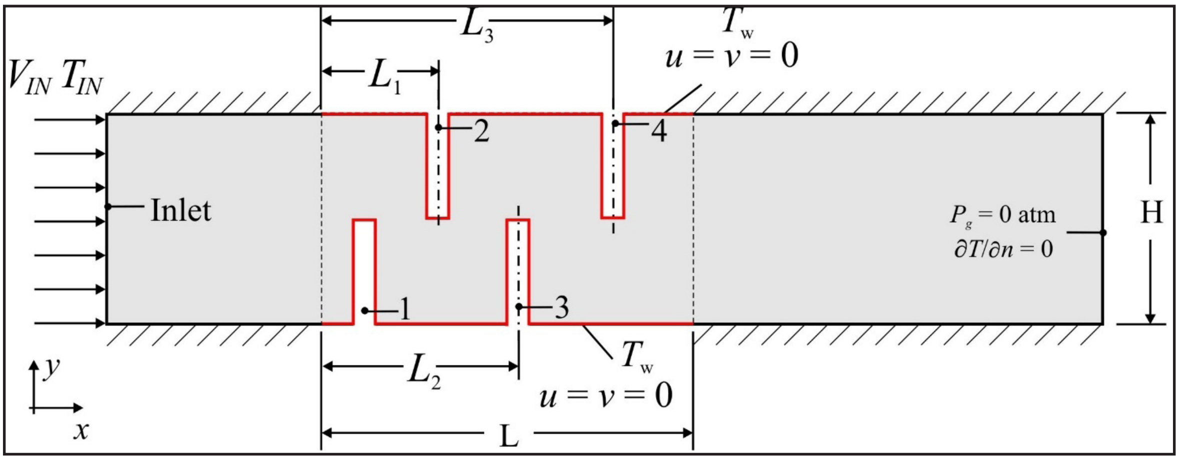
Consequently, the investigation about the influence of the design in thermal and fluid dynamic of the equipment gained even more attention. The importance of the design in microchannels have been widely recognized in the literature for various conditions and configurations. For example, Bello-Ochende et al. (2007) investigated the influence of the aspect ratio of channels subjected to laminar and forced convective flows in a three-dimensional domain using the constructal design (CD) method. Later, Feijó et al. (2018) investigated the influence of the geometrical configuration of two rectangular alternated blocks mounted in channels subjected to laminar and forced convective flows in two-dimensional domain. Shahsavar et al. (2021) investigated the influence of different channel configurations (sinusoidal, trapezoidal, and triangular) on Nusselt number and pressure drop for non-Newtonian, nanofluid, and forced convective flows. Feijó et al. (2022a) performed an investigation of the geometric configuration of two alternated trapezoidal corrugations in channels subjected to boiling convective and laminar flows. Feijó et al. (2022b) performed a similar investigation of Feijó et al. (2018), however considering forced convective turbulent flows. Recently, Santos, Franco and Junqueira (2024) investigated non-Newtonian natural convection flows within open cavities with various square conductive blocks distributed in a uniform way into the cavity.

One of the strategies employed to augment the thermal performance, as seen above, is the use of fins mounted in channel surfaces. Despite of the several performed studies, the geometrical investigation of multiple fins mounted in the channel surfaces has been restricted to two fins or blocks, or multiple fixed obstacles. Therefore, the present numerical study aims to use the computational modeling to investigate the influence of four rectangular isothermal fins mounted in alternate way in channels of subjected to laminar forced convective air flows over thermal and fluid dynamic performance of the system. Here, it is analyzed the influence of one degree of freedom ($L_1/L$) over the fluid dynamics and thermal performance, keeping the remaining ratios ($L_2/L$ and $L_3/L$) constant.

