

Numerical Heat Transfer, Part B: Fundamentals

An International Journal of Computation and Methodology

ISSN: 1040-7790 (Print) 1521-0626 (Online) Journal homepage: www.tandfonline.com/journals/unhb20

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To cite this article: Tarek Nehila, Muneer Ismael, Elhadj Benachour & Mohammad Ghalambaz (2026) Characterization of natural convection in a circular cavity with flexible fin subjected to sinusoidal thermal loading, Numerical Heat Transfer, Part B: Fundamentals, 87:1, 2391494, DOI: [10.1080/10407790.2024.2391494](https://doi.org/10.1080/10407790.2024.2391494)

To link to this article: <https://doi.org/10.1080/10407790.2024.2391494>



Published online: 30 Sep 2026.



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Characterization of natural convection in a circular cavity with flexible fin subjected to sinusoidal thermal loading

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ABSTRACT

This problem simulates engineering problems that humans face in practical life such as circulating fluid in indoor novel enclosure and dry (or steam) thermal sterilization of medical equipment. The investigation focuses on a circular cavity heated and cooled by arched walls and incorporating a flexible fin fixed at the bottom point of the enclosure. The new concept explores the impact of the oscillatory variation of a heat source attached to the free end of the fin, following a sinusoidal pattern. We utilize the Galerkin finite element method to solve this dimensionless nonlinear problem. To ensure the reliability of the numerical results, validation against experimental data is conducted. The various parameters considered are: the angular frequency F (1–4), the amplitude of hot spot of the fin tip A (0.1–0.25), the width of the oscillating fin B (0.01–0.025) and the length of the oscillating fin L (0.01–0.1). The left wall heat source is investigated with three arced lengths: θ ($\pi/3$, $2\pi/3$, π). These parameters are studied concerning the average Nusselt number, isotherms, and streamline patterns. The results indicate that the average Nusselt number elevates when the left heat source expands to occupy a larger segment of the left part of the circular cavity, where for an angle equal to π , the heat transfer increases by 81% compared to a heat source occupying only $\pi/3$. However, it dwindles when the amplitude of the tip source increases. Furthermore, higher angular frequencies stimulate stronger convection flows. It is also found that the length of the oscillating fin should not go be excessive.

ARTICLE HISTORY

Received 7 April 2024
Revised 21 July 2024
Accepted 7 August 2024

KEYWORDS

Circular cavity; fluid–structure interaction (FSI); natural convection; simulation numeric; sinusoidal heat source

1. Introduction

Applications of natural convection in civil and indoor energy management are a crucial demand to benefit humanity. To better understand and enhance the physical phenomena of the natural convection, this study rely on coupling disciplines to solve posed problems or to apply performances to engineering systems. Among these couplings, thermal coupling, convection, is found through the new fluid-structure interaction approach studying the mutual influence between the fluid and the structure. The importance of this topic has led to several studies examining the effect of elastic structures such as fins and membranes, and different types of fluids on enhancing natural convection heat transfer.

Nomenclature

Latin

A, a	Dimensionless and dimensional amplitude of fin tip temperature
d	cavity diameter
E*	The dimensionless Young modulus
F*	The dimensionless transformation gradient tensor
F, f	Dimensionless and dimensional frequency of fin tip temperature
g	Gravitational acceleration
k	Thermal conductivity
l, b	Length and thickness of the fin
Nu	Average Nusselt number
P	Dimensionless Pressure
Pr	Prandtl number
Ra	Rayleigh number
S*	Nondimensional tensor of stress
t	Dimensionless time
T	Dimensionless temperature
u*, v*	Dimensional velocity components
w	The moving coordinate velocity
X,Y	The dimensionless Cartesian coordinates
Xs	The dimensionless displacement of moving heat source

Greek symbols

α	thermal diffusivity
β	The coefficient of thermal expansion
ε	Strain
θ	hot wall angle
μ	Dynamic viscosity
ρ	Density
σ^*	Stress tensor
τ	Dimensionless period of heat source movement
ν	Kinematic viscosity
Ψ	Nondimensional stream function

Subscript

c, h	Cold, hot
f, s	Fluid, Solid
r	Property ratio

Abbreviation

Nuavg	Average Nusselt number
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Among these studies, there is an investigation conducted by Ghalambaz *et al.* [1], where they addressed the sinusoidal movement of thin fins on the distribution of isotherms, streamlines and the Nusselt number. The findings indicated that the structure displacement, as well as the increase in the amplitude of its movement, leads to a significant improvement in the heat transfer rate. In the same context, a study presented by Ismael and Jasim [2] deals with mixed convection, where they found that the presence of flexible fins can increase the transfer rate by up to 230% compared to a cavity without structure. A cavity with an oblique geometry and a partial heat source is presented by Alsabery *et al.* [3]. Among the findings, an increase in the inclination angle (φ) results in a reduction in the Nusselt number. For positive values of the angle, a significant reduction occurs. The effect of non-Newtonian fluids is studied by Shahabadi *et al.* [4]. The results show that an increase in viscosity leads to higher stresses on the fin interfaces but does not affect the Nusselt number. A study on the influence of two flexible fins and a nanofluid on natural convection is addressed by Jamesahar *et al.* [5]. Among the findings, the nanofluid and fin oscillation can increase the heat transfer rate compared to the case of a pure fluid. The effect of a flexible deflector on heat transfer is studied by Mazharmanesh *et al.* [6], where they found that the transfer rate is maximal for a quasi-stable rigid deflector.

An in-depth study of complex geometries (corrugated) is conducted by Tarek *et al.* [7]. They integrated a geometry ratio Ar (dimensionless length of corrugated wall) into the fundamental equations of fluids and solids as well as in the equation of the heat transfer coefficient h . The important results of this study indicate that increasing the Ra number contributes to a significant improvement in the heat transfer rate, while the Ar number is a tool to control the transfer. A recent study on the location of the heat source (three different positions of the fin) on the transfer diffusion is conducted by Tarek *et al.* [8]. They discovered that the Nusselt number decreases by 4.8% when the source is at the center of the fin. The coupling of fin movement and temperature leads to a significant disturbance in the fluid flow distribution and isotherms. The impact of

horizontal deflectors on transfer is studied by Haghani *et al.* [9]. The results show that the symmetric double deflector has a symmetric case and exhibits a higher Nusselt number compared to the asymmetric case. The case of ferrofluid on the phenomenon is studied by Aly and Raizah [10], where they deduced that adding nanoparticle concentration contributes to a reduction in the ferrofluid flux intensity. The effect of structure on a cavity partially filled with a porous medium is studied by Saleh *et al.* [11], where they found that increasing the Darcy number, fluid layer thickness, and fluid elasticity enhances the heat transfer rate. The previous studies elaborate on the coupling of FSI with the natural convection phenomenon for several axes. There are also several studies dealing with the same axes but in relation to membranes. The study of [12, 13], and [14] considers a trapezoidal and square enclosure divided by a flexible partition. Another study [15] deals with membrane elasticity in a circular-shaped cavity. The results show that a decrease of 8.5% in the average Nusselt number for $\eta = 90$ was reported. The effect of a hot flexible inside the cavity is studied by Mehryan *et al.* [16]. Their findings highlight that both the fixed point location and the inclination angle significantly impact heat transfer and fluid circulation within the cavity.

