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Research Article

Oscillating piezofan effects on natural and forced convection flow in a vertical channel with protruding heat sources

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ABSTRACT

The current work investigates the flow characteristics and convective heat transfer performance of a vertically-oriented piezofan in a channel with wall-mounted protruding heat sources. A 2D numerical model is developed in COMSOL Multiphysics to simulate the temperature variations on the heated protrudes under forced and natural convection scenarios, with and without oscillation of the piezofan. A vertical channel with dimensions of L_{ch} = 300 mm × S = 35 mm is considered with four protruding heat sources (H_{hs} = 25 mm × b = 7 mm), which are positioned on the same side of the channel, with a frontal surface heat flux of q_w = 600 W/m^2. The results reveal that active and passive vortices generated by piezofan and protruding objects in the flow domain provide higher momentum and heat exchange. It was found that implementing piezofan inside the channel significantly enhances cooling performance and reduces the surface temperatures of the protrudes. Compared to the pure natural convection within the channel, the average convective heat transfer coefficient on the protrudes increased more than three times using piezofan. Besides, in the case of forced convection, the maximum increments in mean convective heat transfer are obtained as 74% and 59% at Reynolds numbers 1000 and 2000, respectively.

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INTRODUCTION

The thermal issues associated with electronics cooling applications have motivated further research in the scope of convection channels due to factors such as coupling of discrete heating, conduction in the substrate, and components with the fluid convection. Some of the practical importance of such problems was highlighted in several reviews.

Due to the rapid improvements in electronic technology and the miniaturization trends of the electronic industry, the convective air cooling of electronic components mounted on a circuit board has been an attractive

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motivating topic of many papers, and investigations have dramatically increased during the last three decades. Relying on experimental and numerical studies in the literature, discrete cards or parallel plates can be constructed as 2D vertical air channels for natural and forced convection configurations of various electrical components in different types of equipment. As these components cause passive Vortex Generation (pVG) in the flow field, they improve the convective heat transfer rate. Most of the previous analyses focused on the design of effectively cooling electronic packages by controlling geometries and structure layout [1-12]. For example, Young and Vafai (1998a & 1998b) [1,2] published numerical simulation results for forced convection inside a horizontal channel with an array of heated blocks attached to the bottom wall. They demonstrated that choosing proper obstacle arrangements and configurations can considerably influence flow domain and heat transfer characteristics. Such a passive heat transfer improvement can open the way for various electronic applications. They also conducted experimental investigations and compared the measurements against the 2D laminar flow numerical model [3]. They evaluated the obstacle temperatures and Nusselt number variations as a function of flow rate and obstacle configurations to propose better designs for higher heat transfer enhancements. Similarly, Zeng and Vafai [4] developed a numerical model of an array of obstacles in a horizontal channel to obtain two-level Nusselt number correlations to quantify the influence of different factors on cooling performance. Their predictions agreed well with the numerical simulation data from the literature. Sarper et al. [5] studied the buoyancy-induced flow and heat transfer characteristics in a vertical parallel plate channel with four discrete protruding heat source arrays experimentally and numerically to model an electronic package. They focused on the effects of the geometrical parameters on recirculating flow and cooling performance. They revealed that the blockage ratio significantly impacts the system's overall cooling performance. In the numerical study of Haghighi et al. [6], they considered a single or an array of porous metallic designs of different configurations located inside the surface of a partially heated wall of a horizontal channel. They compared the flow and thermal considerations of two working fluids, air and water, and it was concluded that metal foams might be utilized as heat sinks, paving the path for future applications such as solar thermal collectors and electronic cooling.

In the literature, forced convection in protruded channels has also become an issue in many passive vortex generator investigations for electronic air-cooling applications. Various geometries are placed in the streamflow, focusing the vortex and the secondary flow toward the heated surfaces to improve the convective heat transfer. Wang and Jaluria [7] introduced numerical research to study the cooling performance by analyzing unsteady mixed convection through a horizontal channel with two insulated protruding blocks placed on the bottom wall. The results demonstrated the ability to control the primary flow frequency and amplitude by adjusting the geometry of the promoter. Chomdee and Kiatsirioat [8] presented an experimental model to analyze the heat transfer improvements by delta winglet vortex generators placed in front of a staggering array of rectangular electronic modules on a circuit board. They demonstrated in the results that the vortex generators could significantly improve the heat transfer coefficient and reduce the module temperatures. Oztop et al. [9] built a 2D laminar flow model to analyze the convective heat transfer rate and fluid flow around isothermally heated blocks on a horizontal channel wall. They used a triangular cross-sectional bar located at different locations to control and guide the streamflow at three Reynolds numbers from 400 to 1300. They obtained that inserting a triangular cross-sectional bar can enhance the heat transfer rate for all Reynolds numbers compared with the plane channel without the triangular obstacle. Beig et al. [10] introduced an optimization study using a Genetic algorithm (GA) combined with a Gaussian Process (GP) to conduct the highest uniform heat transfer by obtaining the optimal position of a triangular cross-sectional bar vortex generator to cool a blocked channel, i.e., electronic chips. They revealed that the flow domain structures primarily depend on the location of the vortex generator, regardless of the flow Reynolds numbers. Al-Asadi et al. [11] developed a 3D laminar flow numerical model to study the conjugate heat transfer within a micro-channel to explore the effects of quarter-circle and half-circle cylindrical cross-sections vortex generators arranged along the bottom wall of the horizontal channel. They analyzed the flow domain and the cooling performance with Reynolds numbers ranging from 100 to 2300 and uniform heat flux cases. They demonstrated that the vortex generation with these structures significantly improves the heat transfer rate, promising substantial improvements toward miniaturization technology with lightweight and small-volume mobile electronic devices. Zheng et al. [12] explored the effects of trapezoidal cross-section geometric parameters as longitudinal vortex generators on local and overall thermal-hydraulic characteristics at the Reynolds number range of 347–868 inside a longitudinal mini-channel. Numerical results proved that the proposed geometries have better comprehensive thermal performance than rectangular cross-sections. Also, they recommended the optimal geometrical parameters of a trapezoidal channel by achieving the best performance; its height matches half of the channel height.

The works that come after referring to the problems of pVG are the limitation in the number of generated vortices and low cooling performance with high heat generation in modern electronic components. Therefore, many further attempts have been made to improve the cooling performance by introducing different types of active vortex generation (aVG) applications. Active Vortex Generations (aVG) strategies include moving geometries in the flow channels to improve convective heat transfer [13-17]. Yang [13,14] used the ALE method and a Galerkin finite element method in the simulation model to investigate the heat transfer enhancement in the presence of an oscillating bar directed to heated obstacles on the channel wall. The heat transfer rate significantly increases with the Reynolds number and the bar oscillating amplitude. Fu and Tong [15] conducted a numerical simulation using ALE and Galerkin finite element methods to inspect the flow profiles and heat transfer features of a heated diagonally oscillating cylinder inside the cross-flow. They compared the vortex-shedding structures and heat transfer characteristics around the heated cylinder by varying the flow Reynolds number, piezofan's oscillating amplitude, and oscillating velocity. It is concluded that the heat transfer rate improved with increasing oscillating velocity of the cylinder and the Reynolds number. Kurşun and Sivrioğlu [16] investigated the development of the flow routing plates on the structures of laminar mixed convection heat transfer configurations in a horizontal channel, which has protruding heat sources mounted on the top and bottom walls experimentally and numerically using 3D laminar flow analysis. They demonstrated that using flow routing plates has only enhanced heat transfer for the first bottom row of heaters and the first and last top surface row of heaters, and the natural convection improved with the increasing Grashof number and decreasing Reynolds number. Ismael [17] used numerical 2D benchmarked geometry with two baffles bound to the compliant segment to investigate the forced convective heat transfer and the performance of a channel partially heated by constant heat flux under the effects of two alternative upstream-downstream vortex generator baffles. It is concluded that the Nusselt number is improved by 94%, corresponding with the non-baffled channel at Re = 250.