## 2 MATHEMATICAL AND NUMERICAL MODELING

Figure 1 illustrates the computational domain of the studied problem. It is considered incompressible, laminar, steady, and forced convective flows in a two-dimensional domain. The driven flow is caused by the imposition of a constant velocity ($V_{IN}$) at the left surface of the channel. At the exit of the channel (right lateral surface) it is imposed a null gauge pressure ($P_g$ = 0 atm). In the remaining surfaces it is considered a no-slip and impermeability boundary condition ($u = v = 0$ m/s). Regarding the heat transfer in the problem, it is caused by a temperature difference between the isothermal surfaces ($T_W$) indicated in red lines in Fig. 1, and the fresh flow imposed at the inlet of the channel ($T_{IN}$). For all cases, it is considered a temperature difference of $\Delta T$ = 10 K. For the dashed lines of the domain, an adiabatic boundary condition is imposed. Moreover, at the exit of the domain, a null temperature gradient in the flow direction is considered. For all cases the Reynolds and Prandtl numbers are constant ($Re_H$ = 100 and $Pr$ = 0.71).

Figure 1 – Computational domain of the channel with four rectangular alternated fins subjected to forced convective flows




Source: Authors (2024)

For all performed simulations, considering the imposed simplifications, the physical problem is modeled by the conservation of mass, balance of momentum and conservation of energy, given by (Schlichting & Gersten, 2003; Bejan, 2013):

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho}\frac{\partial P}{\partial x} + v\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = -\frac{1}{\rho}\frac{\partial P}{\partial y} + v\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) \tag{3}$$

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{4}$$

where *x* and *y* represent the spatial coordinates (m); *u* and *v* are the velocity components in the *x* and *y* directions (m/s); *ρ* is the density of the fluid (kg/m$^3$); *P* is the pressure (Pa); *υ* is the kinematic viscosity (m$^2$/s); and α is the thermal diffusivity of the fluid (m/s$^2$).

The geometrical investigation performed here consists of the evaluation of the effect of the ratio $L_1/L$, in the range $0.25 \le L_1/L \le 0.5$, over the heat transfer rate (*q*), the efficiency of the finned system (*ε*) and the pressure drop in the channel (*ΔP*). In this study, the remaining degrees of freedom are kept constant: $L_2/L = 0.625$ and $L_3/L = 0.875$.

The performance indicators are given by:

$$q = hA(T_W - T_{IN}) \tag{5}$$

$$\varepsilon(\%) = 100 \cdot \frac{q}{\dot{m}C_P(T_W - T_{IN})} \tag{6}$$

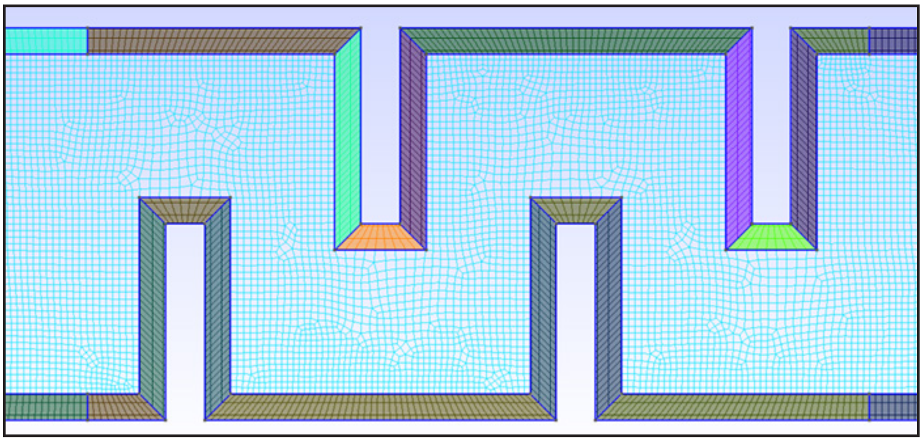
$$\Delta P = (P_{IN} - P_{OUT}) \tag{7}$$

where *h* is the convective heat transfer coefficient (W/m$^2$K) between the isothermal walls and the fresh surrounding flow,  is the mass flow rate of the flow (kg/ms), $C_P$ is the specific heat at constant pressure (J/kgK), $P_{IN}$ and $P_{OUT}$ are the mean pressures at the inlet and exit surfaces of the channel (N/m$^2$).