For circular cavities, recent studies based on the effect of this geometry on the improvement of heat transfer, such as the one conducted by Biswas *et al.* [17], have investigated the impact of wall curvature in semi-circular annular necks on magneto-nanofluidic flow. The findings indicated that the intensity of the magnetic field affects the production of irreversibility and heat transfer. Another investigation by Chatterjee *et al.* [18] examined the position of discrete radiators and coolers in a cylindrical cavity. The cross-over arrangement of the heater and cooler was found to be the most efficient heating-cooling option for achieving optimal heat transfer. The research by Alizadeh *et al.* [19] presented the heat transfer from the surface of a cylinder covered with a catalyst and subjected to an incident flow of Casson rheological fluid. They recorded a significant reduction in the average Nusselt number during the transition from Newtonian to non-Newtonian fluid. The study by Manna *et al.* [20] aimed to systematically address the positional impacts of the heater and cooler using four different quadrant cavities filled with hybrid nanofluid. Among the results, it was determined that the arrangement in the second or third quadrant could be chosen as the best heating-cooling position configuration to achieve superior heat transfer. Finally, the study by Chatterjee *et al.* [21] examines heat transfer and entropy generation in the presence of a magnetic field and a conductive cylindrical obstruction. It was found that the Nusselt number (Nu) is greatest when the size of the internal conductor block is smallest.

Despite these in-depth studies on the coupling of the two phenomena, fluid–structure interaction and natural convection, the study of circular geometry enclosures equipped with a flexible fin and the effect of thermal coupling of sinusoidal temperature has not been addressed in the open literature yet. This scientific gap motivates us to delve into our investigation of this subject. In the real-world, the circular enclosure is encountered in cylindrical enclosures used in dry and steam thermal sterilization of medical instrumentations [22] and in fluid circulation in cylindrical reservoirs [23]. Hence, the outcomes of the current study are sought to serve a broad range of humanity applications.

2. Mathematical formulation

In this research, we are interested in the phenomena of natural convection within a circular cavity of diameter $D = 1$. We analyze the interaction between air and a structure (fin) located in the lower half of the cavity. The temperature of the right and left circular walls of the cavity is kept constant at T_c^* and T_h^* , respectively. The left hot temperature expands over a segment of θ indicating a length of $\theta D/2$. The free end of the fin is followed by a sinusoidal movement X_s to ensure a vibration over time thus maintaining a sinusoidal temperature T_s varies between 0 and 1, as shown in Figure 1.

$Ra = g_v \beta (T_h - T_c) L^3 / \nu_f \alpha_f$ (Rayleigh number: the effects of buoyancy and natural convection in a fluid), and $Pr = \frac{\nu_f}{\alpha_f}$ (Prandtl number: viscosity influences heat transfer in a fluid).

Regarding the solid domain (the fin), consider a slender fin crafted from isotropic elastic material, where its nonlinear geometric changes are taken into consideration due to the pressure exerted by the fluid

Solid displacement equation

$$x_s = 0.1 \sin(2\pi t / 0.1) \quad (5)$$

The elastodynamic displacement equations

$$\frac{1}{\rho^r} \frac{\partial^2 u_s}{\partial t^2} = E \nabla \sigma + E \cdot F_v \quad (6)$$

The energy equation of the flexible elastic fin:

$$\frac{\partial T}{\partial t} = \alpha^r \nabla^2 T \quad (7)$$

where ρ^r , C_p^r , and K^r are the ratio parameters for density, specific heat, and thermal conductivity solid–liquid, respectively. Thus, $F_v = D(\rho^f - \rho^s)g/E^*$ is the body force source term (The effect of the buoyancy force on the fin is disregarded), and $E = E^* D^2 / \rho^f \alpha_f^2$ is the flexibility parameter

The stress tensor is represented as:

$$\sigma^* = J^{-1} F^* S^* F^{*T} \quad (8)$$

where $J^{-1} = \frac{1}{|F^*|} \text{Adj } F^*$ the inverse of a matrix is $F^* = I + \nabla u_s^*$ the gradient tensor of the transformation, the second Piola–Kirchhoff tensor is defined by the following equations:

$$\varepsilon^* = \frac{1}{2} \left(\nabla u_s^* + (\nabla u_s^*)^T + (\nabla u_s^*)^T \nabla u_s^* \right), \quad S^* = \mathbb{C}(E^*, v) : (\varepsilon^*) \quad (9)$$

The colon “:” is the double-dot tensor product.

2.1. The conditions delineating the interaction between fluid and solid boundaries

Here, the interaction between the fin and the contacting fluid may also be described in dimensionless format, we consider that the regular nonslip boundary condition for the fluid at the solid interface, as follows:

$$\begin{aligned} u_f &= \partial u_s / \partial t \\ \left[-PI + Pr(\nabla u_f + (\nabla u_f)^T) \right] \cdot n &= E \sigma \cdot n \\ \nabla T \cdot n|_f &= K^r \nabla T \cdot n|_s \end{aligned} \quad (10)$$

2.2. Boundary conditions

Considering the boundary conditions, the thermal and dynamic characteristics of the walls are expressed in dimensionless coordinates as follows:

- the initial temperature in the fluid and solid is $T = 0.5$ thus the speed and pressure are zero
- At the left circular segment: $T = 1$,
- At the right circular wall: $T = 0$ (11)
- At the free end of the fin, we maintain a sinusoidal temperature T_s with an absolute function to guarantee that the temperature is always positive and between 0 and 1, this external temperature present as follows:

$$T_s = \text{abs}(A \sin (Ft)) \quad (12)$$

where F and A represent the angular frequency and the amplitude of temperature, respectively.

The parameter of interest is the local and the average Nusselt numbers on the left circular wall to quantify the strength of the natural convection at any point over the hot wall. It can be evaluated as:

$$Nu = hD/k^f \quad (13a)$$

and average Nusselt number are expressed as:

$$Nu_{local} = \frac{dT}{dn} \quad (13b)$$

$$\overline{Nu}(t) = \frac{1}{\theta D/2} \int_0^{\theta D/2} Nu_{local} dS \quad (13c)$$

$$\overline{Nu} = \int Nu_{local} d\tau \quad (13d)$$

The stream function is introduced as:

$$u_f = \partial\psi / \partial y, v_f = -\partial\psi / \partial x \quad (14)$$

3. Numerical Simulation

In this study, finite element analysis is utilized for the numerical simulation of this problem. The nonlinear fundamental equations of this fluid-structure phenomenon are described by the weighted Galerkin finite element method and solved numerically, with a detailed presentation of this method provided in [24, 25]. This technique facilitates precise management of both linear and curved boundaries. It employs basis functions to characterize the unknown variables at the nodes of each element. These basis functions serve to interpolate the values of pressure, temperature, and velocity.

$$u_f, v_f \approx \sum_j^N (u_f, v_f)_j^k \phi_j^k(X, Y), T \approx \sum_j^N T_j^k \phi_j^k(X, Y), P \approx \sum_j^N P_j^k \phi_j^k(X, Y) \quad (15)$$

k, j , and N denote the time iteration, the node number, the total number of nodes, respectively

In particular, a weak version of the governing equations was applied across the mesh elements to formulate residual equations, which were then addressed using a second-order computational algorithm. The Arbitrary Lagrangian–Eulerian (ALE) method is used to capture the movement of the structure and ensure mesh stability throughout the simulation, with a complete description of this method presented in the references of [26] and [27]. The numerical computation estimates an error lower than 10^{-6} .

