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Research Article

Computational investigations of perforated annular fins under natural convection heat transfer

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ABSTRACT

The cylindrical shape of objects has always been preferred by thermal engineers as a means of transferring heat, owing to their compact design and high surface-to-volume ratio. Additionally, fins were commonly used over the surface of a cylinder to enhance heat transfer even further by providing a larger surface area for heat dissipation. Therefore, this work investigates the effect of perforated annular fins, protruded from a hot vertical cylinder, on heat transfer performance due to free convection. This conjugate heat transfer study was performed for three Rayleigh numbers ($Ra = 0.68 \times 10^7$, 1.37×10^7, and 1.8×10^7) in a laminar flow regime. The study involved varying numbers of fins (ranging from 3 to 7) along the cylinder height, resulting in different fin pitch to cylinder diameter ratios (S/d = 2, 2.4, 3, 4 and 5.8). The fluid flow and energy equations were solved using ANSYS Fluent to understand the correspondence between heat transfer and flow behavior. The grid dependence test and the data validation with published work were successfully performed. The Nusselt number is observed to increase with the increase in Rayleigh number, up to 41.75%, irrespective of the number and type of fins used. The heat transfer enhancements due to perforated fins are found to be higher than those without perforations. The maximum augmentation in Nusselt number is found to be 49% in case of the highest number of perforated fins as compared to solid fins, corresponding to an S/d ratio of 2 and a Ra of 1.8×10^7. Conversely, the minimum enhancement in augmented Nusselt number is 20% for an S/d ratio of 5.84 and Ra of 0.68×10^7. The flow characteristics are found to be in commensurate with the heat transfer results.

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INTRODUCTION

Natural convection occurs due to the movement of fluid over a hot or cold surface, caused by density differences between the near-wall fluid and the outer fluid resulting from temperature variations. In contrast, forced convection occurs due to movement of fluid caused by pressure difference, created by external means such as fans, blowers, or pumps. The heat transfer rate in natural convection is generally lower than forced convection due to slower bulk fluid motion and a correspondingly lower convection coefficient.

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To enhance natural convection heat transfer, one can increase the dissipating surface area by attaching fins (extended surfaces) to the heat transfer surface. These fins are typically made from materials with high thermal conductivity, such as copper and aluminum.

For designing efficient and reliable thermal systems, it is of immense importance to understand the correlation between flow phenomenon and their impact on heat transfer characteristics. Natural convection finds a wide range of applications, including cooling engine cylinder block, high-voltage electrical transformers, hot wires, and electronic chips. In these cases, faster heat dissipation is essential for safe and reliable operation and maintenance.

Numerous investigations have been conducted on natural convection with different types of fins utilizing various schemes and methods. While studying free convection over a horizontally suspended cylinder, Churchill and Chu [1] explored the influence of the Rayleigh number (*Ra*) and the Prandtl number (*Pr*) on the Nusselt number (*Nu*). Subsequently, correlations were developed for *Nu* under both laminar and turbulent flow, claiming improved accurately compared to previously established correlations. Bilgen [2] numerically investigated free convection in a cubic space with thermally insulated top and bottom walls, along with differentially heated side walls. The study included a single straight fin protruding from the hot vertical wall. It was reported that *Nu* is directly proportional to *Ra* and inversely proportional to fin length as well as the dimensionless conductivity ratio (k_{fin}/k_{air}).

Varol et al. [3] numerically studied the buoyancy-driven flow across a porous medium in a cavity. The flow was induced by a sinusoidal heated bottom wall, resulting in multiple recirculation zones, irrespective of all the problem variables. The heat transfer rate was found to increase with the amplitude of the sinusoidal function, decrease with aspect ratio of the rectangular enclosure, and increase with *Ra*.

Bocu and Altac [4] carried out a numerical investigation on cubic cavity embedded with an array of pin fins. One wall was heated to a specified temperature, while the opposite wall was cooled. The remaining walls were made adiabatic. An array of isothermal pin fins was fixed on the heated base in an inline and staggered manner to optimize the heat transfer. By varying different geometrical parameters, flow patterns and heat distribution were observed. They found that *Nu* increases with increasing length and number of fins irrespective of how fins are arranged. The horizontal inline arrangement provided around 4-7% higher heat transfer than that due to vertical inline arrangement. However, the staggered arrangement was reported to be the most efficient configuration.