Concerning the numerical approach of the problem, for the solution of Eqs. (1) – (4) it is employed the finite volume method (FVM) (Patankar, 1980; Versteeg & Malalasekera, 2007) in a code implemented in the commercial software FLUENT (version 2023 R1) (Ansys, 2021). In the computational model, a pressure-based solver is employed, the SIMPLE method is used to model the pressure-velocity coupling, SOU (second order upwind) advection scheme is employed for solving the advective terms, and the standard method is used for the spatial discretization of pressure field. The convergence of the numerical solution for each case is achieved when the residuals for the conservation equations of mass, momentum and conservation of energy are less than $R_{mass} < 1.0 \times 10^{-6}$, $R_{momentum} < 1.0 \times 10^{-6}$, and $R_{energy} < 1.0 \times 10^{-8}$. All simulations are realized using a computer with a processor i7 5820K @ 3.30 GHz, and memory of 16 GB of RAM.

The computational domain was discretized with 79,782 rectangular finite volumes after the realization of a grid independence study. The grid was constructed using the software GMSH (Geuzaine & Remacle, 2009). For a better capture of velocity and temperature gradients near the walls, a subregion was created around the walls to augment the refinement of the mesh in this region and to improve the structure of the volumes. Figure 2 illustrates the strategy used to refine the grid in the near wall region for a coarse mesh.

Figure 2 – Example of grid performed in the software GMSH with a region of refinement for capturing of velocity and temperature gradients



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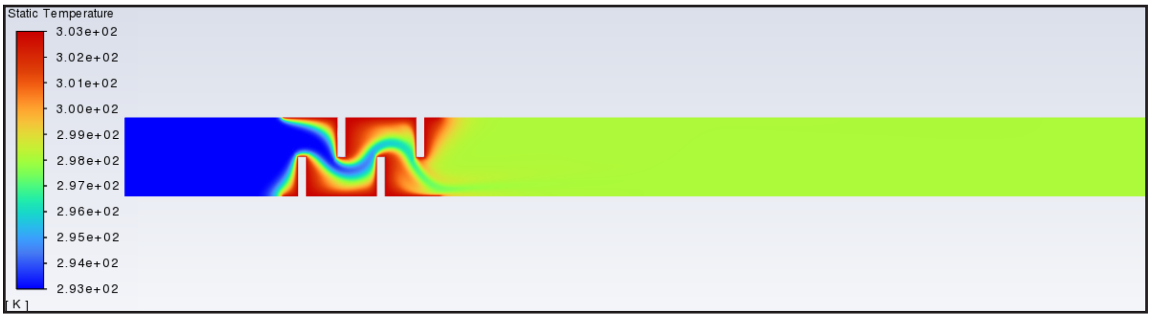
For the verification of the computational model, the same case used in the verification of the work of Feijó et al. (2018) was simulated here. More precisely, a forced convective flow over a bluff body with $Re_D$ = 60 (being $D$ the size of the bluff body) and $Pr$ = 1 was simulated and results of the Nusselt number are compared with previous findings of Sahu, Chhabra and Eswaran (2009). The results obtained here agreed very well with the previous findings of literature, with differences lower than 0.5 % for prediction of the Nusselt number.

## 3 RESULTS AND DISCUSSIONS

Initially, it is presented the temperature and pressure fields for different configurations adopted in this study, highlighting the influence of the geometry on the fluid dynamic and thermal behavior of the channel with fins. Later, due to the interest of this study to obtain the highest heat transfer rate and efficiency, as well as, minimizing the pressure drop (multiobjective problem) the effect of the ratio $L_1/L$ over $q$, $\varepsilon$ and $\Delta P$ are presented.