3.1. Mesh testing

In this stage, we present the mesh model used in this study (as depicted in Figure 2), where we employed a triangular mesh for the air and a quadratic mesh for the boundaries of the circular cavity and the solid. To ensure that the numerical results are independent of the mesh type, a mesh sensitivity test is conducted for four different cases, as presented in Table 1. For each case,

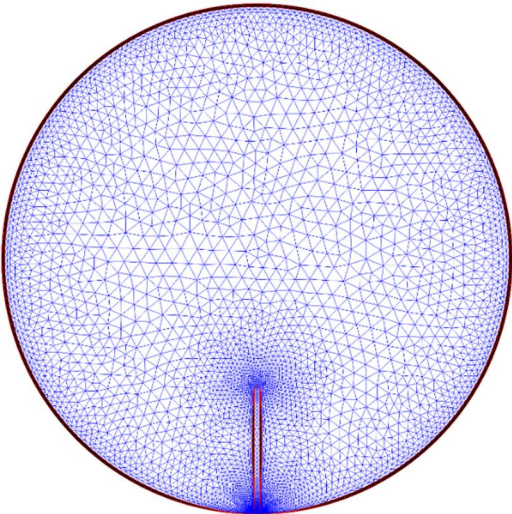


Figure 2. Displays the constructed mesh of the model under examination.

Table 1. Sensitivity testing of the mesh involves examining various mesh sizes.

	Cases			
	Case1	Case2	Case3	Case 4
Domain elements	4408	5580	7462	14,834
Boundary elements	540	592	720	812

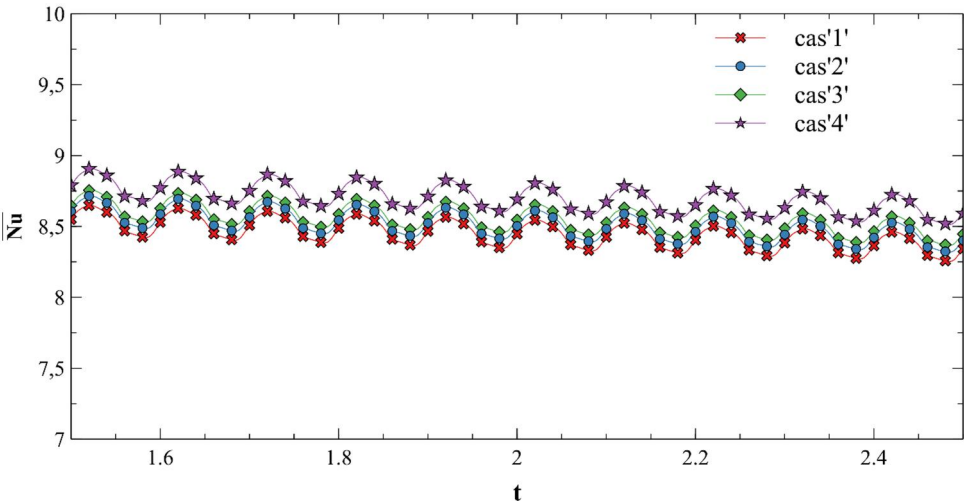


Figure 3. Grid testing of average Nu on the hot wall with different grid cases at $F=0.1$, $A=1$, $L=0.25$, $\theta=\pi$, $Ra=1E6$, $E=1E11$ and $Kr=10$.

the number of boundary and domain elements are presented. Figure 3 depicts the variation of the average Nusselt number with respect to the mesh type for parameters $F=0.1$, $A=1$, and $L=0.25$. It is observed that the mesh type impacts the numerical results; for cases 1 and 4, the difference is significant, on the order of 0.4. Therefore, increasing the number of elements from case 1 to case 4 corresponds to an increase in precision. Consequently, we have chosen the mesh case 4 to present all the numerical results of our study.

3.2. Validation

We conducted a validation study by comparing our current numerical results with existing experimental and numerical data to demonstrate the reliability of the results presented in the following section.

Figures 4 and 5 present a numerical validation with the study of [8], where they investigated the natural convection phenomenon in a semi-circular cavity with a radius of 30 mm experimentally and numerically. They varied the Rayleigh number and inclination angle for a Prandtl number of 6.2 (distilled water). By comparing the streamlines' results (Figure 4) and the isotherms (Figure 5), we can infer a good agreement between the present study and the previous one.

In a second validation (Figure 6), we compared our results with a previous study on natural convection in a partially heated square cavity, conducted both numerically and experimentally. Our validation was based on the local Nusselt number, a crucial parameter in our study. The results of this comparison are presented in the figure, where we fixed the parameters at

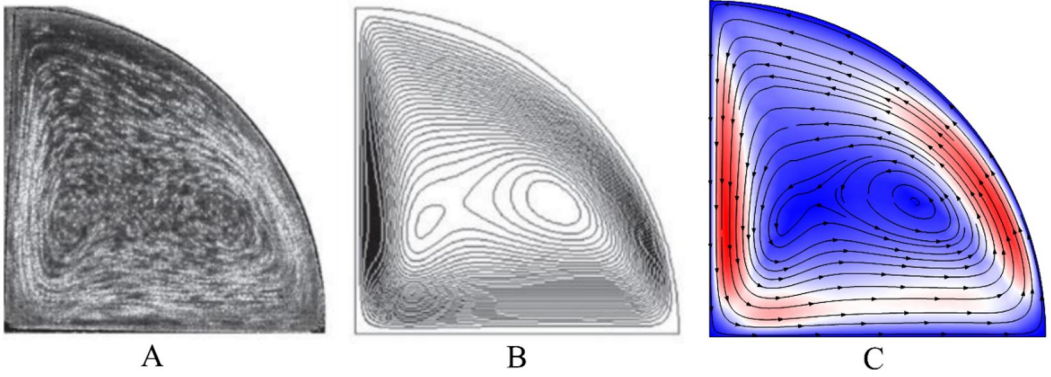


Figure 4. Comparison of streamlines (A) and (B) experimental and numerical results of [8] and (C) the present numerical results for $Ra = 10^6$ and $Pr = 6.62$.

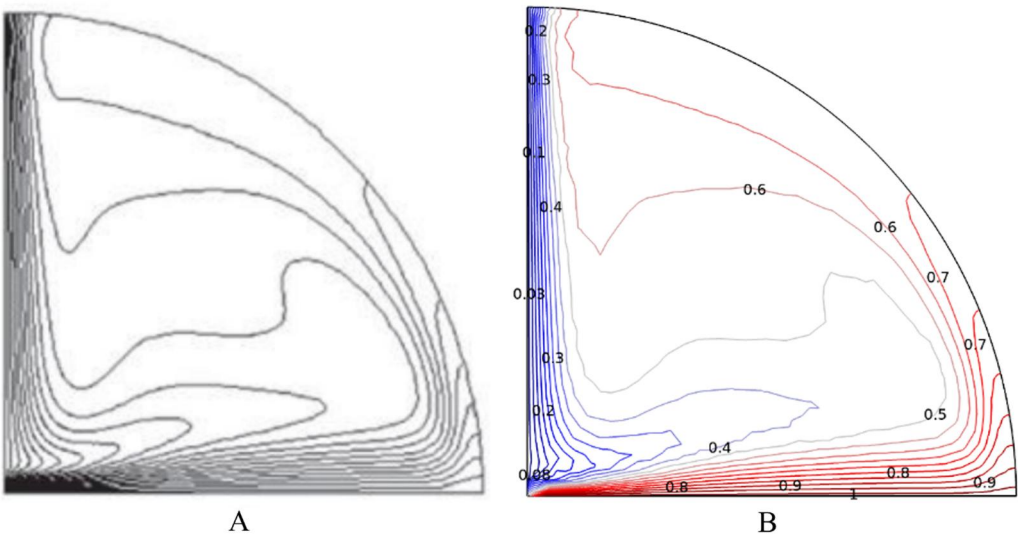


Figure 5. Comparison between the isotherms, where (A) represents the numerical result from [8] and (B) denotes the present study for $Ra = 10^6$ and $Pr = 6.62$.