Based on the outcomes of literature research on the active vortices generated by piezofan and their properties consistent with modern technology. Its properties make it a good choice for easy air cooling and as an economical alternative in terms of energy consumption, easy construction, and free noise and vibration. Therefore, the further addition of state-of-the-art numerical analysis in this field will be of significant importance in the technology industries. Numerous researchers have studied the cooling performance of piezofans. Kim et al. [18] tested the 2D model of Choi et al. [19], and it was found that the static pressure difference across the fan tip plays an essential role in generating and developing double counter-rotating vortices. A 2D model of an oscillating piezofan in a vertical channel is proposed by Florio and Harnoy [20], and there is a 52% increase in the local heat transfer coefficient compared to pure natural convection. Lin [21] used flow visualization techniques and 3D numerical simulations to investigate the cooling performance of a vibrating piezofan placed in the wake area of a cylinder in the presence of forced cross-flow. They found that the piezofan increased the cylinder's overall and local heat transfer ratios by 132% and 214%, respectively. Jeng and Liu [22] conducted experiments to discuss the effects of piezofan's transverse and axial oscillations on the channel airflow to improve the heat transfer of various heat sink structures. They found that the square pin-fin heat sink improves heat transfer, and the transverse oscillation increases heat transfer slightly higher than axial oscillation. Li et al. (2017 and 2018) [23,24] investigated the cooling effects of vertically oriented piezofan on a heated flat surface in the presence of forced cross-flow inside a channel both experimentally and numerically. The main outputs are summarized as follows: *(i)* Combining piezofan and channel flows improves heat transfer, especially at small gaps between the piezofan tip and heated surface; *(ii)* The heat transfer coefficient increased by about 56.2% compared to the cross-flow channel; *(iii)* With the cross-flow, the oscillating amplitude of piezofan was reduced by more than 56%; *(iv)* The piezofan cooling performance improved only in the presence of low-velocity cross-flow conditions and *(v)* Due to high momentum cross-flow, the piezofan-generated vortices were forced downwards and/or swept away. Oh et al. (2018, 2019) [25,26] presented a 3D simulation of a vibrating cantilever piezofan between two walls to highlight the interaction flow between the created vortices. They discovered that the interaction of the piezofan tip and side vortices with each other formed some complex secondary flow configurations. By examining different gap heights, the size and strength of the vortices, as well as their interaction, are all affected by the end wall placement.

Park et al. (2019) [27] conducted 2D simulations with eight inlet velocities ranging from 0 to 7 m/s and 3D simulations with five velocities of 0, 1, 3, 5, and 7 m/s to investigate the effects of the free stream on the flow around a piezofan. ANSYS-FLUENT software is used in the simulations, and the plate motion is defined in the software with a user-defined function (UDF) subroutine. The results were obtained from 2D simulations in accordance with the 3D predictions and the experimental data. It is concluded that, considering the computational cost of the 3D simulations, the 2D model results would be sufficient to determine the effects of piezofan motion in conditions where the side vortex behavior is negligible. They showed that the free-stream has a greater influence on the counter-rotating vortices formed at the piezofan tip than on the vortices generated on the piezofan side. The axial velocity around the channel end wall increased at low free-stream velocities. This suggests that a low-speed free-stream may be the only way to improve the piezoefan's flow performance. The flapping dynamics of two types of piezofans in channel flow and their impacts on heat transfer enhancement performance were investigated experimentally by Chen et al. (2020) [28]. They compared three turbulent Reynolds values of 0.85×10^4, 1.37×10^4, and 19.1×10^4 to see how thermal performance was affected by the flow conditions. According to their findings, the flapping piezofan improved heat transfer performance over a wide range of Reynolds numbers. The flapping amplitude and frequency increase as the Reynolds number increases, resulting in improved heat transfer performance. The heat transfer is improved by

30% at low Reynolds numbers, i.e., Re = 1.37×10^4, while the maximum local heat transfer was increased by 46% at Re = 19.1×10^4 under a frequency of 43 Hz. Tiwari and Yeom (2021) [29] used a piezofan to examine how to increase convection heat transfer in an air channel flow experimentally and numerically. ANSYS-FLUENT software is used to simulate the flapping motion of the piezofan. A sinusoidal motion is defined in UDF with a time step size of 2.76E-5 seconds. It is noted that the numerical results are in agreement with the experimental counterparts, and the difference is less than 10%. The piezofan position at the front end of the heated surface achieves the highest heat transfer enhancement. They showed that the development, propagation, and impinging dynamics of vortical structures significantly affected the convective heat transfer rate of the heated surface in the channel. Ko et al. (2021) [30] studied the effects of piezofan on cooling a flat plate experimentally and numerically. ANSYS-FLUENT software is used to simulate the 3D piezofan model. Predicted plate temperatures are compared with experimental measurements. It is noted that the difference is less than 1°C in the case of piezofan being activated. The installation was performed to investigate the relationship between the generated vortices and the heated plate's temperature distribution. They verified that without piezofan, the temperature is highest at the center of the plate and lowers toward the plate edge. They also confirmed that the local temperature at the center of the plate is diminished by 28°C, and the improvements in the local convective heat transfer coefficients showed that the piezofan is very practical for cooling the plate.

In the previous works of the authors [31,32], the effects of oscillating piezofan in horizontal channels on the flow field at various flow-stream velocities (Reynolds numbers 1040 to 2080) were addressed. It was obtained that the high flow and cooling performance of this fan was described by several characteristics, such as the vertical and horizontal positions and their effects on the flow and temperature distributions. The results demonstrated that at low flow-stream velocity and position of piezofan near the center of the heated surface, the convection heat transfer coefficient increased by 231% compared to pure laminar flow condition. Hasan et al. (2023) [32] discuss the influences of vortices generated by vertically oriented piezofan operating at different oscillation ratios (0.15 to 0.3) on the flow domain and cooling performance in a vertical channel heated by the side walls. It was proved that the heat transfer rate over the entire channel increased by 28% compared to non-piezofan natural laminar flow conditions at a 0.3 oscillation ratio. Boz et al. [33] investigated the influence of the piezofan amplitudes ($2A$ = 6 – 12 mm) and frequencies (f = 10 – 20 Hz) on the heat transfer inside the vertical plane channel and compared it with the pure natural convection case. Results revealed that the heat transfer enhancement can reach up to 169% by using the piezofan.

The main goal of the current paper is to investigate the general trends associated with protruding heat sources inside natural and forced convection vertical channels. By combining passive and active vortex generation effects during the operation of vertically oriented piezofans, we will determine the effects on flow domain structure and convective heat transfer.

Problem Definition and Assumptions

As is fundamentally known, increasing the surface area exposed to the convective heat transfer is one of the main significant parameters for passively improving the heat dissipation rate. Thus, electronic packages composed of various configurations of series of flush-mounted or protruding discrete boards with or without a substrate to form thermally generated components in channels or parallel plates have become a subject in the advanced electronics industry. Many researchers have achieved numerical simulations to analyze the flow around piezofans, in particular, thanks to the rapid development of computational facilities and approaches. Even though 3D models enable better comprehensive solutions, they require additional computational capacity and cause re-mesh failure in deformed and stretched elements with longer simulation times. Most of the previous research has focused on cooling performance and the flow around the piezofans for 2D numerical simulations in order to reduce computation time while depending on assumptions that could decrease the solution's error rate. Accordingly, we relied on 2D model assumptions in this work to focus on vortex generation in the air domain and cooling performance enhancement based on the suggested realistic structures.

The physical geometry considered in this work is illustrated in Figure 1. Laminar 2D natural convection air-cooling in a vertical channel (L_{ch}=300mm and S=35mm) with four protruding heat sources (H_{hs}=25mm and b=7mm) mounted on one side of the channel wall is studied. Interface heat flux (q_w=600 W/m^2) is defined on the vertical surfaces of the protrudes to simulate an actual heat dissipation from the discrete electronic packages. Other walls, i.e., the left and right channel walls and horizontal surfaces of the protrudes, are considered to be adiabatic. The current investigations include forced flow with Re=1000 and 2000 and natural convection. The flow Reynolds number is defined as Re = $\rho V_{in} S/\mu$, where V_{in} is the average air velocity at the channel inlet, and S is the channel width. The numerical results of heat transfer and flow behavior with an oscillating piezofan are compared to a non-piezofan state under the same design and flow conditions.