Senapati et al. [5] conducted a numerical study on heat transfer from vertical cylinders embedded with annular fins in quiescent air with varying *Ra*. The effect of varying fin to tube diameter ratio (D/d) and fin gap to tube diameter ratio (S/d) on *Nu* were analyzed. The highest heat transfer augmentation for turbulent flow was obtained with an optimal S/d ratio between 0.28 and 0.31. Additionally, correlations were developed for average *Nu* in terms of S/d, D/d, and *Ra*. Pathak et al. [6] performed numerical simulation of free convection from an array of vertical fins with variable height. The authors observed improved performance due to an array of fins of variable height, for certain conditions, in comparison with that of fixed height.

Dash and Dash [7] conducted a 3-D computational study on free convection from a thick hollow cylinder kept horizontally at *Ra* starting from 10^4 up to 10^8. They calculated the Nusselt number by varying thickness ratio (L/D) and diameter ratio (d/D). They observed the thermal plume, velocity vector, and temperature contours at various sections in different planes. Their findings indicated that reducing the thickness ratio (d/D) led to increased Nusselt number. They proposed a useful correlation suitable for many industrial applications, noting that as *Ra* increases, heat transfer from the inner surface increases while that from the outer surface decreases.

Several studies (Abu-Hijleh [8], AlEssa and Al-Widyan [9], Huang et al. [10], Awasarmol and Pise [11], Sobamowo et al. [12], Sunder et al. [13], Kiwan et al. [14]) have reported the influence of fin shape alterations, by incorporating different forms of perforations and cavities in the fin, on heat transfer intensification. Through numerical study, Abu-Hijleh [8] reported that permeable fins provide significantly faster heat transfer than solid ones. Al-Essa and Al-Widyan [9] analysed the effect of triangular perforations inside a horizontal rectangular fin on the enhancement of heat dissipation under free convection in comparison with its solid counterpart. The extent of enhancement was found to be proportionate with the thickness as well as the thermal resistivity of the fin material used. Moreover, the introduction of perforations resulted in the reduction of material costs.

Huang et al. [10] employed a perforated fin base to study the enhancement of free convection due to an array of rectangular fins and observed substantial improvement in ventilation resulting in higher heat transfer enhancement. Smaller perforations were reported to provide better enhancement in heat transfer. Free convection from an array of perforated rectangular fins with varying inclinations was experimentally investigated by Awasarmol and Pise [11]. They found better augmentation in heat dissipation with 12mm diameter perforated fins oriented at an angle of 45°. Sobamowo et al. [12] numerically analyzed natural convection heat dissipation from porous fins assuming internal heat generation along with variable thermal conductivity. They found that with the increase in the porosity, fin thickness-length ratio, *Nu*, Darcy number, and *Ra*, heat dissipation enhances, reaches an optimum value, and then becomes almost constant afterward.

Sunder et al. [13] had performed experimental as well as numerical investigations of the heat sink (cylindrical) meant for LED bulb cooling, wherein short perforated fins were radially mounted all over the hot outer surface. A numerical model was developed to explore the effectiveness of finning and porosity factors, along with the angle of orientation, on the heat dissipation and the fin mass required. It was observed that the perforated staggered fin configuration caused around a 7 to 12% reduction in thermal resistance (depending on orientation angle) and around a 9% reduction in mass when compared with non-perforated ones. The effect of porous annular fins protruding from the heated vertically cylinder was experimentally investigated by Kiwan et al. [14]. A highly permeable fin exhibited a higher heat dissipation rate. The authors reported a minimum enhancement of 7.9% in heat transfer when using a less permeable fin of 10 mm thickness, whereas the maximum enhancement was around 131% when the entire cylinder was covered with a highly permeable layer.