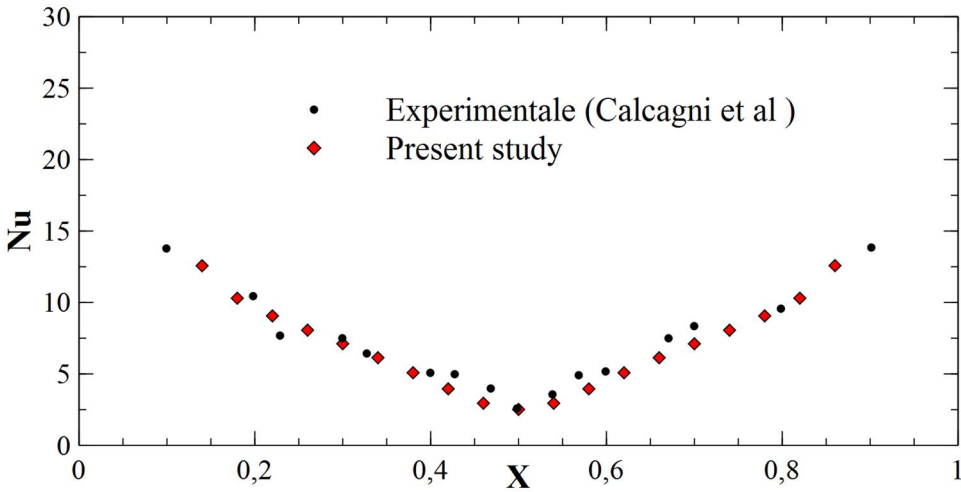


Figure 6. The local Nusselt number comparison in the experimental work conducted by Calcagni *et al.* [28] and the current numerical study.

$Ra = 1.2 \times 10^5$ and a source length equal to 0.8 times of the enclosure bottom wall. Our results fall within the ideal range, confirming the robustness of our approach.

4. Results and discussions

This part of the study deals with the numerical results and discussions relating to various parameters influencing the phenomenon of natural convection within the circular cavity with a sinusoidal heat source. The dimensionless parameters considered include the angular frequency (F) at values of 0.1, 0.2, 0.3, and 0.4, the source amplitude (A) ranging from 0.4 to 1 and the fin length (L) ranging from 0.2 to 0.35. The time interval considered is $1.5 < t < 2.5$, while maintaining the constants: Young's modulus ($E = 10^{11}$), conductivity ratio ($Kr = 10$), Prandtl number ($Pr = 0.71$) and Rayleigh number ($Ra = 10^6$) constant across all simulations.

4.1. Frequency effect and dimensionless amplitude of temperature

Figure 7 illustrates the effect of the angular frequency F (0.1, 0.2, 0.3, and 0.4) of the heat source on the average Nusselt number as a function of dimensionless time. According to the results, for each frequency F , the average Nusselt number oscillates in sinusoidal fashion and gradually decreases over time. This can be justified by two elucidations, the first is that the sinusoidal heating arising from the fin's end source, which heats up the lower part of the circular cavity. Consequently, the temperature difference between the heated wall on the left and the air becomes smaller, resulting in reduced heat exchange. The second elucidation is that the oscillation motion of the fin itself creates the peaks of Nusselt due to the proximity and farness of the fin from the hot wall which destructs the thermal boundary layer. It can also be observed that increasing the angular frequency of temperature leads to a decrease in the average Nusselt number. This decrease is not constant but becomes significant over time, as the increase in frequency accelerates temperature changes at the fin and speeds up fluid heating.

Figure 8 depicts feasibility of fin oscillation on heat transfer (with an amplitude of 0.25) compared to a stationary fin. The results demonstrate the aforementioned role of the periodic proximity and farness of the heat source from the hot and cold arched walls which significantly affects the thermal boundary layer on the hot arched wall. For a rigid fin, the Nusselt number is

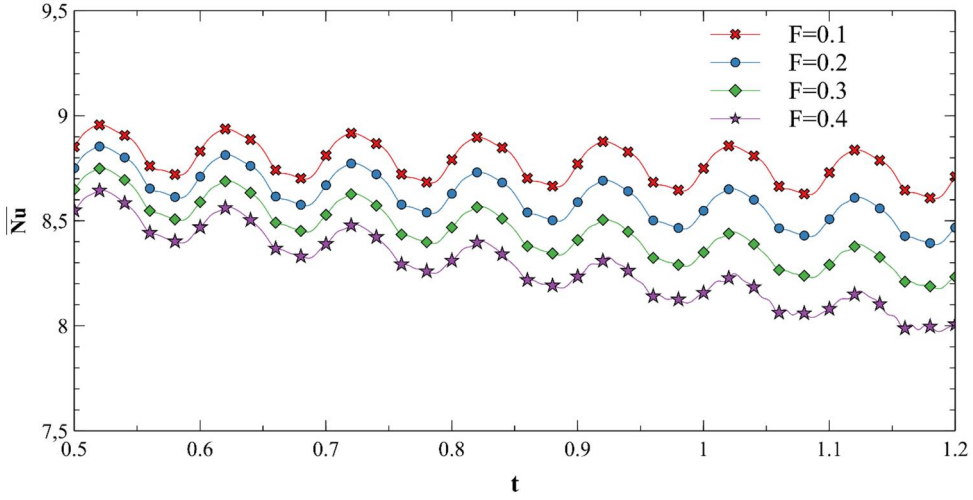


Figure 7. The influence of the non-dimensional frequency F on average Nu across dimensionless time at $a = 1$, $L = 0.25$, $\theta = \pi$, $Ra = 1E6$, $E = 1E11$ and $Kr = 10$.

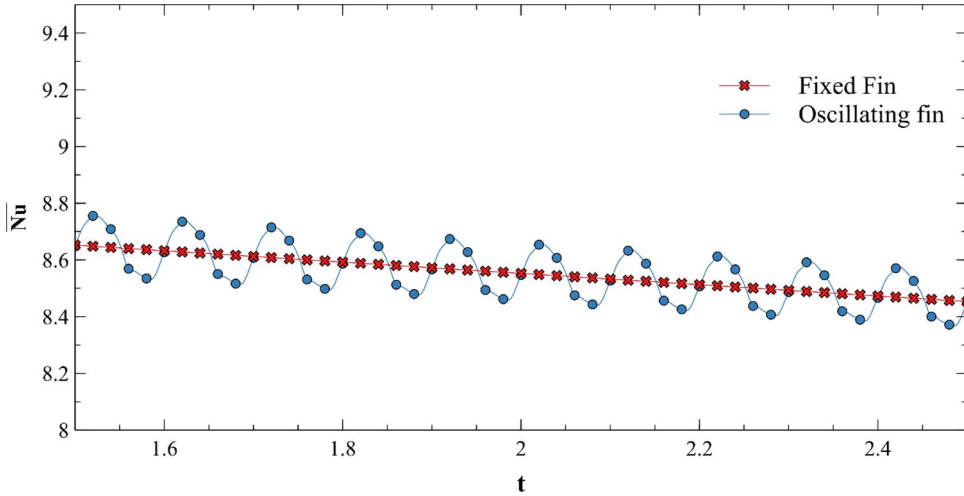


Figure 8. The influence of moving fin on average Nu across dimensionless time at $L = 0.25$, $\theta = \pi$, $Ra = 1E6$, $E = 1E11$ and $Kr = 10$.

linear. Thus, an oscillating structure can provide peak heat transfer while the fixed structure may provide a physical solution to maintain stable transfer over time. The decay pattern of the figure refers to the heating up of the fluid with time and the declining temperature difference between the hot fluid and the hot arched wall.