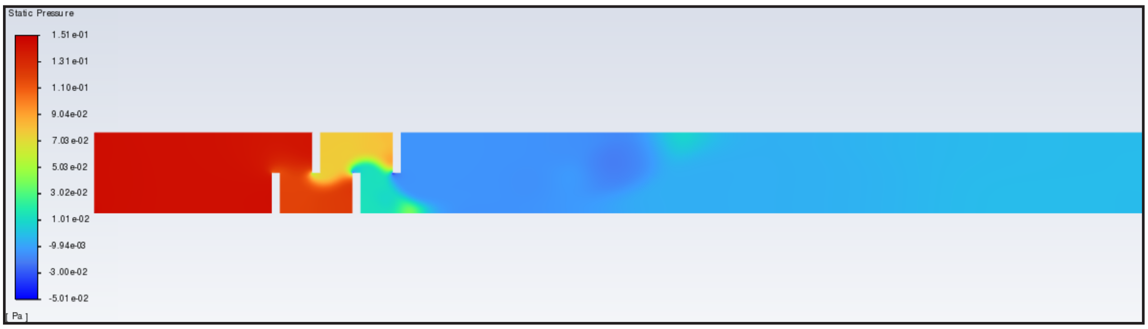
Figures 3 and 4 illustrate, respectively, the thermal and pressure fields for the symmetrical arrangement of fins with $L_1/L$ = 0.375, where it is obtained a pressure drop of $\Delta P$ = 0.144 Pa, a heat transfer rate of $q$ = 9.52 W, and an efficiency of $\varepsilon$ = 51.48 %. Figure 3 shows that the most intense temperature gradients occurred in the upstream surfaces of fins 1 and 2. Meanwhile, the fins 3 and 4 are impinged by a heated stream of fluid flow. For pressure field, Fig. 4, the decrease of pressure magnitude is more pronounced in the region between fins 2 and 3.

Figure 3 – Temperature field for a symmetrical arrangement with $L_1/L$ = 0.375




Source: Authors (2024)

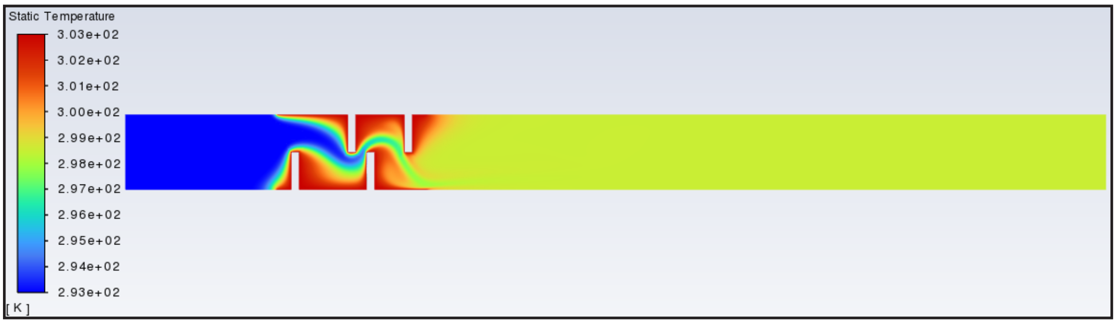
Figure 4 – Pressure field for a symmetrical arrangement with $L_1/L$ = 0.375




Source: Authors (2024)