Through experimental and numerical investigations, Ding et al. [15] had found that three-dimensional fin protrusions on a heated tube provided up to 207% higher Nusselt number than smooth tube. The authors observed that the variation in fin height affects the Nusselt number the most. Krishnayatra et al. [16] performed a detailed numerical study of free convection from straight longitudinal fins attached to a horizontally placed circular cylinder under laminar flow (Ra£10^6). They investigated efficacy of the fins in terms of heat transfer and effectiveness for varying length, thickness, number and material of the fins. For a fixed length of the fin, there exist an optimum number of fins and vice versa. The highest effectiveness was found to be 4.34 for 12 fins. Pradhan et al. [17] investigated the cumulative effect of free convection as well as radiation from a vertically placed cylinder with straight circumferential fin protrusions. The authors found that the total as well as radiation heat dissipation with fin is more prominent in case of laminar flow than turbulent flow.

Jalili and colleagues exploited the potential of computational analysis based on the finite volume method (FVM) to investigate heat transfer in various applications. These applications include geothermal systems (Jalili et al. [18]), countercurrent double-tube heat exchangers (Jalili et al. [19]), shell-and-tube heat exchangers (Jalili et al. [20]), two-phase cross flow (Jalili and Jalili [21]), and boiler economizers with spiral geometry (Salehipour et al. [22]). The heat exchangers equipped with both rectangular and curved fins exhibited 81% and 85% greater efficiency compared to finless heat exchangers (Jalili et al. [19]).

Mohamad and colleagues utilized ANSYS Fluent to investigate several aspects: (i) entropy production and cooling time of a blast furnace, considering variations in its cross-sectional area (Mohamad et al. [23]), (ii) conjugate natural convection in a cylindrical open cavity with isothermal boundary conditions at the inner wall (Mohamad et al. [24]), (iii) conjugate natural convection from a vertical cylindrical open cavity made of aluminum with varying thicknesses (Mohamad et al. [25]) and (iv) natural convection thermal dissipation in an upright cylinder with a closed lower base (Pulagam et al. [26]). It has been observed that tripling the number of fins led to a maximum 50% increase in the heat flow rate (Pulagam et al. [26]).

The above-cited research highlights the significant impact of fin design on natural convection heat transfer across various applications. However, to the best of the authors' knowledge, a detailed investigation comparing the heat transfer performance of perforated annular fins at different spacings on a vertical cylinder under free convection conditions remains unexplored. Therefore, this study employs computational analysis on a vertical cylindrical geometry with perforated fins. By solving fundamental fluid dynamics equations in Ansys-Fluent, the present numerical study intends to quantify the natural heat dissipation from a vertical cylinder embedded with varying numbers of perforated annular fins at different Rayleigh numbers. This conjugate heat transfer problem has been simulated for five inter fin distance to fin diameter S/d ratios (2, 2.4, 3, 4, and 5.8) for three Rayleigh numbers $0.68{\times}10^7$, $1.37{\times}10^7$, and $1.8{\times}10^7$. Additionally, this study correlates the velocity vector field and temperature contours to gain insights into Nusselt number variations.

METHODOLOGY

The vertical cylinder integrated with perforated annular fins is shown in Figure 1. The study intends to numerically examine the effect of different configurations of perforated annular fins on the heat dissipation and the underlying flow phenomenon at varying *Ra*.

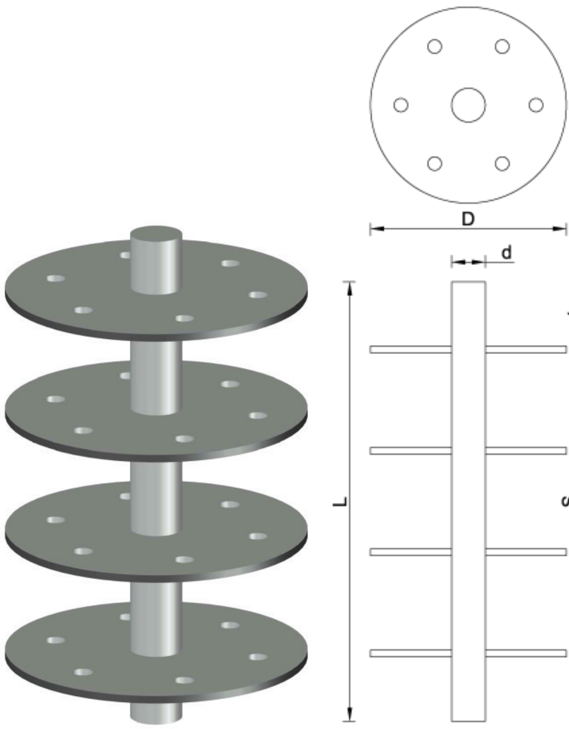

Figure 1. Perforated annular fins integrated with vertical cylinder.