Figure 9 portrays the effect of sinusoidal temperature amplitude on the average Nusselt number as a function of dimensionless time. For each amplitude, it is noticeable that the curve repeats for each time interval of 0.1, where the effect of structural movement is evident. Thus, an increase in temperature amplitude “ A ” exhibits a negative effect on the enhancement of the Nusselt number, with minimal values recorded for $A = 1$. This result can be justified by the fact that as the temperature amplitude increases, the temperature at the fin’s end becomes significant, leading to an elevation in temperature in the lower part of the circular cavity which reduces the temperature difference between the fluid and the arched wall resulting in a decrease in heat exchange between the air and the hot wall.

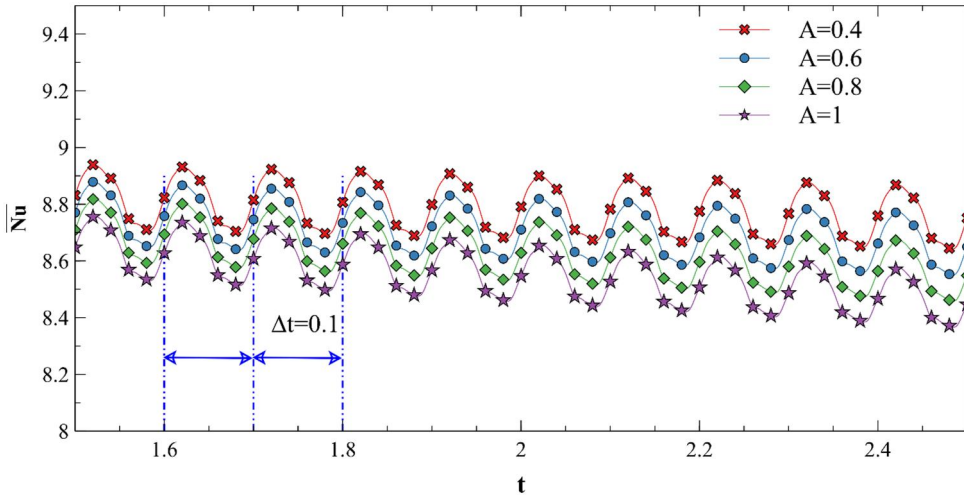


Figure 9. The influence of the dimensionless amplitude on average Nu over dimensionless time at $a = 1$, $F = 0.1$, $L = 0.25$, $\theta = \pi$, $Ra = 1E6$, $E = 1E11$ and $Kr = 10$.

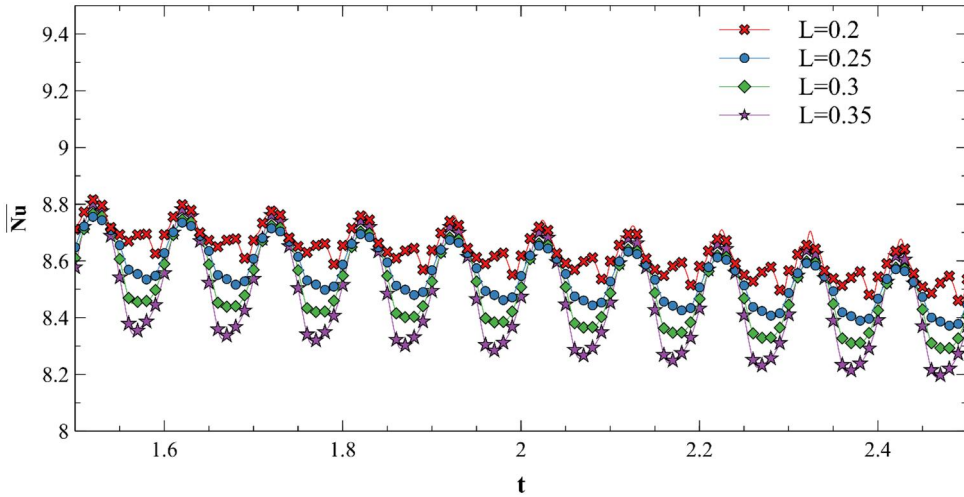


Figure 10. The influence of the dimensionless length of fin on average Nu across dimensionless time $a = 1$, $F = 0.1$ and $\theta = \pi$, $Ra = 1E6$, $E = 1E11$ and $Kr = 10$.

4.2. The effect of the fin dimension

The impact of the fin length L is depicted in Figure 10. It can be observed that over time, the curve of the Nusselt number repeats periodically, which is due to the periodic movement of the fin. For $L = 0.35$, corresponding to the maximum length of the fin, the curve exhibits a maximal amplitude, while the opposite occurs for the shorter length of $L = 0.2$. It is also noticeable that a reduction in the fin length leads to an improvement in heat transfer. This can be justified by the fact that a longer length contributes in disrupting the natural fluid flow from bottom to top, thereby decreasing the efficiency of transfer. Moreover, too long fins may act as a blocker and isolate the convective heat transfer between the hot and cold arched walls, the reason of the reduction of the Nusselt number.

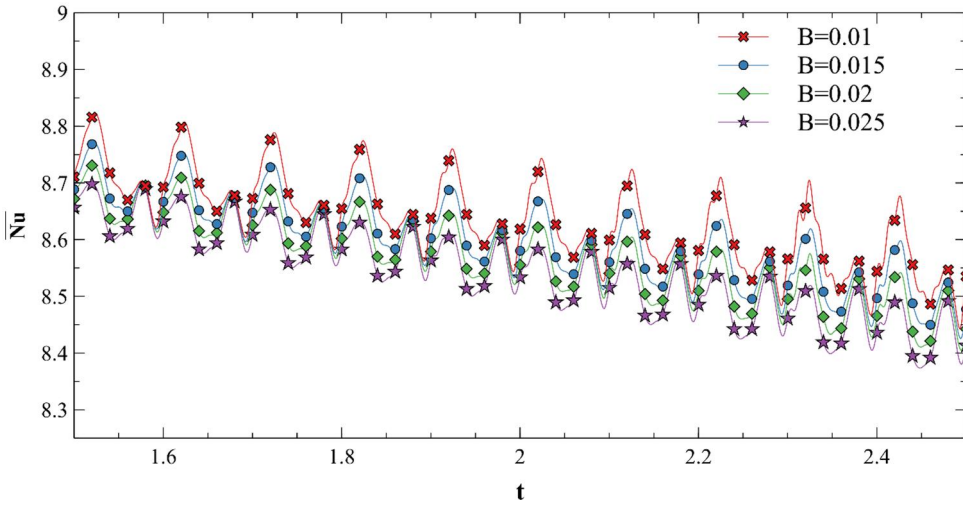


Figure 11. The influence of the dimensionless width of fin on average Nu across dimensionless time at $\alpha = 1$, $F = 0.1$ and $\theta = \pi$, $Ra = 1E6$, $E = 1E11$ and $Kr = 10$.

The Nusselt number variations with the fin width are shown in Figure 11. It can be observed that for thicker fin, $B = 0.025$, the average Nusselt number is minimal. For a thinner fin ($B = 0.01$), maximal transfer is recorded, manifested by a maximal average Nusselt number. The physical explanation of this effect can be attributed to the strain associated with the fins. The thinner fin exhibits more strain and therefore manifests more modulation forms. These modulations in fin provide effective mixing of fluid and hence boosts the exchange of momentum. Thus, decreasing the width of the fin elevates the peaked Nusselt number while increasing the fin width can hinder the elastic movement of the structure, leading to a negative effect on the enhancement of heat transfer by natural convection.

