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Effect of Fin Geometry on Natural Convection in a Rotated Rectangular Enclosure

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Abstract

The natural convection in enclosures is a significant issue in thermal engineering, and internal fins are widely used to improve heat transfer. This current investigation focuses on a numerical investigation of the effects of fin geometry on natural convection heat transfer in a rotated rectangular enclosure. Both square and triangular fins are considered in this investigation, and their effects on fluid flow, temperature distribution, and heat transfer are presented. Numerical simulation of the problem is performed using the finite element method in COMSOL Multiphysics. The numerical simulation is validated with benchmark solutions, and excellent agreement is found. It is found from this investigation that with an increase in Rayleigh number, the effects of buoyancy force on fluid flow and heat transfer increase for both fins. It is found that triangular fins have significant effects on fluid flow and heat transfer compared to square fins. Also, with an increase in fin length, heat transfer increases for low Rayleigh numbers, whereas excessive length of fins reduces heat transfer for high Rayleigh numbers due to blockage of fluid flow. It is found that triangular fins with moderate lengths provide better heat transfer compared to square fins.

Keywords: Natural convection, Rotated rectangular enclosure, Fin geometry, Rayleigh number, Heat transfer enhancement, Fin length.

1 | Introduction

Natural convection heat transfer in enclosed cavities has been extensively investigated due to its critical importance in a wide range of thermal engineering applications such as electronic cooling, solar collectors, heat exchangers, and energy-efficient building systems. In these configurations, heat transfer is primarily governed by buoyancy-driven fluid motion arising from temperature gradients between the enclosure boundaries.

The resulting flow structure and thermal performance are strongly influenced by geometric parameters, boundary conditions, enclosure orientation, and external forces such as rotation. Among these factors, the

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introduction of internal fins is recognized as one of the most effective passive techniques for enhancing natural convection heat transfer.

Early studies on natural convection in rectangular enclosures focused on understanding the fundamental effects of cavity geometry and boundary heating. Hasnaoui et al. [1] were among the first to investigate rectangular cavities equipped with adiabatic fins attached to the heated wall. Their results revealed that fin length plays a crucial role in determining the average Nusselt number and that an optimal fin length exists beyond which convection is suppressed due to flow blockage. Shortly afterward, Lakhal et al. [2] extended this analysis to inclined rectangular enclosures with perfectly conducting fins and demonstrated that enclosure inclination significantly alters buoyancy-induced flow patterns, especially at high Rayleigh numbers. These studies established the foundation for understanding fin-induced modifications of natural convection in enclosed domains.

Subsequent investigations emphasized the role of enclosure configuration and inclination without rotation. Waheed [3] examined natural convection in rectangular enclosures with one thermally active vertical wall and showed that heat transfer enhancement is strongly dependent on buoyancy forces and cavity geometry. Guo et al. [4] further studied rectangular cavities at various aspect ratios and inclination angles, reporting substantial changes in flow structure and average Nusselt number due to tilting effects. Later, Cianfrini et al. [5] identified an optimal inclination angle that maximizes natural convection heat transfer in air-filled rectangular enclosures, highlighting the strong coupling between gravitational direction and thermal boundary layer development.

Parallel to these efforts, several studies investigated natural convection in non-standard geometries and porous or open cavities. Numerical simulations using the lattice Boltzmann method by Seta et al. [6] demonstrated that porosity significantly affects convection strength and heat transfer efficiency in enclosed media. Similarly, Haghshenas et al. [7] analyzed open-ended cavities filled with porous media and showed that increasing porosity enhances flow circulation and Nusselt number. Investigations into triangular cavities further revealed the existence of critical Rayleigh numbers separating conduction-dominated and convection-dominated regimes [8-10].