Numerical model and boundary conditions

The equation of piezofan's oscillating motion is defined in terms of the frequency and the geometric parameters as [31, 32].

$$x_{pz}(y,t) = \left[\psi A_c \left\{ \begin{array}{l} [(sin(\beta L_{pz}) - sinh(\beta L_{pz}))(sin(\beta x) - sinh(\beta x)) \\ + (cos(\beta L_{pz}) - cosh(\beta L_{pz}))(cos(\beta x) - cosh(\beta x))) \end{array} \right\}\right] sin(2\pi f_r t) \quad (1)$$

Dimensionless driving coefficients are defined as ψ = 1.862 and β = 21.233. The piezofan with L_{pz} = 40 mm length is adjusted to operate at a constant oscillating frequency and amplitude of f_r = 20 Hz and OA = 10.5 mm, respectively. Except for the front surface of the heated sources, all sidewalls are considered no-slip and adiabatic. Table 1 summarizes the boundary conditions that are defined in the current simulations.

Transient governing equations of airflow and heat transfer physics are simultaneously resolved with solid mechanics physics for the 2D domain. This task was completed by calculating the deformation in the fluid domain using a moving boundary option. The deformation mesh is formed by connecting the induced forces to the cantilever motion of the piezofan in solid mechanics with fluid flow using FSI (fluid-structure interaction analysis). In the time-dependent study, the moving boundary conditions were set up as an automatic re-meshing option utilizing the ALE (Arbitrary Lagrangian-Eulerian) computation method. COMSOL Multiphysics software is used to couple physics dynamically. The grid size with 15546 mesh elements was chosen for all cases based on the mesh-independency analysis in the previous studies [31,32]. Continuity, momentum, and energy equations are defined for incompressible fluids as [31, 32].

$$\text{Continuity: } \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{2}$$

$$\text{x- momentum: } \rho\frac{\partial u}{\partial t} + \rho u\frac{\partial u}{\partial x} + \rho v\frac{\partial u}{\partial y} = -\frac{\partial P}{\partial x} + \mu\frac{\partial^2 u}{\partial x^2} + \mu\frac{\partial^2 u}{\partial y^2} \tag{3}$$

$$\text{y- momentum: } \rho\frac{\partial v}{\partial t} + \rho u\frac{\partial v}{\partial x} + \rho v\frac{\partial v}{\partial y} = -\frac{\partial P}{\partial y} + \mu\frac{\partial^2 v}{\partial x^2} + \mu\frac{\partial^2 v}{\partial y^2} \tag{4}$$

$$\text{Energy: } \rho c_p\frac{\partial T}{\partial t} + \rho c_p u\frac{\partial T}{\partial x} + \rho c_p v\frac{\partial T}{\partial y} = k\frac{\partial^2 T}{\partial x^2} + k\frac{\partial^2 T}{\partial y^2} \tag{5}$$

Similar to previous research [32], fine triangular elements are used, and the mesh intensity of 15546 is selected in the simulations at adjusted oscillating amplitude OA = 10.5 mm. The time-stepping size of Δt = $(1/fr\times100)$ = $1/(20\times100)$ = 0.0005 s is used to provide a suitable convergence during the computations. In addition, different sets of verifications with experimental works from the literature are discussed in detail in the authors' previous works [31,32].

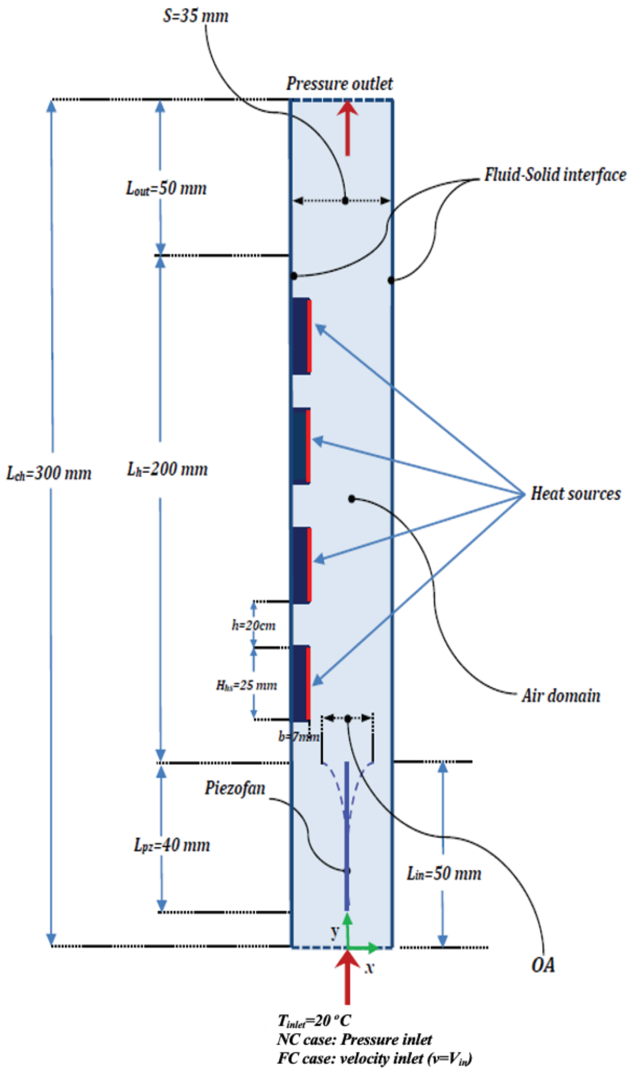


Figure 1. The 2D vertical channel model structure.