Figure 12 illustrates the impact of the fin length L on the distribution of isotherms within the circular cavity for different fin modulations. It can be observed that the isotherms concentrate near the two arched walls where the heat source (hot) and the heat sink (cold) are located and align horizontally. The isotherms deviate to conform to the circular shape of the cavity. The lines indicate that the hot air occupies the upper part of the cavity, unlike the cold air. Apart from the fin, the isotherms look stratified due to the strong convection at $Ra = 10^6$. Conversely, the flagging of the fin has an impact on the distribution of isotherms, disrupting the stratification area surrounded by the fin. Increasing the length of the fin presents a notable disruption on these lines in the surrounding area, where minimal disruptions are recorded for $L = 0.2$. The presence of temperature at the fin's end affects the lines, where the fluid temperature values follow and depend on the sinusoidal nature of the temperature. This temperature can be used as a new parameter to add specific control inside the cavity, either by the magnitude or by flow direction.

Figure 13 depicts the effect of the fin length on the distribution of streamlines pattern for different fin modulations. From the results, it can be observed that the streamlines are intense near both the hot and cold walls. This is because of the significant difference in fluid density due to the steep temperature gradient. Thus, the circular shape of the cavity contributes to smoother clockwise fluid motion and facilitates fluid circulation. A primary core forms at the center of the cavity due to fluid movement. This fluid motion, along with the presence of the sinusoidal temperature at the free end of the fin, produce new fluid flow patterns. For maximum temperature values at the free end, the fluid absorbs this heat, accelerating its motion.

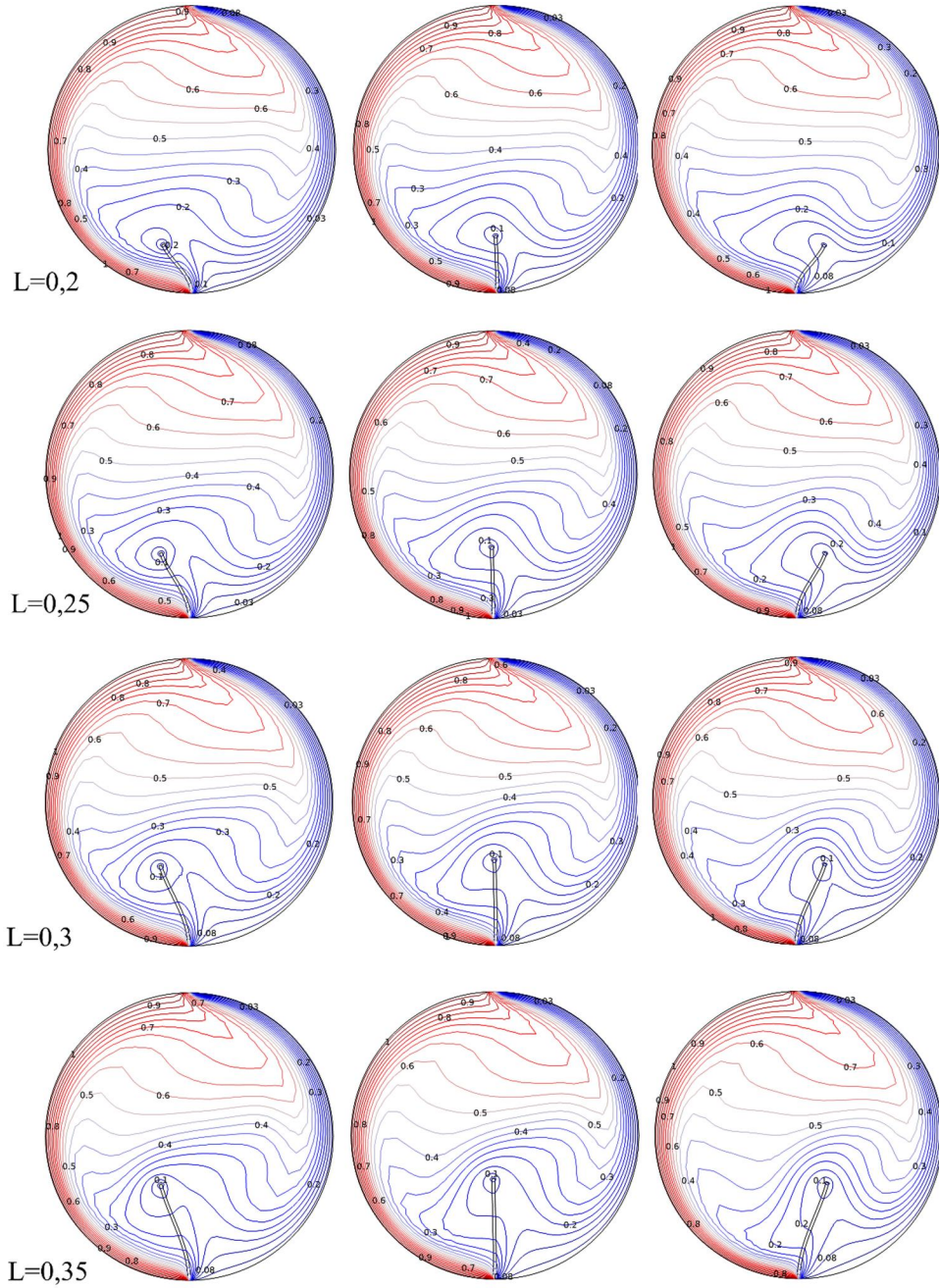


Figure 12. Distributions of isotherms for different fins lengths (L) at $F=0.1$, $A=1$, $\theta=\pi$, $Ra=1E6$, $E=1E11$ and $Kr=10$.

In this geometry, the absence of secondary circulation is noted, unlike in the corrugated cavities of Tarek *et al.* [7] and Ghalambaz *et al.* [1]. The impact of the fin length is evident and clearly visible in the figure. Increasing the fin length hinders the natural fluid circulation and disrupts the lower part of the primary core, thereby adding additional disturbances in the cavity, resulting in a negative effect on heat transfer.