The solutions of three-dimensional governing Equations are obtained using ANSYS Fluent 18.0. The vertical cylinder is 325 mm length *L* and 25 mm diameter d, and is embedded with annular fins of 1mm thickness and 145mm diameter. The annular fins contain 6 circular perforations (10 mm diameter) at a distance of 10 cm from the cylinder axis. The Material selected for the cylinder and the fins is aluminum.

Boussinesq approximation is used to simplify the governing equations derived from fundamental principles and are used to implement SIMPLE algorithm. The major assumptions made in this work are steady-state heat transfer and fluid flow, uniform wall temperature, incompressible fluid, laminar flow, constant thermos-physical properties of solid and fluid, negligible radiation heat transfer.

Continuity Equation,

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

X-momentum equation,

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

Y-momentum equation,

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\nu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)+g\beta(T-T_\infty) \tag{3}$$

Z-momentum equation,

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial z}+\nu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \tag{4}$$

Energy equation,

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \tag{5}$$

The computational domain and the boundary conditions are shown in Figure 2. Thermal boundary conditions of constant wall temperature 350K were applied to the outer surface of the solid cylinder. The pressure outlet boundary conditions were implemented to all the boundaries of the flow domain. The two ends of the solid cylinder are taken as adiabatic. The density of air is taken as 1.225 kg/m^3 with an operating temperature of 300K. The gravity is taken into consideration along the negative y-axis. Eq. 6 is used to estimate the Rayleigh number (*Ra*).

$$Ra=\frac{g\beta(T_w-T_\infty)L^3}{\nu\alpha} \tag{6}$$

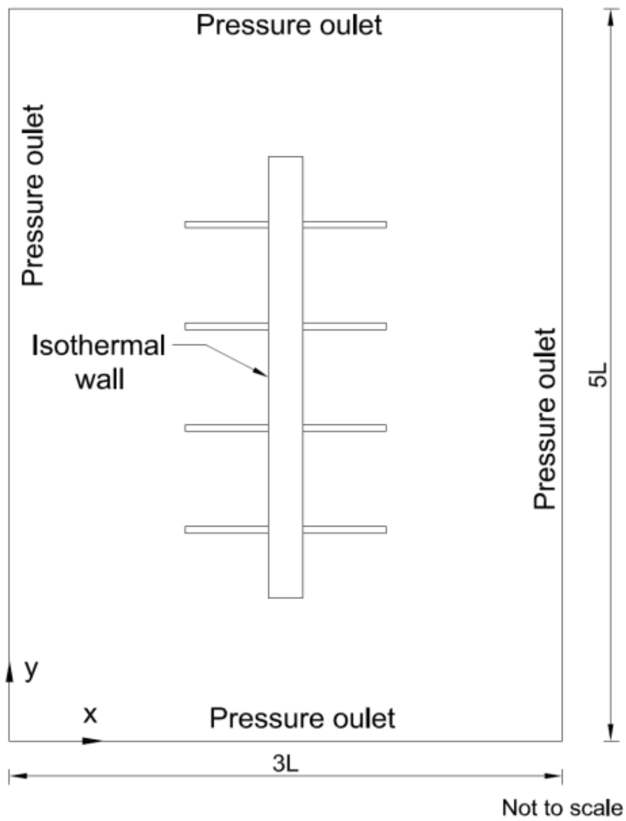

Figure 2. Computational domain and boundary conditions.

Where T_w, T_∞ are temperatures of wall surface, surrounding air respectively, and ν is the kinematic viscosity of air. Rayleigh number is the function of wall surface temperature. Thus, three different wall temperatures were applied resulting in three Rayleigh numbers.