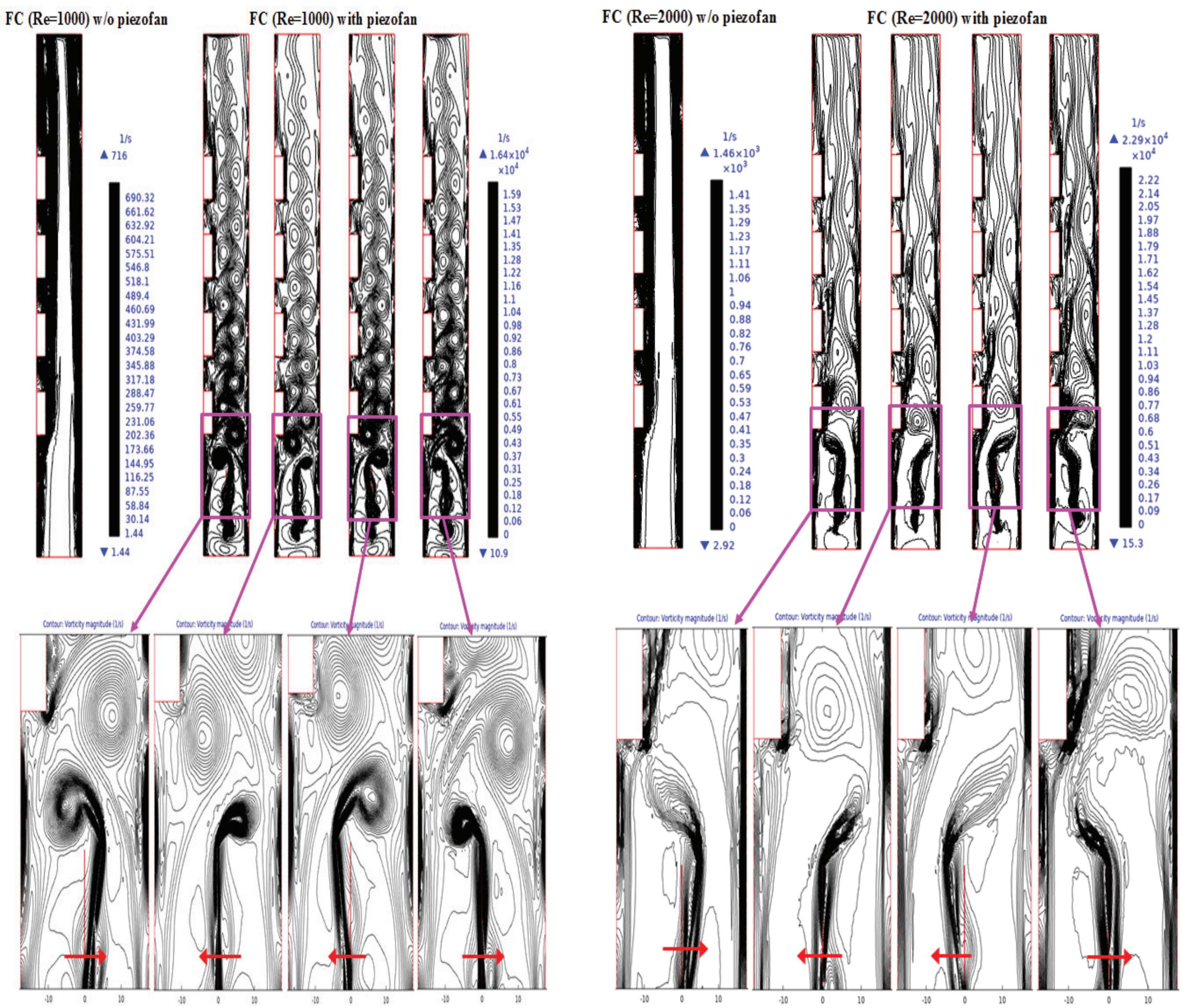
Table 1. Boundary conditions summary

Boundary	Assumption	Type	Fluid flow/ Heat transfer
Inlet section	$y = 0 \rightarrow P = P_{atm}$, T_{in} = 293.15K, $v=V_{in}$ (*at* Re = 1000, V_{in} = 0.439 m/s, *at* Re = 2000, V_{in} = 0.878 m/s)	*NC case: Pressure inlet* *FC case: velocity inlet*	Normal flow condition
Channel walls	$x = \pm S/2 \rightarrow u = v = 0$; $\partial T/\partial x = 0$	*No-slip flow with adiabatic wall*	Flow domain: gravitational acceleration in the negative y-direction; incompressible laminar flow with Boussinesq approximation
Protrudes	$x = -S/2+b \rightarrow u = v = 0$; $q=q_w$	*No-slip flow with constant heat flux on the wall*	Front-side heat sources
Outlet section	$y = L_{ch} \rightarrow u = 0$, $\partial v/\partial y = 0$	*Pressure outlet*	Normal flow with suppressing backflow condition

RESULTS AND DISCUSSION

In the numerical analyses, heat transfer and flow were investigated in cases where natural convection and Reynolds numbers were 1000 and 2000, with and without piezo fan. As previously stated, continual vortices are generated at the piezofan tip during the cantilever motion. These vortices are generated in counter-rotating movement, then separate and travel left and right toward the channel's outlet. They will join with the low-velocity vortices created at the corners and edges of the protruding blocks during their passage, generating longitudinal vortices with high velocity and momentum that will spread across all surfaces and protrusions, as shown in Figure 2 and b. This diffusion breaks the boundary layers and reduces their thickness, moving in this manner continually, lowering the temperatures of the hot surfaces and greatly improving the heat transfer rate by convection significantly, hence improving the thermal performance of electronic devices or components. The vortices break the thick boundary layer at low cross-flow velocities in the channel; consequently, the heat transfer rates increase at all surfaces and edges. At high velocities, the vortices drift upwards with the main flow and reduce the heat transfer rate.

In contrast, as shown in Figure 3, the double effects of the generated vortices by oscillating piezofan and the passive low-velocity vortices that are generated at the corners and edges of the protruding heat sources are combined with the secondary flows of the bouncy effects of natural convection to produce high momentum and flow energy vortices. These flow domains significantly impact the boundary layers during their free movements towards the channel exit, increasing the cold region close to the hot surfaces. This improves the heat transfer rate by free convection



a) Re=1000 b) Re=2000

Figure 2. Vortices generation on the flow domain in the forced convection channel during one- operating cycle of the piezofan.

and enhances cooling performance compared to a channel without this fan.

Figure 4a, b, and c comparatively depict the influence of a fluctuating piezofan on the streamlines inside the flow domain. The generated vortices in Figure 4a have a longitudinal shape. They are separated further on the left protruded surface and take the sufficient time required for the heat transfer. They then expand the cold surface domain through interaction with these hot surfaces and edges, which improves the cooling performance of the channel flow in the presence of the fan. In Figure 4b, the double counter-rotated generated vortices and their separation in the flow domain toward the channel exit at low-velocity stream flow channels are all clearly visible. However, as shown in Figure 4c, due to the quick flow separation at the piezofan tip, these vortices do not emerge noticeably in the high-velocity stream flow domain.

At low velocities, the vortices break the hydrodynamic boundary layer; consequently, the heat transfer rates increase at all surfaces and edges. At high cross-flow velocities, the vortices drift upwards with the main flow, and their effects on heat transfer decrease. In all considered cases, there is a fluctuation in the y-component of the velocity. Figure 5 compares the variations in v-velocities at different channel sections along the flow direction. Oscillating the piezofan significantly affects the flow domain in natural

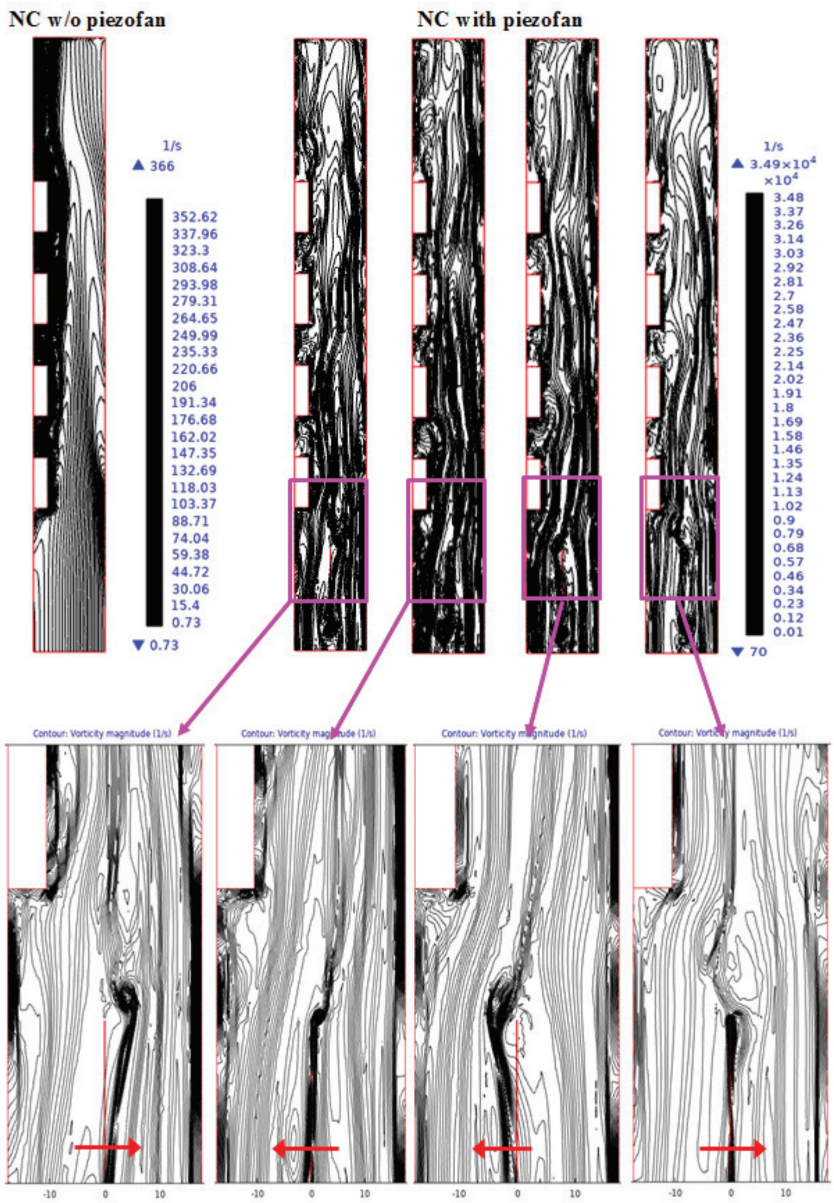


Figure 3. Vortices generation on the flow domain in the natural convection channel during one- operating cycle of the piezofan.