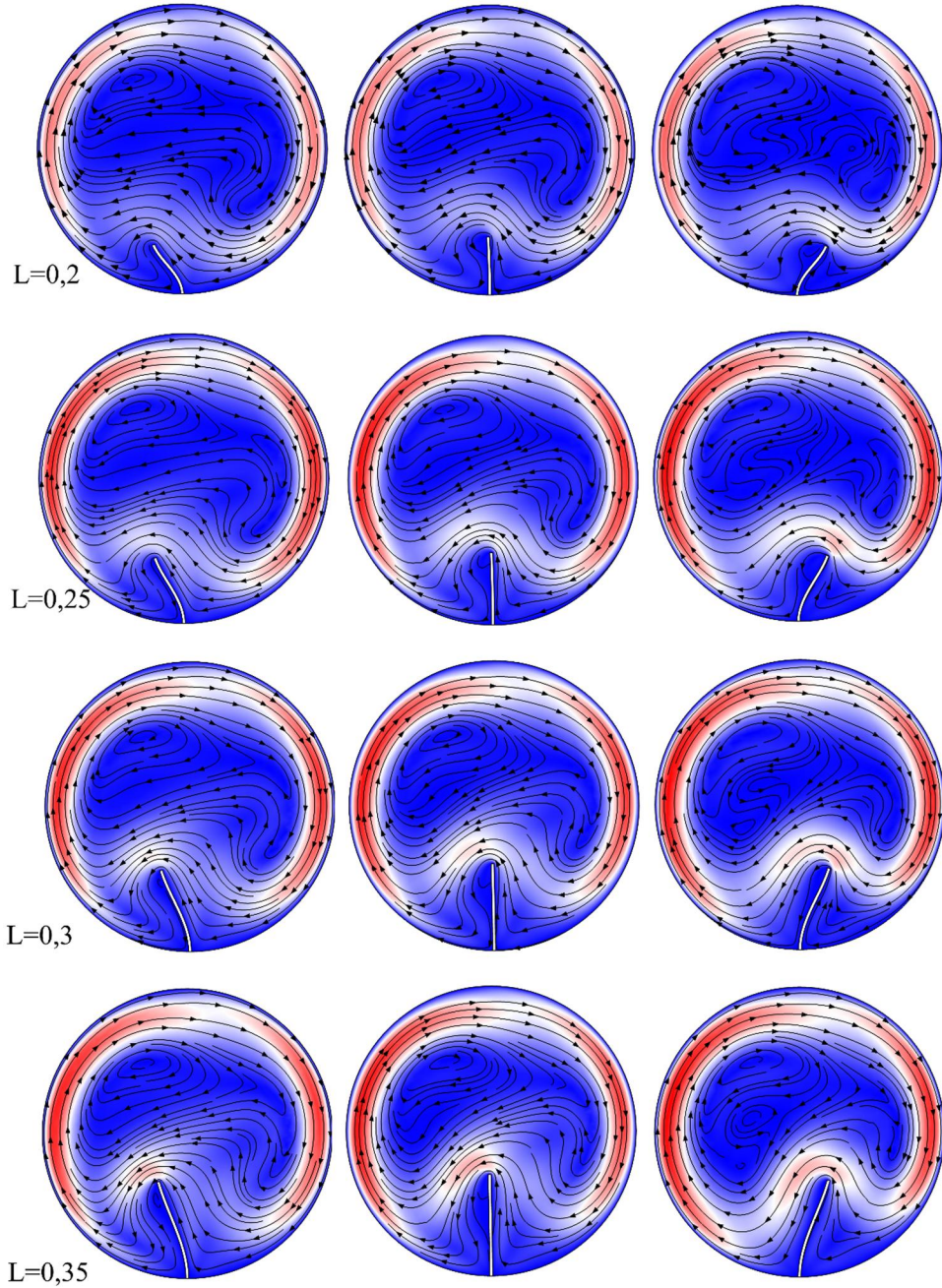


Figure 13. Distributions of streamlines for different fins lengths (L) at $F = 0.1$, $A = 1$, $\theta = \pi$, $Ra = 1E6$, $E = 1E11$ and $Kr = 10$.

4.3. The influence of the heat source segment θ

Figure 14 illustrates the effect of the arc length of the heat source (denoted by angles θ : $\pi/3$, $2\pi/3$, π) on the Nusselt number. It can be observed that for $\theta = \pi$, the heat transfer is maximal, attributed to a larger contact surface area with the air particles, which provides more heat to the air. For an arc length corresponding to $\theta = 2\pi/3$ and $\theta = \pi/3$, it is noticeable that the contact between

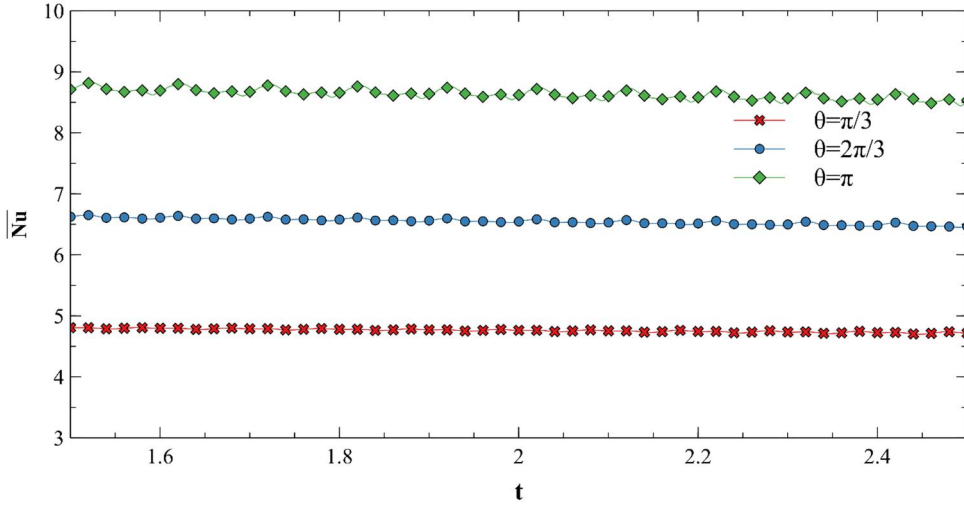


Figure 14. The influence of the heat source segment θ on average Nu over dimensionless time at $a = 1$, $F = 0.1$, $Ra = 1E6$, $E = 1E11$ and $Kr = 10$.

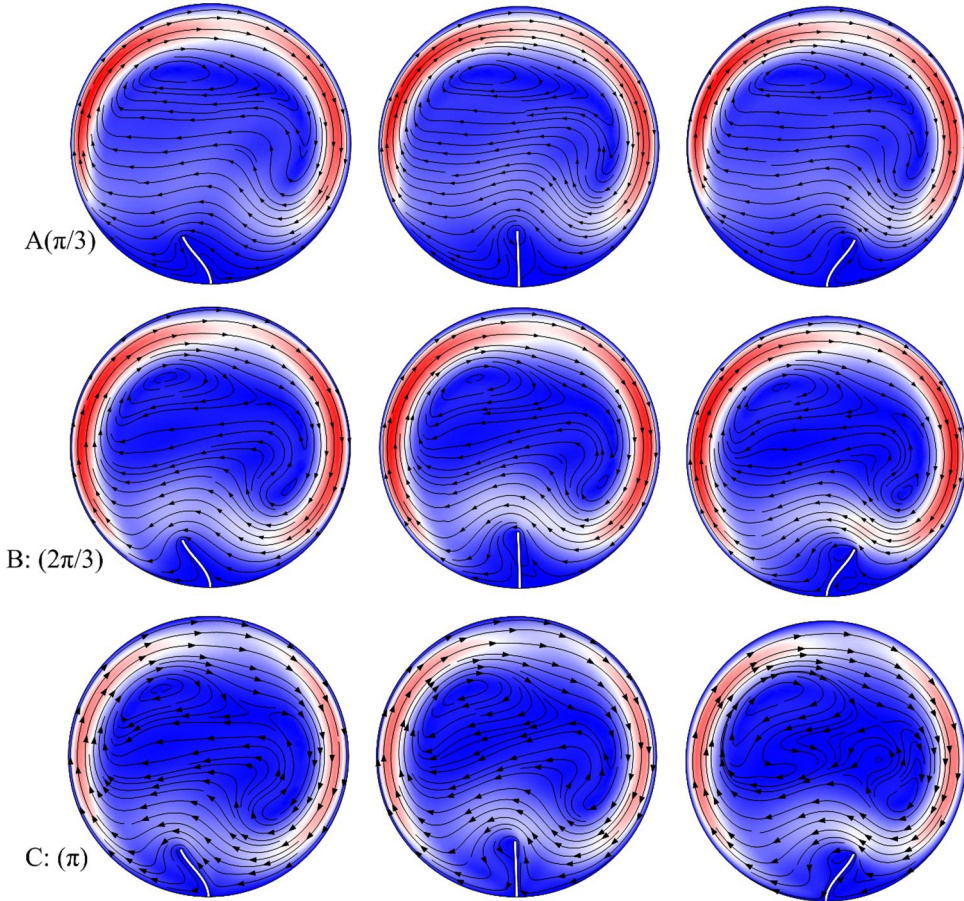


Figure 15. Distributions of streamlines for different heat at $F = 0.1$, $A = 1$, $Ra = 1E6$, $E = 1E11$ and $Kr = 10$.