With growing interest in heat transfer enhancement techniques, researchers began to focus more explicitly on fin number, spacing, and arrangement. Roy et al. [11] numerically studied natural convection in a finned rectangular enclosure heated from below and cooled from above, showing that although increasing the Rayleigh number enhances heat transfer, an excessive number of fins can hinder fluid motion and reduce overall performance. A comprehensive parametric study by Krishnayatra et al. [12] confirmed that for a fixed fin number, an optimal combination of fin length, thickness, and orientation exists that maximizes convective effectiveness. These findings underline the importance of balancing heat transfer surface area with flow accessibility.

In addition to conventional fins, advanced fin geometries and innovative enhancement strategies have been proposed. Awasarmol and Pise [13] experimentally demonstrated that perforated fins can significantly enhance natural convection heat transfer, particularly at specific inclination angles. Chen et al. [14] reported that swirling flow structures generated between plate fins can substantially improve heat dissipation in natural convection regimes. Later, Haghighi et al. [15] introduced novel plate-fin heat sink designs and achieved notable enhancement in heat transfer performance under laminar natural convection conditions.

More recent studies have extended natural convection analysis to nanofluids and complex fin configurations. Sheikhzadeh et al. [16] showed that the combined effects of fin geometry, nanoparticle concentration, and Rayleigh number play a decisive role in heat transfer enhancement, with fin length having a more pronounced influence than fin number at lower Rayleigh numbers. Zhang et al. [17] investigated W-type fin arrays and demonstrated that fin spacing and inclination angle are key parameters governing temperature reduction and thermal uniformity. Gholami et al. [18] further reported that nanofluid-filled channels equipped with dimple fins exhibit significantly higher Nusselt numbers compared to conventional fluids.

Another important aspect affecting natural convection is enclosure rotation. Unlike stationary or inclined cavities, rotating enclosures experience additional inertial and Coriolis forces that alter the balance between buoyancy and viscous effects. Kumar et al. [19] numerically studied natural convection in differentially heated rotating enclosures and showed that rotation can substantially modify flow structures and heat transfer rates, particularly beyond critical rotational speeds. Their results revealed a strong dependence of the average Nusselt number on both Rayleigh number and rotational speed.

Despite the extensive literature on fin geometry, cavity inclination, and advanced enhancement techniques, studies addressing the combined influence of fin geometric parameters and enclosure rotation remain limited. Most previous investigations have considered stationary or inclined enclosures, while the interaction between fin-induced flow modification and rotation-induced forces has received comparatively little attention.

Therefore, the present study aims to numerically investigate natural convection in a rotated rectangular enclosure equipped with internal fins, with particular emphasis on the effects of fin length, number, and arrangement under different rotation angles and Rayleigh numbers. The outcomes of this study are expected to provide deeper insight into the coupled mechanisms governing flow and heat transfer and to offer practical guidelines for the optimal design of finned enclosures in rotating thermal systems.

2 | Physical Model and Governing Equations

In the present study, natural convection heat transfer within a two-dimensional rectangular enclosure inclined at an angle α with respect to the horizontal axis is numerically investigated. The enclosure has a width W and a height H. The flow is assumed to be laminar, incompressible, and Newtonian, and the Boussinesq approximation is employed to account for buoyancy effects.

No-slip boundary conditions are imposed on all solid walls, such that the velocity components vanish at the enclosure boundaries. The top and bottom walls are considered adiabatic and thermally insulated. The left vertical wall is maintained at a lower temperature Tc, while the right vertical wall is kept at a higher temperature Th.

Since the governing equations are solved in a dimensionless form, the dimensionless temperatures of the cold and hot walls are set to 0 and 1, respectively. Heat transfer occurs solely due to natural convection driven by the temperature difference between the vertical walls. To examine the influence of fin geometry on the flow structure and heat transfer characteristics, two different fin configurations are introduced inside the enclosure: a square fin and a triangular fin. The two-dimensional schematic of the rectangular enclosure equipped with a square fin is illustrated in *Fig. 1a*, while the configuration with a triangular fin is shown in *Fig. 1b*.