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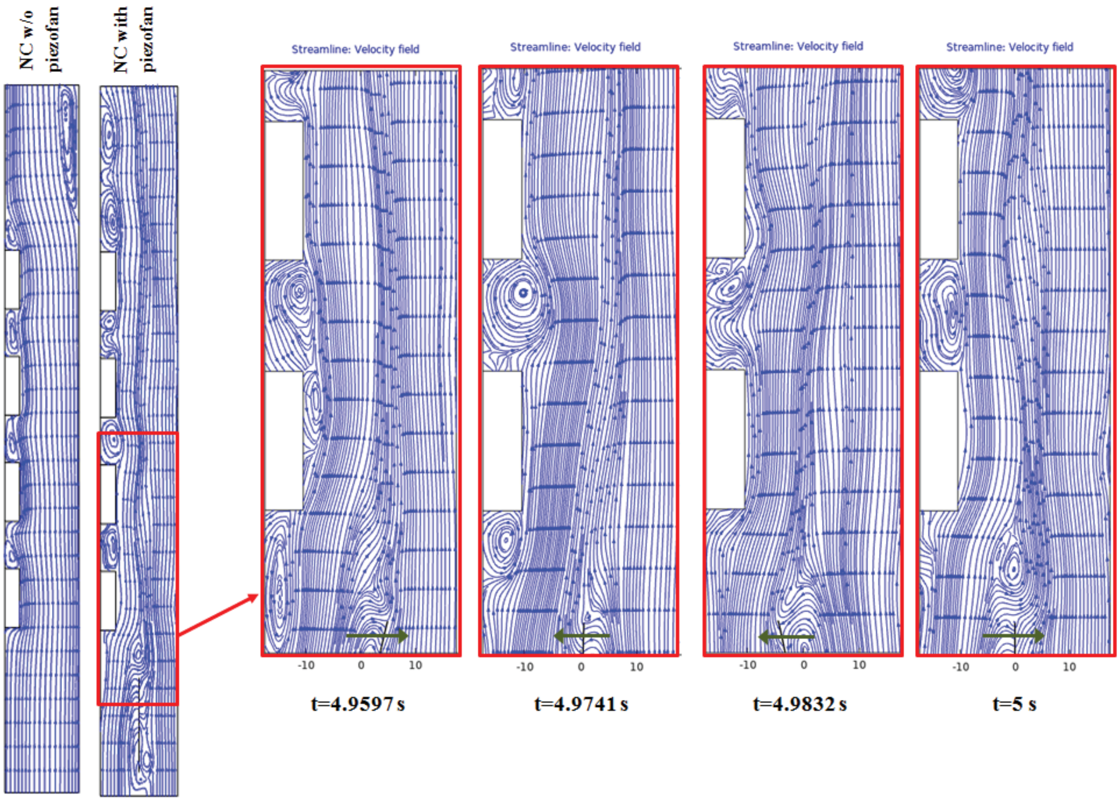
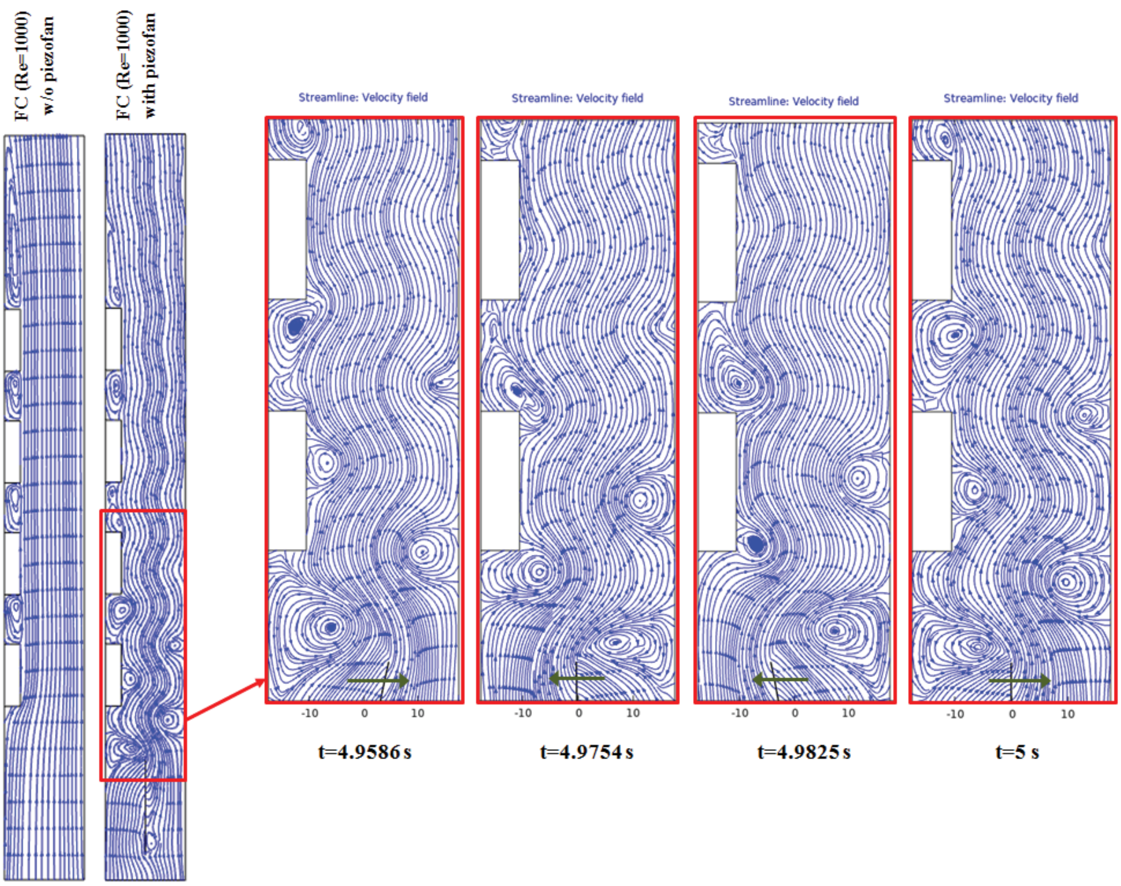
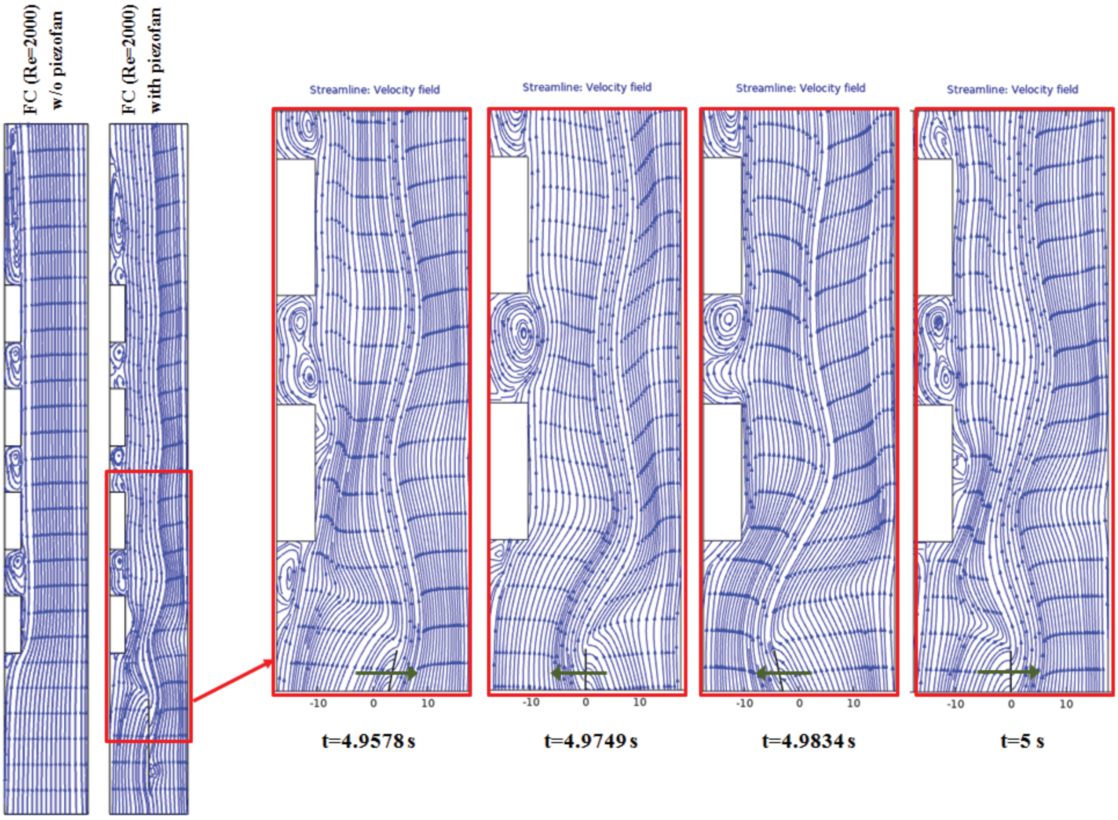