Newton's law of cooling is employed to determine the heat transfer dissipation (*Q*).

$$Q=hA(T_w-T_\infty) \tag{7}$$

Where *A* is the total area exposed to convective heat transfer. For an N_{fin} number of fins, the exposed area can be calculated by Eq. (8)

$$A=N_{fin}A_{fin}+A_b \tag{8}$$

A_{fin} for perforated and solid fins are calculated by Eq. 9 and Eq. 10.

$$\left(A_{fin}\right)_{\text{Perforated}}=2\times\frac{\pi}{4}D^2-6\times\frac{\pi}{4}d'^2-2\times\frac{\pi}{4}d^2 \tag{9}$$

$$\left(A_{fin}\right)_{\text{Solid}}=2\times\frac{\pi}{4}D^2-2\times\frac{\pi}{4}d^2 \tag{10}$$

A_{fin} and A_b are fin surface area and base area not occupied by the fin respectively. The *Nu* is calculated by Eq. (11)

$$Nu = \frac{hL}{k} = \frac{QL}{A(T_w - T_\infty)k} \tag{11}$$

Grid Independence Test and Data Validation

Figure 3 shows the cell arrangement in a cylinder along with the fin and the adjacent air. A grid independence test is performed by changing the number of grid elements starting from 1 million up to 2 million for a typical case in the present work. It was found that variation in Nusselt number is almost negligible after 1.8 million elements as shown in Figure 4; therefore, an almost similar number of elements were considered for all the simulations.

The present numerical model is validated by comparing the present outcome with that of Senapati et al. [5] as shown in Figures 5 and 6. The maximum deviation in the Q and the Nu is found to be less than 8.3% which can be considered reasonably accurate.

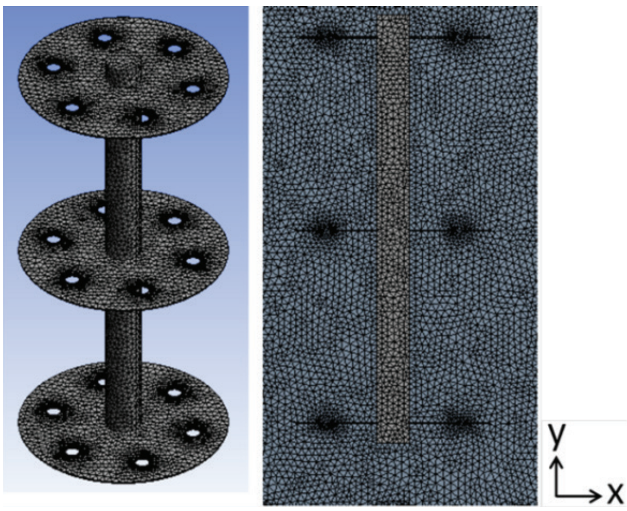

Figure 3. Typical mesh generated for the cylinder with perforated annular fins.

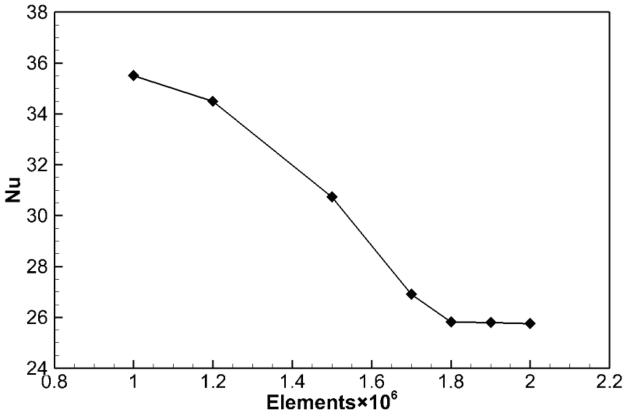

Figure 4. Variation of Nusselt number with number of elements for S/d= 2.4 and Ra = 1.37×10^7.