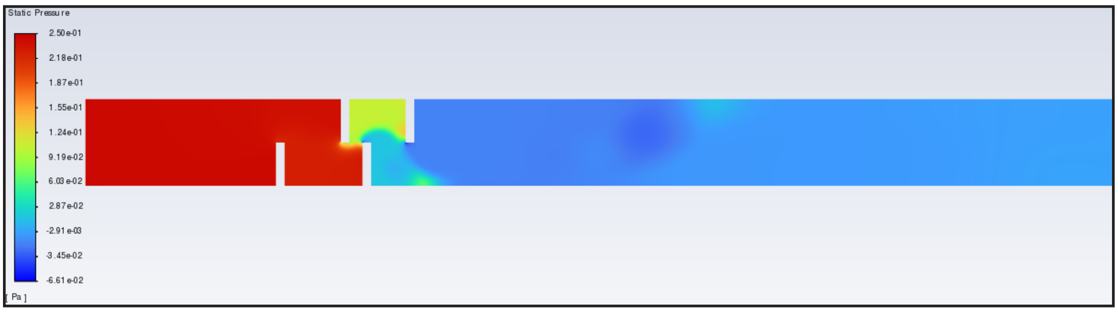
For the ratio $L_1/L$ = 0.5, it is possible to notice for the temperature field (Fig. 5) that the region downstream of the fins has a higher magnitude of temperature in comparison with the case $L_1/L$ = 0.375. The thermal fields for both cases demonstrate that, despite of the occurrence of an increase in the thickness of thermal boundary layer between the first and second fins for the case $L_1/L$ = 0.5, which could led to a decrease of heat exchange in this region, it is also observed an increase of momentum between fins 2 and 3. The latter effect intensifies the heat exchange between the isothermal walls and the fluid flow, mainly in the region between fins 2 and 3, conducting to a higher heat transfer rate ($q$) for $L_1/L$ = 0.5 in comparison with the previous case ($L_1/L$ = 0.375). For the case $L_1/L$ = 0.5, it is achieved a heat transfer rate of $q$ = 10.28 W and an efficiency of $\varepsilon$ = 55.61 %. Figure 6 shows the pressure field for the arrangement with $L_1/L$ = 0.5, being observed an increase of pressure drop in comparison with the case with $L_1/L$ = 0.375, and leading to a pressure drop of $\Delta P$ = 0.24 Pa.

Figure 5 – Temperature field for the superior extreme of $L_1/L$ = 0.5




Source: Authors (2024)

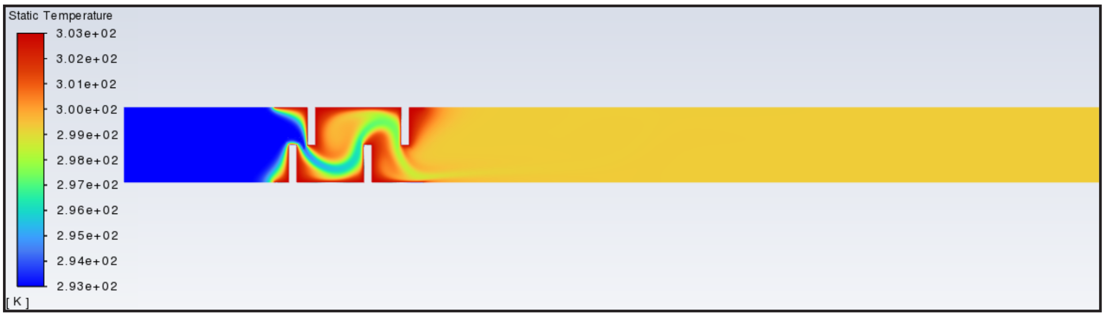
Figure 6 – Pressure field for the superior extreme of $L_1/L$ = 0.5




Source: Authors (2024)

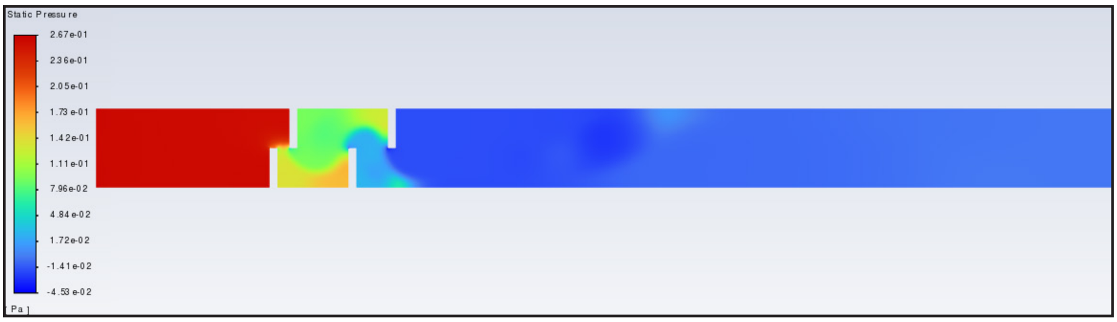
Figure 7 shows the temperature field for the ratio $L_1/L$ = 0.25, i.e., the lowest extreme of the ratio $L_1/L$. It is possible to observe that, the thickness of the thermal boundary layer upstream of fins 1 and 2 are straighter in comparison with the fields for $L_1/L$ = 0.375 (Fig. 3) and mainly $L_1/L$ = 0.5 (Fig. 5), indicating stronger temperature gradients in the frontal fins of the arrangement. This behavior associated with the augmentation of the momentum between the fins 1 and 2 conduct to an increase of the heat transfer rate in the arrangement, which is attested with the magnitudes of $q$ = 11.69 W and $\varepsilon$ = 63.19 % obtained for the case with $L_1/L$ = 0.25. Concerning the pressure field, Fig. 8, it is possible to observe a strong pressure drop ($\Delta P$ = 0.26 Pa) in comparison with the case ($L_1/L$ = 0.375). In comparison with the other asymmetric configuration ($L_1/L$ = 0.5) the difference in pressure drop is not significant. The sole difference is that for $L_1/L$ = 0.25 the strong pressure drop happens between fins 1 and 2, while for $L_1/L$ = 0.5 the occurrence is noticed between fins 2 and 3.