a) Natural convection



b) Forced convection flow (Re=1000)



c) Forced convection flow (Re=2000)

Figure 4. The effects of piezofan oscillation motion on the flow domain of channels during one-operating cycle compared to non-piezofan flow channels.

convection and low flow-stream velocities. The maximum velocity decreases in the flow domain towards the channel exit.

As mentioned above, natural convection with a piezofan helps reduce temperatures on the heated walls due to the high-velocity level over the entire domain. Mixing active and passive vortex generation with buoyancy effects reduces the boundary layer thickness and sweeps the heated surface with high momentum impacts toward the channel exit. The temperature contours given in Figure 6 illustrate this physics.

Figure 7. provides the fluctuations of protruding front surface temperatures (T_s) for five seconds of piezo fan operation. The steady-state condition was reached at about 3 s. for natural convection, and at 0.5 s for forced convection.

Figure 8 shows the time-dependent fluctuations of the average convective heat transfer coefficient at each protruding surface during the piezofan operation. Results show that, for each case, the steady-state oscillations are achieved after around 0.5 s at forced convection, and 3 s at natural convection. It is also important to note that the heat transfer rate significantly increases for the piezo fan-containing channels compared to non-piezofan channels. During piezofan oscillation, the heat transfer coefficients on the protruding surfaces increased by about 46% to 74% for Re = 1000 and about 2% to 59% for Re = 2000, respectively. Yang [13] studied the influence of an oscillating vortex generator inside a horizontal channel at low Reynolds numbers, i.e., 250, 500, and 750. Since the flow Reynolds numbers are relatively lower, the enhancements in heat transfer are achieved by as much as 97%, depending on the oscillating amplitude and the speed of the horizontal vortex generator. For the natural convection-induced case, the increments on the protruding surfaces are around 182% and 435%. One can conclude that due to the combined effects of generated vortices in the whole flow domain, piezofan has significant cooling performance effects in natural convection and low flow velocity forced convection channels.

After reaching the steady state, between 3 s and 5 s, the average temperatures of protruding surfaces are given in Table 2. It was shown that, for force convection channel at low flow stream velocity (Re=1000), the average surface temperature reduces by 16.2K, 19.6K, 16.6K, and 18.6K on the front surface 1st, 2nd, 3rd, and 4th protruding heat sources respectively, when compared to the flow in the same channel without piezofan. When the flow stream velocity increased (Re=2000), the temperature

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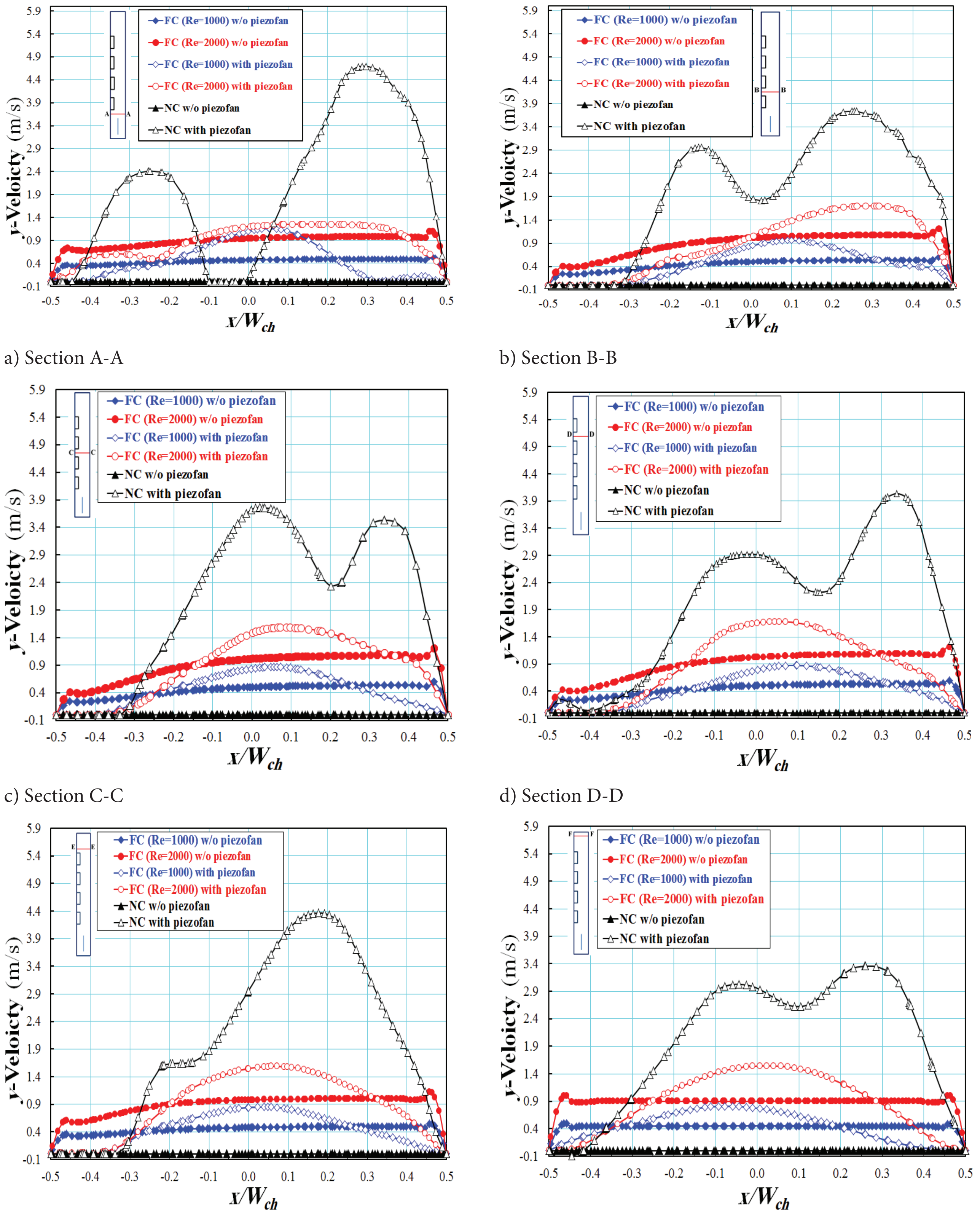


a) Section A-A

b) Section B-B

c) Section C-C

d) Section D-D

e) Section E-E

f) Section F-F

Figure 5. Variations in y-velocity for different flow cases at different channel sections at t = 5 s.