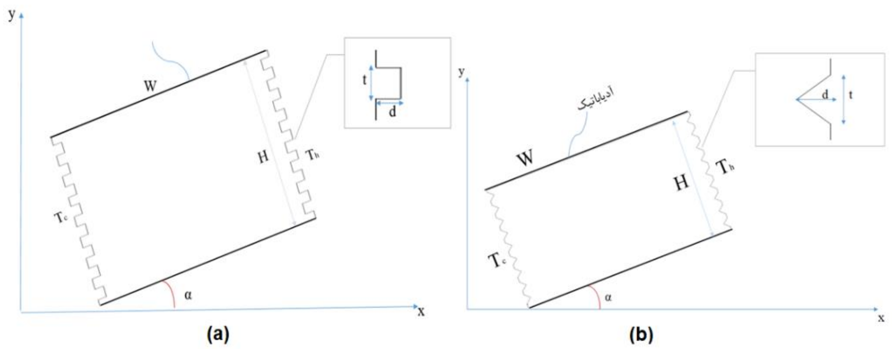


Fig. 1. Physical model.

The natural convection flow inside the inclined rectangular enclosure is governed by the two-dimensional, steady, laminar conservation equations of mass, momentum, and energy. The fluid is assumed to be Newtonian and incompressible, and buoyancy effects are taken into account through the Boussinesq approximation. Thermophysical properties of the fluid are considered constant, except for the density variation in the buoyancy term.

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

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\upsilon\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right)+g\beta(T-T_c)\sin\alpha. \tag{2}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\upsilon\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+g\beta(T-T_c)\cos\alpha. \tag{3}$$

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

where u and v are the velocity components in the x and y directions, respectively, p denotes the pressure, T is the temperature, ν is the kinematic viscosity, α is the thermal diffusivity, g is the gravitational acceleration, and β is the thermal expansion coefficient. The governing equations are rendered dimensionless using the following characteristic scales:

$$X=\frac{x}{H},\quad Y=\frac{y}{H},\quad U=\frac{uH}{\alpha},\quad V=\frac{vH}{\alpha},\quad \theta=\frac{T-T_C}{T_H-T_C},\quad P=\frac{pH^2}{\rho\alpha^2}. \tag{5}$$

Substituting the above dimensionless variables into the governing equations yields the nondimensional forms.

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0. \tag{6}$$

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+Pr\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right)+RaPr\theta\sin\alpha. \tag{7}$$

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+Pr\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+RaPr\theta\cos\alpha. \tag{8}$$

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}. \tag{9}$$

For clarity and reproducibility, the main dimensionless parameters, geometric dimensions, and boundary condition values employed in the present simulations are summarized in *Table 1*.

Table 1. Dimensionless parameters and physical constants used in the simulations.

Parameter Description	Symbol	Value
Dimensionless cold wall temperature	T_c	0
Dimensionless hot wall temperature	T_h	1
Prandtl number	Pr	0.71
Reference pressure	P_0	0
Rayleigh number	Ra	0
Inclination angle with respect to the horizontal	α	10°
Enclosure width	W	1
Enclosure height	H	1.5

3 | Numerical Method

COMSOL Multiphysics's Finite Element Technique (FEM) is used to solve the governing equations and related boundary conditions numerically. The Laminar Flow and Heat Transfer in Fluids modules, which are fully coupled through the buoyancy force represented using the Boussinesq approximation, are used in the simulations. An unstructured triangular mesh is used to discretize the computational domain, and local mesh refinement is applied in areas close to the walls and around the fins to precisely capture the abrupt temperature and velocity gradients. To make sure that the numerical results are independent of the mesh size, a grid independence study is carried out. A trade-off between computational cost and solution accuracy is used to choose the final mesh. Solving the time-independent problem yields steady-state solutions. Convergence is assumed when the relative residuals of all dependent variables drop below 10^{-6}. The numerical solver automatically adjusts the relaxation parameters to ensure robust convergence for all considered Rayleigh numbers and inclination angles.