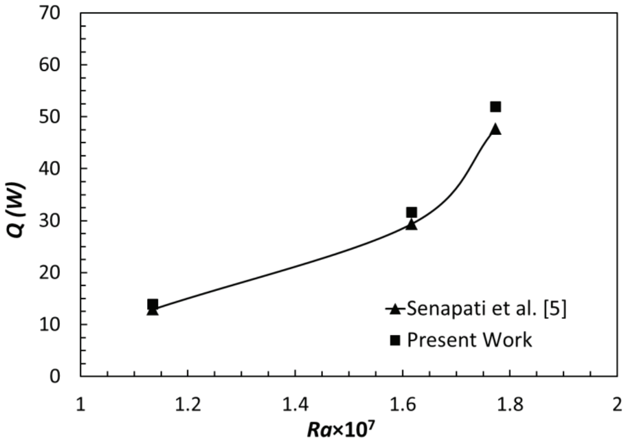

Figure 5. Heat transfer vs. Rayleigh number with S/d=6, D/d=5 for solid annular fins.

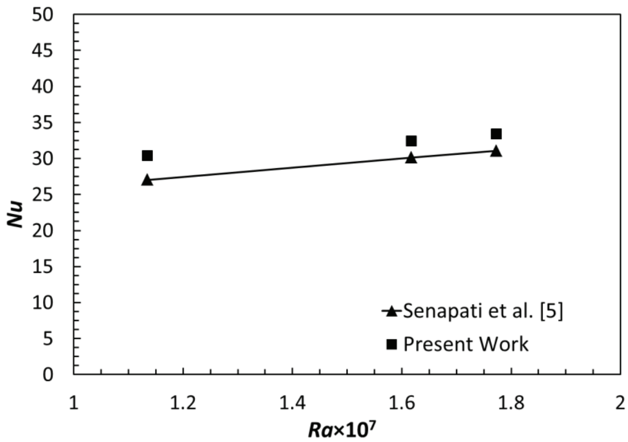

Figure 6. Nusselt number vs. Rayleigh number with S/d=6, D/d=5 for solid annular fins.

RESULTS AND DISCUSSION

In this work, a numerical solution of 3-dimensional natural convection due to perforated annular fins is obtained and presented in terms of qualitative visualization of temperature plume, velocity vectors around the investigated geometry. The influence of different factors such as fin pitch to cylinder diameter ratio (S/d ratio) and Ra on Q, average Nu, augmented Nusselt number (Nu/Nu_o) were also presented quantitatively.

The influence of Ra and S/d on the Q is shown in Figure 7. Three Ra values of 0.68×10^7, 1.37×10^7, and 1.80×10^7, are considered. It is evident from the figure that Q increases as the number of perforated fins increases (with decreasing S/d ratio) for any Rayleigh number. The increase in Q with the increase in the number of fins can be attributed to the increase in the area exposed to heat transfer. However, Q increases with increase in Ra for any S/d ratio (or number of perforated fins) which may be attributed to higher fluid

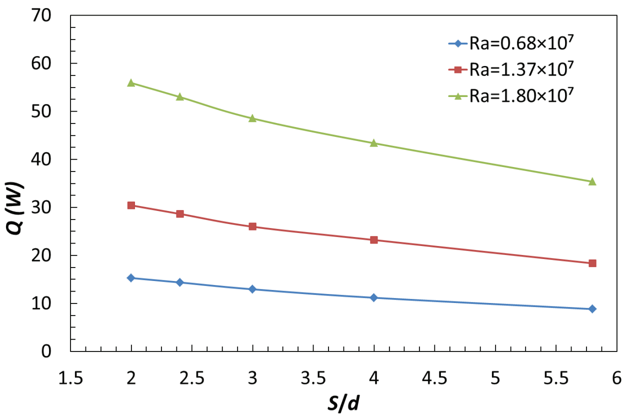

Figure 7. Effect of *S*/*d* ratio on heat transfer rate for different Rayleigh numbers.