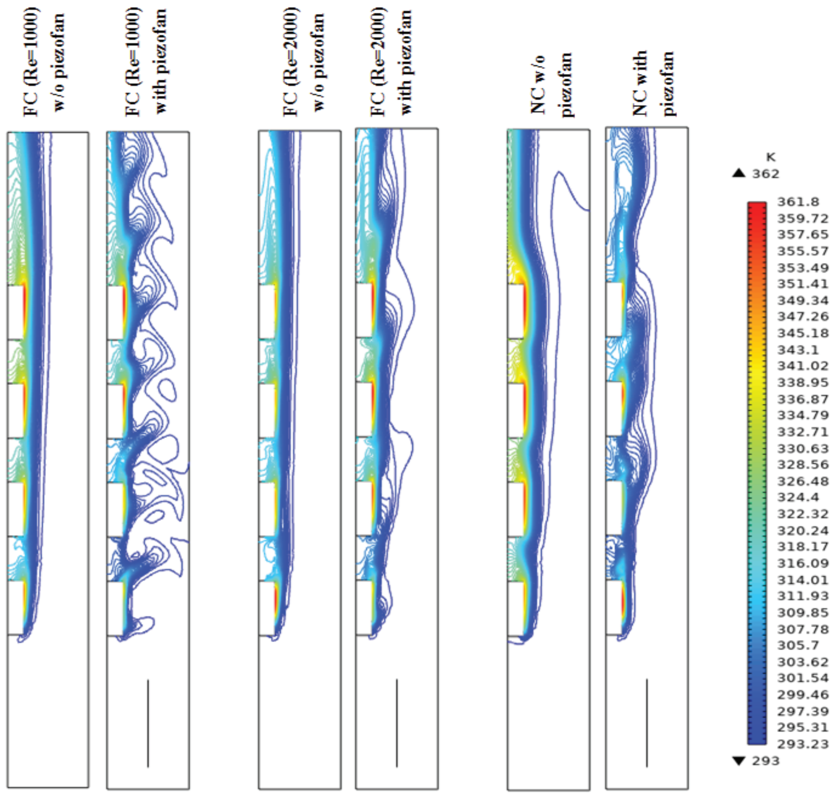
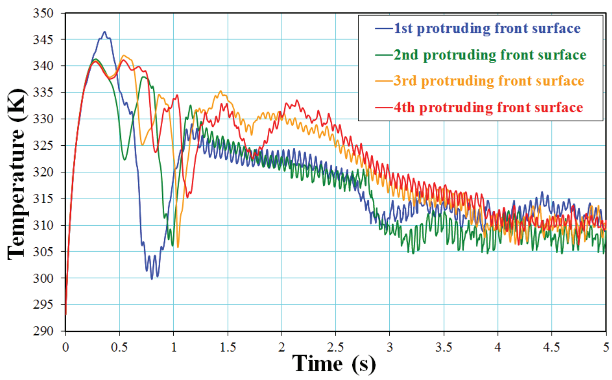
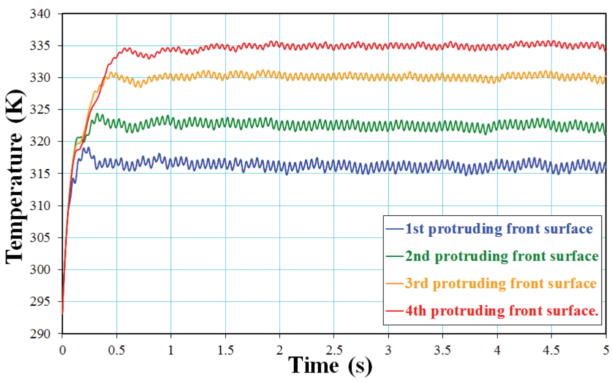


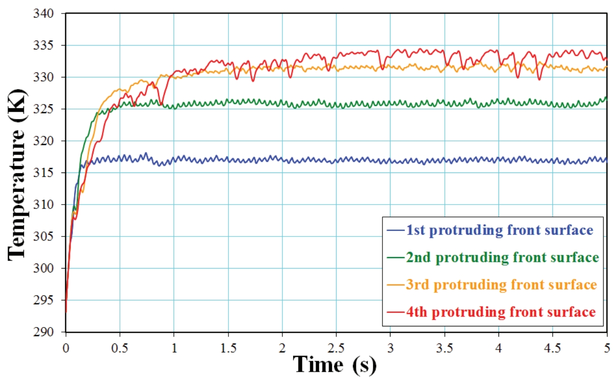
Figure 6. Comparison of temperature contours of different flow cases at t = 5 s.



a) Natural convection channel

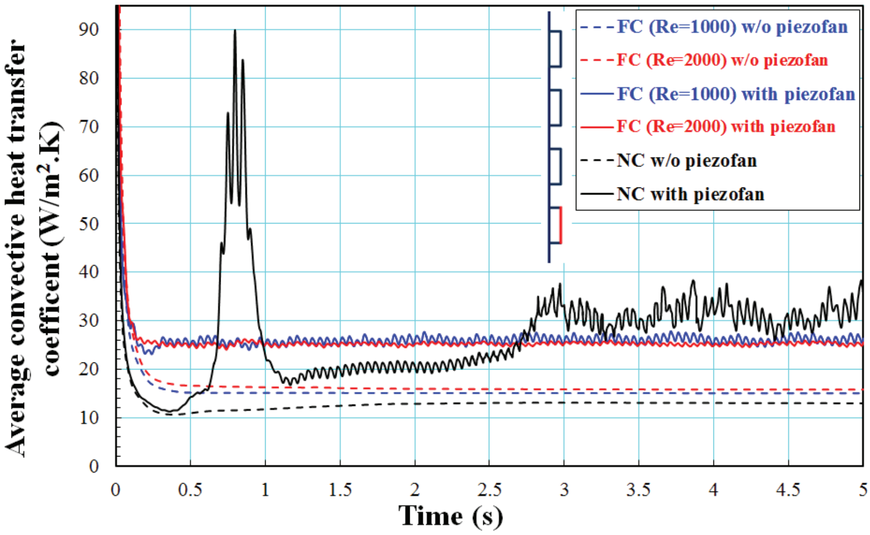


b) Forced convection (Re=1000)

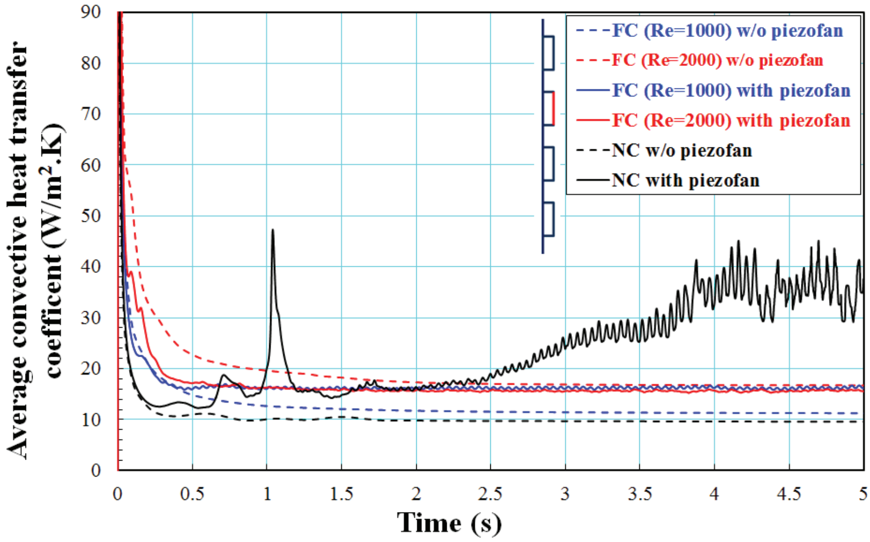


c) Forced convection (Re=2000)

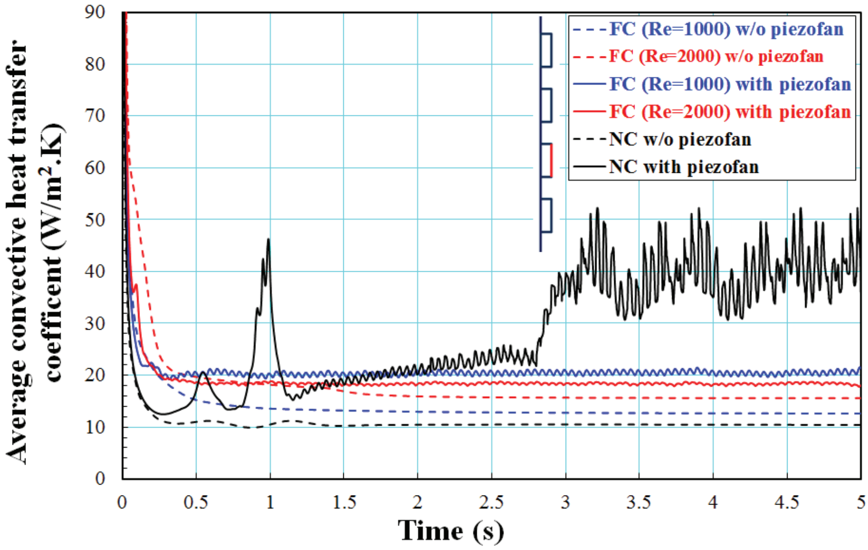
Figure 7. Timewise variations in protruding front surface temperatures for the considered cases during 5 s of piezofan operation.