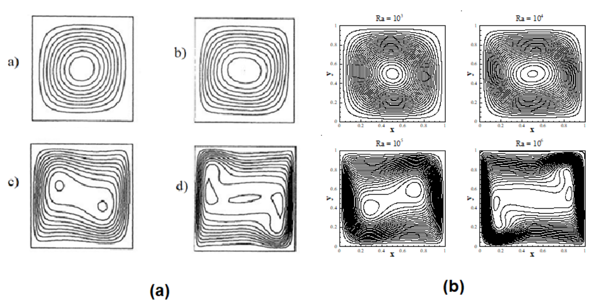
4 | Result and Discussion

The present numerical model is validated against the benchmark study by De Vahl Davis [20] for natural convection in a square enclosure without fins at α=0. *Table 2* compares the average Nusselt number obtained in the present work with the reference results for different Rayleigh numbers. Excellent agreement is observed, confirming the accuracy of the numerical approach.

Table 2. Comparison of the average Nusselt number for a square enclosure without fins.

Ra	Nu (Present)	Nu (Reference)
10^3	1.117	1.116
10^4	2.244	2.234
10^5	4.518	4.510
10^6	8.793	8.798

Qualitative comparisons of streamlines and temperature contours between the present study and the reference are shown in *Figs. 3*. The flow patterns and isotherms exhibit strong similarity, further confirming the validity



of the present model. Overall, the quantitative and qualitative comparisons demonstrate that the present numerical approach reliably captures natural convection heat transfer in enclosed domains.

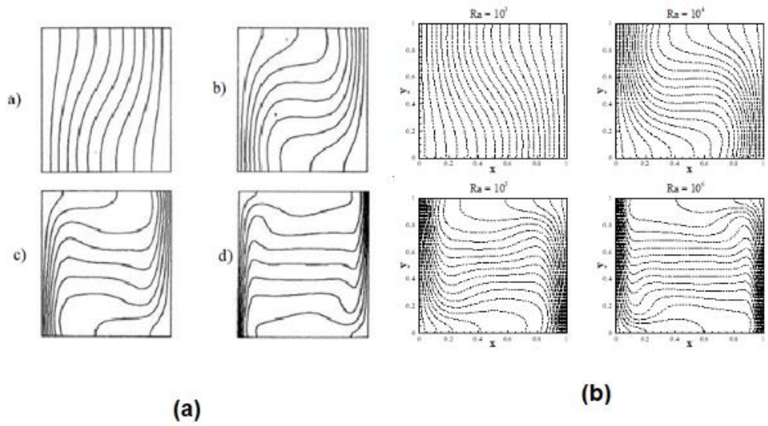


Fig. 3. Comparisons of streamlines and temperature contours between the present study [21].

The effect of fin geometry on natural convection heat transfer in a rectangular enclosure equipped with square and triangular fins under non-rotating conditions is investigated, with particular emphasis on the roles of Rayleigh number.

Figs. 4 and *5* illustrate the dimensionless velocity contours for square and triangular fins at different Rayleigh numbers. As the Rayleigh number increases, buoyancy forces become dominant, significantly intensifying the flow field and leading to a pronounced increase in velocity magnitude for both fin configurations. At low Rayleigh numbers, the flow remains stable and well-organized, whereas higher Rayleigh numbers induce stronger circulation and localized high-velocity regions near the heated wall and fin surfaces.

A direct comparison reveals that triangular fins consistently generate higher local velocities than square fins due to their streamlined geometry, which reduces flow resistance and enhances buoyancy-driven motion, particularly at elevated Rayleigh numbers.