Figure 7 – Temperature field for the inferior extreme of $L_1/L$ = 0.5




Source: Authors (2024)

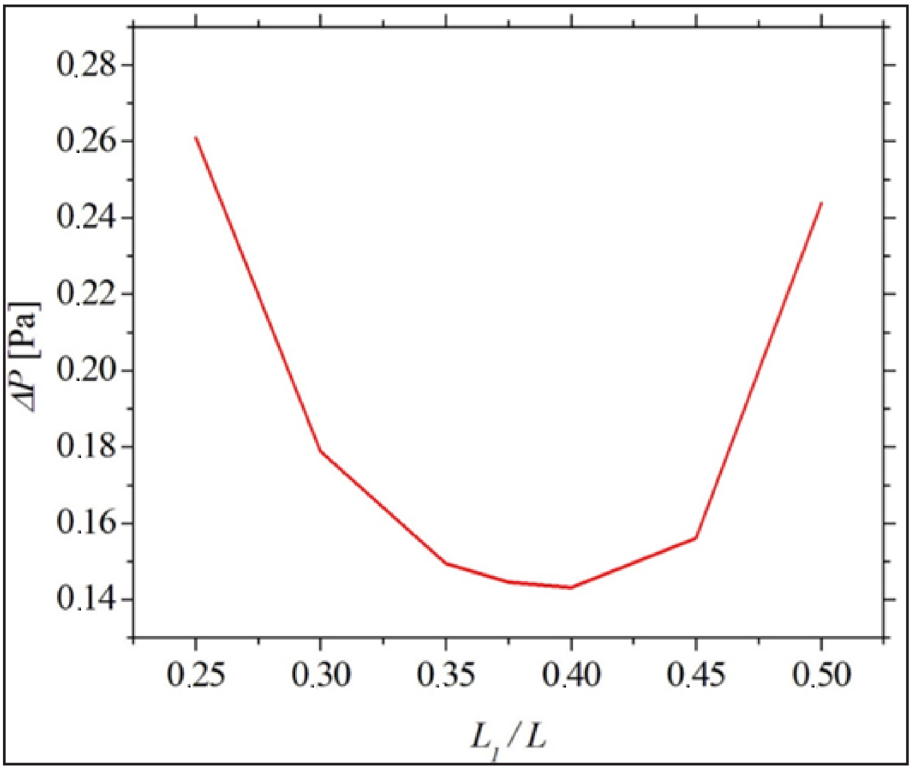
Figure 8 – Pressure field for the inferior extreme of $L_1/L$ = 0.5




Source: Authors (2024)

Regarding the quantitative comparison, Fig. 9 shows a parabolic behavior of *ΔP* as a function of the ratio $L_1/L$ with the minimal magnitude being obtained for an intermediate ratio of $L_1/L$. Therefore, it is possible to notice that the magnitudes of the pressure drop are higher when the second fin is closer the first or third fin, and the best fluid dynamic performance is obtained for the intermediate configuration $(L_1/L)_{o,F}$ = 0.40, which conducted to a minimum pressure drop of $(\Delta P)_{min,F}$ = 0.143. In general, results of pressure drop indicated that the best fluid dynamic performance is obtained for intermediate ratios of $L_1/L$ for the thermal conditions and constant ratios of $L_2/L$ and $L_3/L$ studied here, i.e., when the distance between fins 1-2 and 2-3 is almost symmetrical.