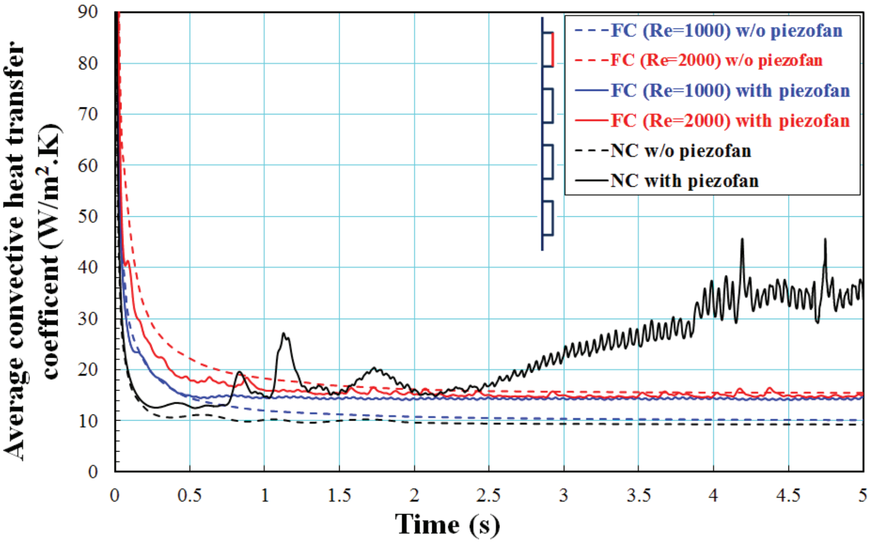
a) 1st protruding front surface



b) 2nd protruding front surface



c) 3rd protruding front surface



d) 4th protruding front surface

Figure 8. Time-dependent variation of average heat transfer coefficient at each protruding surface.

Table 2. Average temperature and heat transfer coefficient at each protruding surface after steady-state oscillation

	Re_m	Surface temperature (K)		Heat transfer coefficient (W/m²×K)	
		w/o piezofan	with piezofan	*w/o* piezofan	with piezofan
		1st protruding surface			
Case: NC	-----	338.20	309.13	13.30	37.50
Case: FC	1000	331.15	314.92	15.80	27.50
	2000	328.62	315.45	16.90	26.90
		2nd protruding surface			
Case: NC	-----	350.10	303.78	10.53	56.41
Case: FC	1000	339.51	319.92	12.94	22.40
	2000	330.37	325.13	16.12	18.76
		3rd protruding surface			
Case: NC	-----	355.05	308.56	9.70	38.93
Case: FC	1000	345.48	328.89	11.46	16.78
	2000	328.18	326.37	17.12	18.10
		4th protruding surface			
Case: NC	-----	357.10	307.83	9.40	40.80
Case: FC	1000	351.50	332.91	10.30	15.10
	2000	331.11	330.30	15.80	16.10

reduction decreased to 13.2K, 5.2K, 1.8K, and 0.8K on the 1st, 2nd, 3rd, and 4th protruding heat sources, respectively. While it is reduced by 29K, 46.3K, 46.5K, and 49.3K on the 1st, 2nd, 3rd, and 4th protruding heat sources, respectively, compared to a pure natural convection case without piezofan.

CONCLUSION

In this paper, the cooling performance of a vertically oriented single oscillating piezofan is numerically investigated. The computations were carried out using corresponding forced and natural convection channels assembled with wall-mounted protruding structures heated sources. The physics analysis is described by a combination of passive and active vortex generation and natural convection or forced convection effects. It is observed that when the passive and active impacts of vortices are combined, the flow momentum towards the channel exit increases, breaking the boundary layers in all swept surfaces and enhancing the convection heat transfer rate significantly. The results were calculated as an average in the steady-state range after 3 seconds of piezofan operation for all preceding cases. The following are the major flow and thermal considerations that this investigation has revealed:

- In forced convection channels, the presence of an oscillating piezofan increased air velocity by about 2.5% in different flow domain sections compared to non-piezofan channels. In contrast, the increment in the natural convection channel was about 21%.
- The calculated average convective heat transfer coefficients for flow with piezofan are higher than without piezofan for all considered cases. Especially in the natural convection channel shows the highest enhancements of about 182%, 435%, 300%, and 334% for the 1st, 2nd, 3rd, and 4th front surfaces of the protruding heat source, respectively.
- The vortices generated by the active oscillations of piezofan mixed with the bouncy effects through the natural convection channel have a high performance by decreasing the temperatures of protruding heated surfaces by around 29 to 49.3°C.
- As the flow stream velocity is increased in the forced convection channel, the presence of an oscillating piezofan has relatively minor effects on flow and heat transfer improvements. For example, reductions in the temperatures of the protruding heated surfaces do not exceed 0.8 to 13.2°C as the Reynolds number is 2000.
- Finally, the 2D laminar flow computations, which show a high rate of vortex generation and secondary flows mixing in the air flow domain, as well as a good improvement in cooling performance, can be used in a wide range of modern electronic industry applications, such as miniaturized electronic cooling systems or/and electronic components.

NOMENCLATURE

A_c	The cross-sectional area of the piezofan (m^2)
b	Protruding heated source thickness (m)
c_p	Specific heat at constant pressure (J/kgK)
E_{pz}	Young's modulus (Pa)
f_r	First resonant frequency (Hz)
h	Convective heat transfer coefficient (W/m^2K)
I	The second moment of area of piezofan (m^4)
k	Thermal conductivity (W/mK)
L_{ch}	Length of the channel (m)
H_{hs}	Protruding heated source height (m)
L_{pz}	Length of the piezofan (m)
$\dot{m}$	The mass flow rate of air through the channel (kg/s)
m_{pz}	Mass of the piezofan (kg)
Nu	Nusselt number
P	Pressure (Pa)
q	Heat transfer rate (W)
q''	Wall heat flux (W/m^2)
Re	Reynolds' number of mainstream channel flow
S	Vertical channel width (m)
T	Temperature (K)
t	Time (s)
t_{pz}	Piezofan thickness (m)
U	The average velocity of air inside the channel (m/s)
u	The velocity of air in the x-direction (m/s)
V_{in}	The average velocity of air at the channel inlet (m/s)
v_{pz}	Tip velocity of the piezofan (m/s)
v	The velocity of air in the y-direction (m/s)
x, y	Cartesian coordinates
$X_{pz}(y)$	Maximum piezofan displacement at arbitrary y location (m)
$x_{pz}(y,t)$	Instantly displacement of the piezo fan (m)

Subscripts

a	Air
ch	Channel
h	Heated wall
in	Channel inlet
out	Channel outlet
pz	Piezofan

Greek Symbols

β	Characteristic coefficient, Eq. (2)
σ	Poisson's ratio
μ	Viscosity (kg/m.s)
ρ	Density (kg/m^3)
ψ	Dimensionless drive coefficient

Abbreviations

ALE	Arbitrarily Lagrangian-Eulerian scheme
aVG	Active vortices generation
FC	Forced convection
FEM	Finite Element Method
FSI	Fluid-Structure-Interaction analysis
NC	Natural convection
OSR	The oscillating ratio of piezofan
OA	The oscillating amplitude at the tip of the piezofan (m)
pVG	Passive vortices generation
TBL	Thermal boundary layer
VGS	Vortex Generations Strategies
w/o piezofan	The channel without piezofan

DECLARATIONS

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AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

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