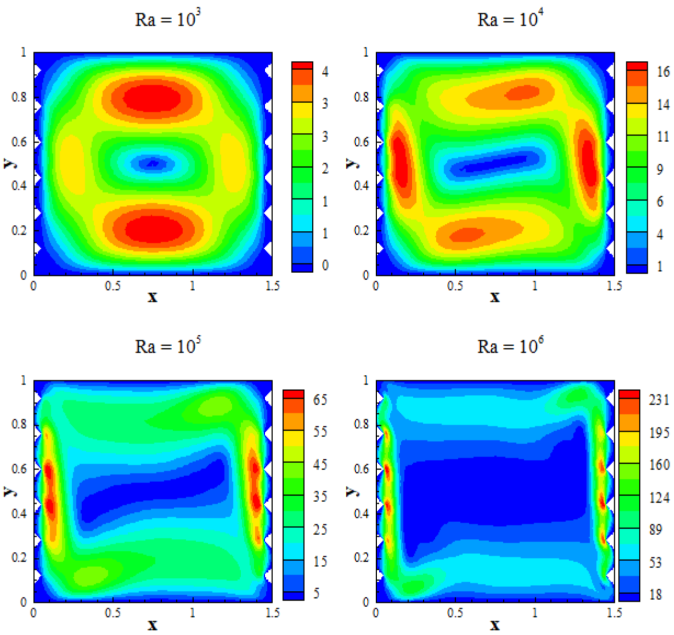


Fig. 4. Velocity contour with triangle fins at different values of the Rayleigh number.

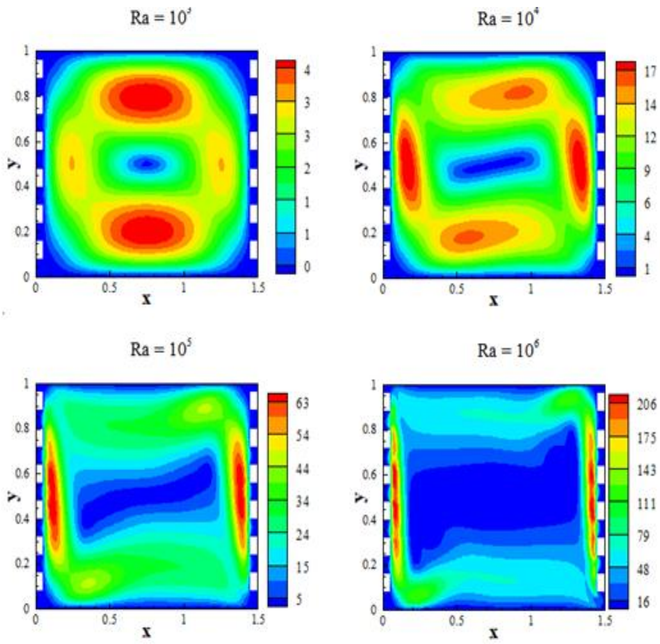


Fig. 5. Velocity contour with square fins at different values of the Rayleigh number.

The influence of Rayleigh number and fin geometry on the thermal field is demonstrated in *Figs. 6* and *7*, which present the temperature contours for square and triangular fins, respectively. Although the dimensionless temperature limits remain unchanged with increasing Rayleigh number, the structure of the thermal field is strongly affected. At low Rayleigh numbers, the heat transfer process is conduction-dominated, resulting in nearly parallel and uniformly spaced isotherms. As the Rayleigh number increases, convective transport becomes dominant, producing increasingly distorted isotherms. For identical Rayleigh numbers, triangular fins yield a more homogeneous temperature distribution inside the enclosure compared to square fins, indicating more effective thermal mixing.