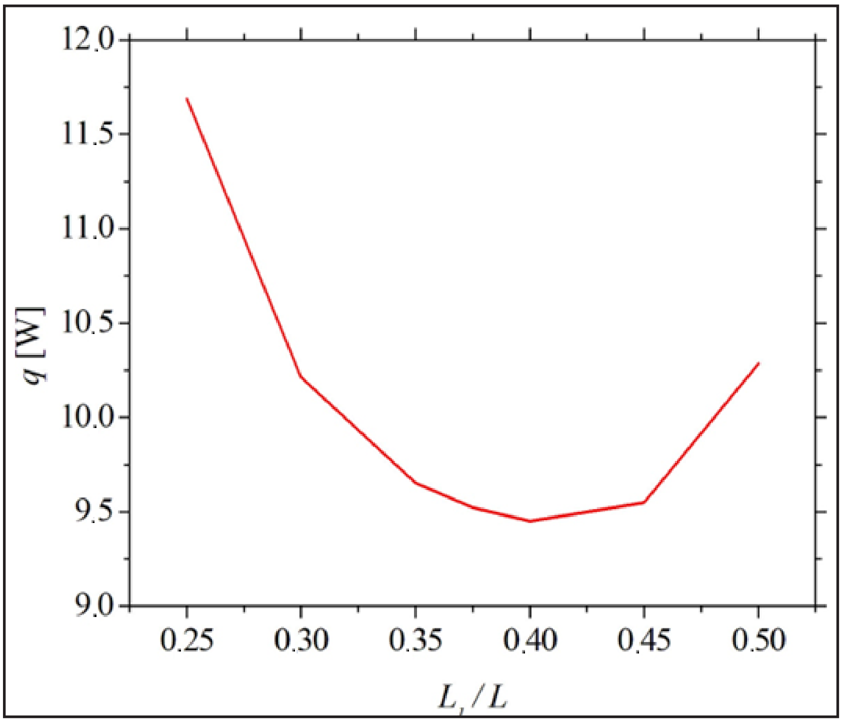
Figure 9 – Effect of the ratio $L_1/L$ over the pressure drop for $Re_H$ = 100 and $Pr$ = 0.71




Source: Authors (2024)