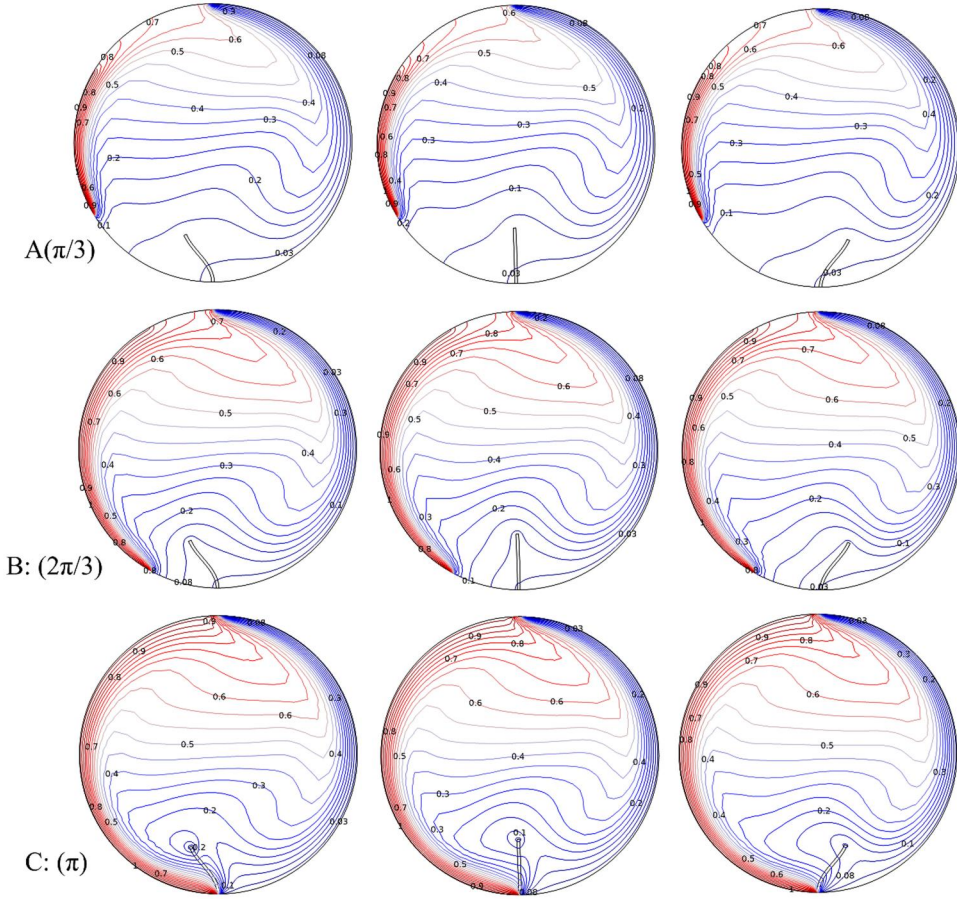


Figure 16. Distributions of isotherms for different heat source at $F = 0.1$, $A = 1$, $Ra = 1E6$, $E = 1E11$ and $Kr = 10$.

the solid and fluid mediums shrink, resulting in a decrease in the Nusselt number. However, the Nusselt number of $\theta = \pi$ is greater than that of $\theta = \pi/3$ and $\theta = 2\pi/3$ by 81% and 32%, respectively. Moreover, the wider heat source ($\theta = \pi$) showcases a sinusoidal Nusselt number as the chance of an influential hot segment increases because the impact of the oscillated fin becomes effective and closer to the heat source.

The impact of θ of the heat source on the distribution of streamline patterns is portrayed in [Figure 15](#) for different fin modulations. For all inspected three angles, it is noticeable that the lines concentrate close to the boundaries of the circular cavity, with this geometry facilitating circular movement of the air. For $\theta = \pi/3$, the appearance of a small primary core in the upper part of the cavity is observed, with horizontal lines moving from right to the heat source. Increasing the length of the source to $2\pi/3$ and π enlarges the primary core, attributed to the increased contact surface area between the air and the heat source, which excites more fluid quantity. The efficiency of the structural movement is evident, where when the fin is deflected the right; the fluid encounters resistance hence, it moves and flows upward, circumventing its movement toward the left wall.

The trends of the isotherms with hot source length are shown eventually in [Figure 16](#) for different fin modulations. It is observed that the isotherms concentrate near the hot source and the cold sink and adjacent to their extents. For $\theta = \pi/3$, the lines align horizontally (stratification behavior) due to the high Rayleigh number ($Ra = 10^6$), and temperatures are below 0.5 in the upper part of the cavity. The movement of the structure is not significant in

the lower part of the cavity. For $\theta = 2\pi/3$ and $\theta = \pi$, with the increase in the size of the source, the isotherm lines become denser, and the fluid temperature elevates in the cavity, especially in the upper part, indicating the significant presence of hot air. The displacement of the structure also becomes more pronounced, as shown by the isotherms following the movement of the fin in both opposite directions, right and left. Stratification behavior is observed in all values of θ .

5. Conclusions

This study delves into the effect of the sinusoidal heat source fixed on the tip of an oscillating fin on the natural convection in a cylindrical enclosure with arched hot and cold walls. The problem is approximated to 2-D circular geometry and solved using the ALE method associated with the weighted Galerkin finite element method. We investigated the influence of several parameters, including the length of the hot arched wall (θ), the length of the fin (L), the width of the fin (B), the amplitude of the fin tip heat source (A), the angular frequency of the heat source (F), as well as the flexibility feature of the fin. We think that the present results contribute in advancing further control to the thermal sterilization process of medical equipment using dry and steam heat treatment in cylindrical vessels. The conclusions of this study are as follows:

- i. The dimensions of the fin have a significant impact on enhancing the Nusselt number and heat transfer, with the optimal dimensions being a width (B) of 0.01 and a length (L) of 0.2.
- ii. The fin should not be too long as it acts as an isolator and blocks the heat transfer, where elongating the fin deteriorates the uniformity of the isotherms and dwindles the Nusselt number.
- iii. The length of the heat source (characterized by an angle θ) contributes to an enhancement in the heat transfer rate. For an extension angle π , the Nusselt number elevates by 81% compared to an extension angle of $\pi/3$. Decreasing the arched wall angle results in a decrease in the primary core of fluid circulation, and the isotherms become scattered, while the fin oscillation becomes less pronounced.
- iv. The sinusoidal temperature at the free end of the fin (characterized by the amplitude A and angular frequency F) has a significant effect on enhancing the transfer. The maximal Nusselt number is obtained for $A = 0.4$ and $F = 0.1$.
- v. The coupling of the fin's modulations and the sinusoidal temperature on the fin tip contributes in expediting the rate of heat transfer compared to a fixed rigid fin.
- vi. The recommended geometrical and operating conditions that give the best heat transfer are: fin length $L = 0.2$, fin width $B = 0.01$, amplitude of the fin tip temperature $A = 0.4$ with a frequency of $F = 0.1$.

Varying the fluid type to comply with a fossil fuel can switch the study to simulate the process of heating of oil reservoirs to keep the fluidity at ambient temperatures. This can be achieved by varying the Prandtl number and/or transition to non-Newtonian fluid, the topic of a future work.

Acknowledgments

- The authors would like to thank all reviewers for taking the time and energy to review our work.
- The authors would like to thank the officials of the Directorate General for Scientific Research and Technological Development (DGRSDT) for having included this journal so that Algerian researchers can contribute to the development of their research in this field.

Disclosure statement

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this article.

Funding

This research received no external funding.

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