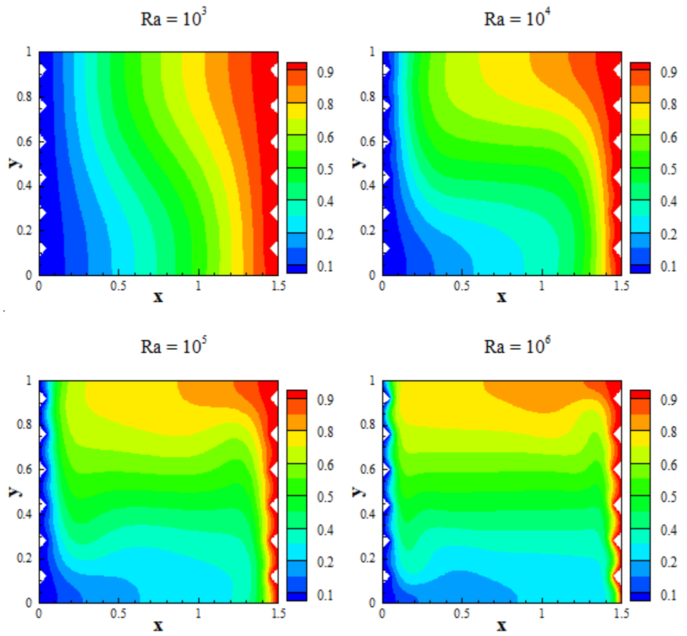


Fig. 6. Temperature contour with triangle fins at different values of the Rayleigh number and fin length t=0.05.

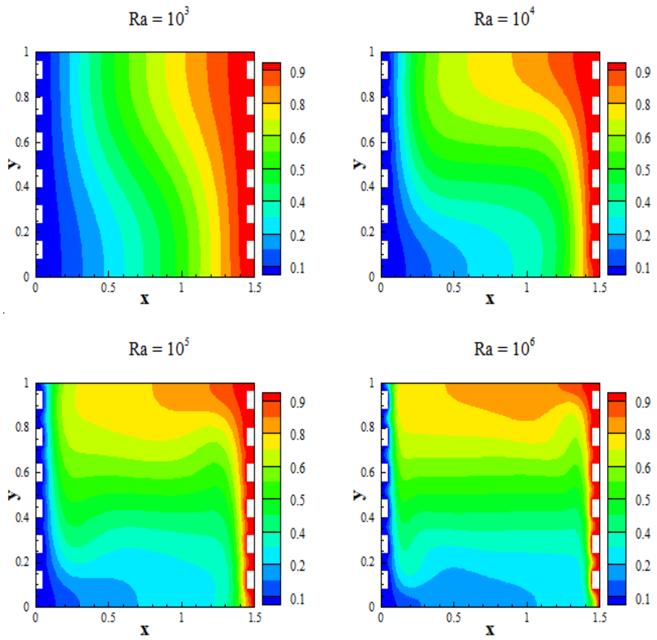


Fig. 7. Temperature contour with square fins at different values of the Rayleigh number.

The interaction between the Rayleigh number and fin shapes can also be observed by examining the rate at which the local Nusselt number changes along the wall. This can be observed in *Figs. 8* and *9*. It can be observed that for both fin shapes, an increase in the Rayleigh number results in an increase in the local Nusselt number. This indicates an increase in convective activity.

However, it can also be observed that the local Nusselt number decreases as we move downstream along the streamwise direction. This indicates that despite the presence of fins, which can have different shapes and sizes, the hydrodynamic constraint always results in a decrease in the temperature difference. This can also be observed with the help of the fin shapes. It can be observed that, despite the shape and size of the fin, it starts disturbing the thermal boundary layer as we move downstream.

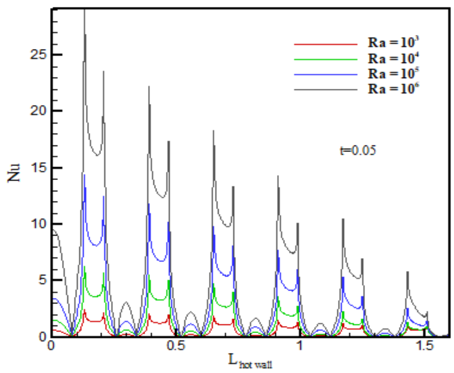


Fig. 8. Local Nusselt number with square fins at different values of the Rayleigh number.