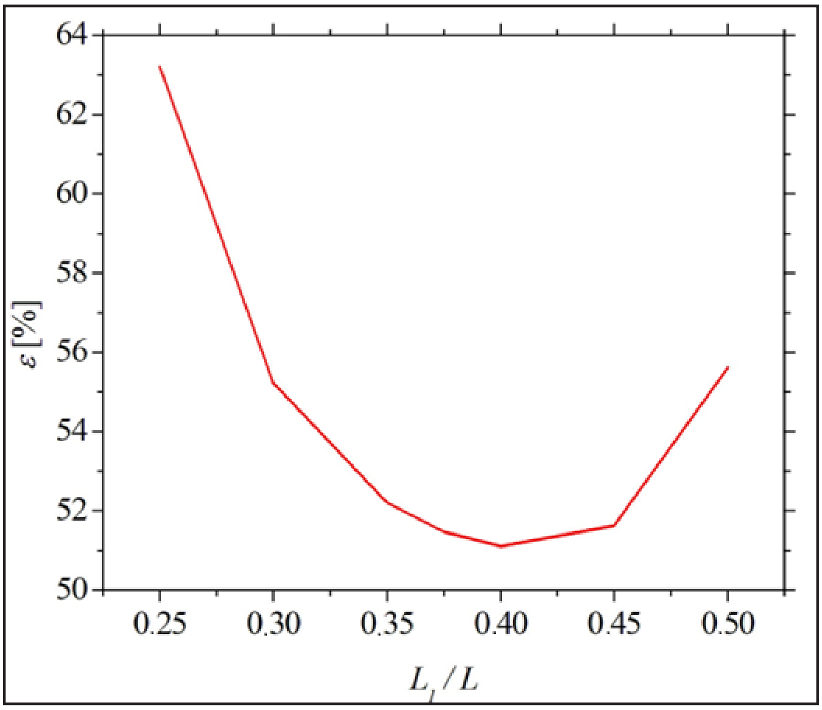
Figures 10 and 11 illustrate the effect of the ratio $L_1/L$ over the heat transfer rate ($q$) and the efficiency of the arrangement of fins in the channel ($\varepsilon$), respectively. Similarly to the behavior observed for the pressure drop, the curves of $q$ and $\varepsilon$ as a function of the ratio $L_1/L$ also presented a parabolic trend. However, for the thermal purpose, the aim is to maximize the performance indicators $q$ and $\varepsilon$. It is also observed that the behavior seems in Figs. 10 and 11 is quite similar, which is expected once there is a direct correlation between the heat transfer rate and the efficiency of the arrangement. In general, results indicated that the best thermal performance is obtained for the inferior and superior extremes of the ratio $L_1/L$, with the optimal thermal configuration reached for $(L_1/L)_{o,T}$ = 0.25. Therefore, from the thermal purpose viewpoint, the best behavior is obtained for the lowest possible distance between the upstream fins 1 and 2 for the present thermal conditions and constant degrees of freedom ($L_2/L$ and $L_3/L$). This configuration allows the augmentation of the momentum between the upstream fins and also a straight thickness of the thermal boundary layer in the isothermal walls before the fins and impinging over the arrangement.

Figure 10 – Effect of the ratio $L_1/L$ over the heat transfer rate ($q$) for $Re_H$ = 100 and $Pr$ = 0.71




Source: Authors (2024)

Figure 11 – Effect of the ratio $L_1/L$ over the efficiency of the arrangement ($\varepsilon$) for $Re_H$ = 100 and $Pr$ = 0.71




Source: Authors (2024)

In general, results demonstrated the importance of the geometrical investigation in finned channels. This observation can be attested in the differences of fluid dynamic and thermal performance between the best and worst configurations, where differences of nearly 44 % and 23 % are achieved.

## 4 CONCLUSIONS

The present numerical study investigated the geometry of an arrangement of isothermal rectangular fins, mounted in alternate form in two-dimensional channels, and cooled by forced convection airflows. The main purpose here is to obtain recommendations about the influence of the position of the second fin on the fluid dynamics and thermal performance of the arrangement for a forced convective flow with $Re_H$ = 100, $Pr$ = 0.71, $L_2/L$ = 0.625, and $L_3/L$ = 0.875.

Results indicated that the most appropriate configuration to reduce the pressure drop (ΔP) was obtained for $L_1/L$ = 0.4, i.e., an almost symmetrical configuration. The best configuration performed nearly 44 % better than the worst configuration. For the thermal purpose, the placement of the second fin near the first or third fins led to improvements on the thermal performance in comparison with more symmetrical configurations of the arrangement. The variation of just one degree of freedom ($L_1/L$) allowed an improvement on the thermal performance up to 23 %. Therefore, it is expected that the complete investigation (varying also $L_2/L$ and $L_3/L$) can improve even more the system performance. Results also indicated that the composition between the distribution of the thermal boundary layer and momentum among the fins are the main responsible for the design construction of the arrangement. Therefore, if the pressure drop is not a restriction for the problem, asymmetrical configurations with minimal magnitudes of $L_1/L$ are recommended.

For future works, it is recommended the investigation of the degrees of freedom kept constant in the present work ($L_2/L$ and $L_3/L$) and other parametric conditions as Reynolds and Prandtl numbers.

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Contribution: Methodology, Project Management, Write – Revision and Edition, Conceptualization, Supervision

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