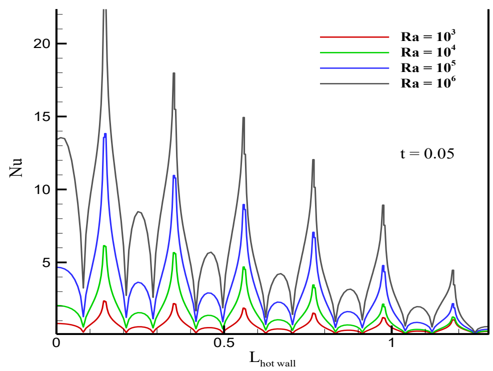


Fig. 9. Local Nusselt number with triangle fins at different values of the Rayleigh number.

Lastly, *Figs. 10* and *11* present how the average Nusselt number changes with the Rayleigh number for different fin lengths. From the figures, it is evident that as the Rayleigh number increases, the average Nusselt number also increases. It is also evident that as the fin length increases, the heat transfer increases at lower Rayleigh numbers.

However, as the Rayleigh number increases, longer fins hinder the flow and reduce natural convection. This results in a decrease in the average Nusselt number. In summary, triangular fins perform better at shorter lengths and at higher Rayleigh numbers compared to square fins. Square fins have a more stable performance as the fin length changes.

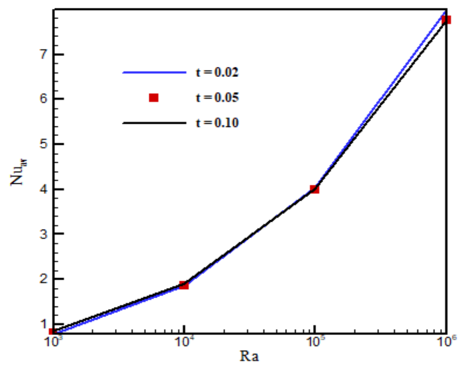


Fig. 10. Average Nusselt number with square fins at different values of the Rayleigh number.

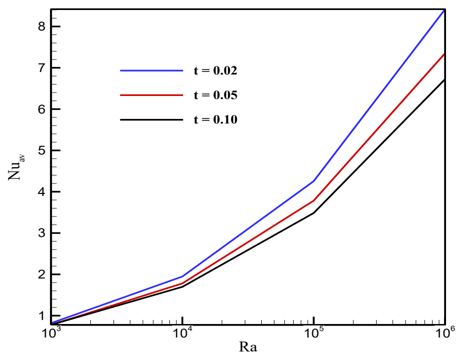


Fig. 11. Average Nusselt number with triangle fins at different values of the Rayleigh number.

5 | Conclusion

The present investigation utilizes numerical methods to examine the phenomenon of natural convection within a rotated rectangular enclosure that contains internal fins, particularly focusing on the impact of fin configuration and the Rayleigh number on the heat transfer characteristics. The results of the investigation indicate that increasing the Rayleigh number significantly increases the buoyancy flow, leading to a significant improvement in the overall heat transfer characteristics.

In addition, the investigation indicates that the configuration of the fins significantly affects the fluid flow and temperature characteristics. In particular, triangular fins are more effective compared to square fins in creating strong buoyancy flow and ensuring temperature uniformity, leading to better heat transfer characteristics. This effect is most pronounced in terms of the higher average Nusselt numbers at higher Rayleigh numbers. The investigation also indicates that there exists an optimal fin length beyond which the fins act in a detrimental manner compared to their absence, particularly at higher Rayleigh numbers.

Conflict of Interest Disclosure

The authors declare they have no competing interests as defined by the journal, or other interests that might be perceived to influence the results reported in this paper.

Data Access

Anonymized data can be requested from the corresponding author following journal data sharing policies.

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