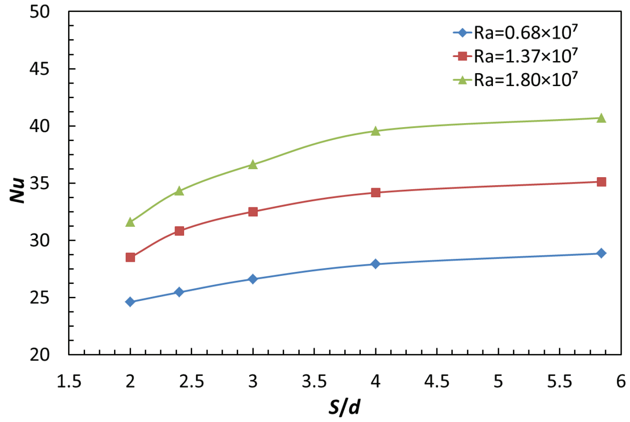

Figure 8. Effect of *S*/*d* ratio on Nusselt number for different Rayleigh numbers.

velocity and mixing around the fin perforations at higher *Ra*.

Conversely, *Nu* decreases when the number of perforated fins increases (with decreasing *S*/*d* ratio) as shown in Figure 8. The *Nu* is directly dependent on *Q* and inversely dependent on the entire dissipation area as evident from Eq. (11). The heat transfer rate increases with the decrease in the *S*/*d* ratio or increase in the number of perforated fins. Perforated fins modify the flow field around the fin and impact heat transfer. When fins are perforated, the effective surface area for heat transfer increases due to the additional surface provided by the holes. However, the flow resistance becomes more pronounced because of the holes. This effect is evident from the low velocity contours observed in the inter-fin regions (Fig. 9 and Fig. 10), resulting in a decrease in the Nusselt number. Therefore, as an overall effect, the

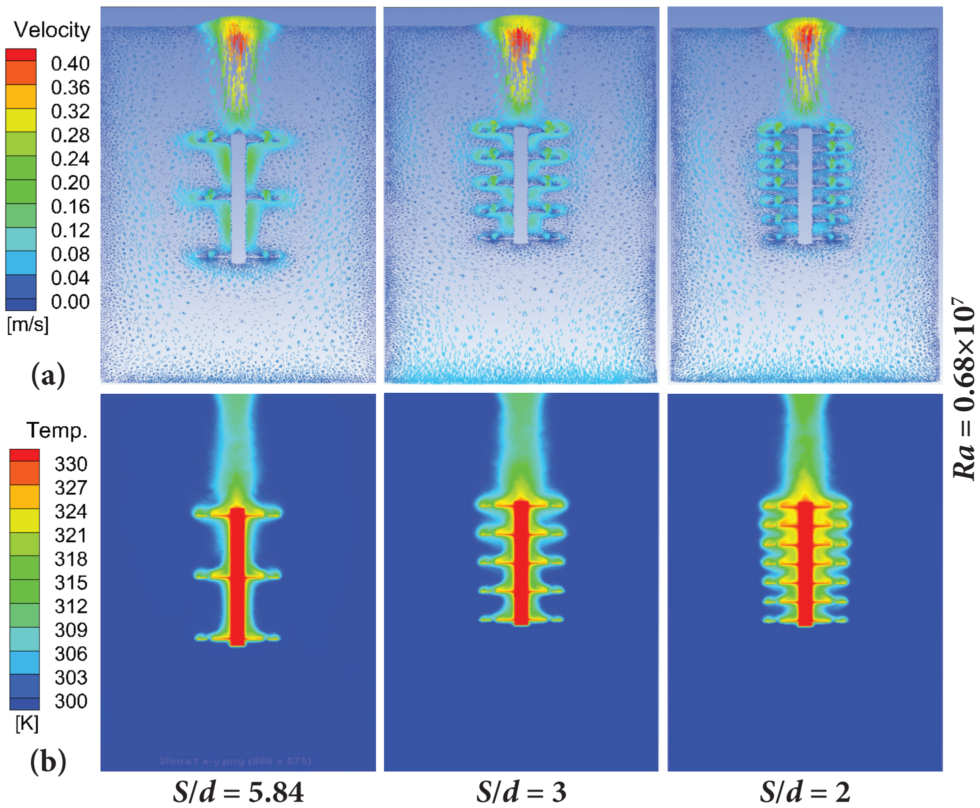

Figure 9. (a) Velocity vector field and (b) Temperature contours at the central x-y plane for *S*/*d* ratio 5.84, 3, and 2 corresponding to Ra=0.68×10^7.

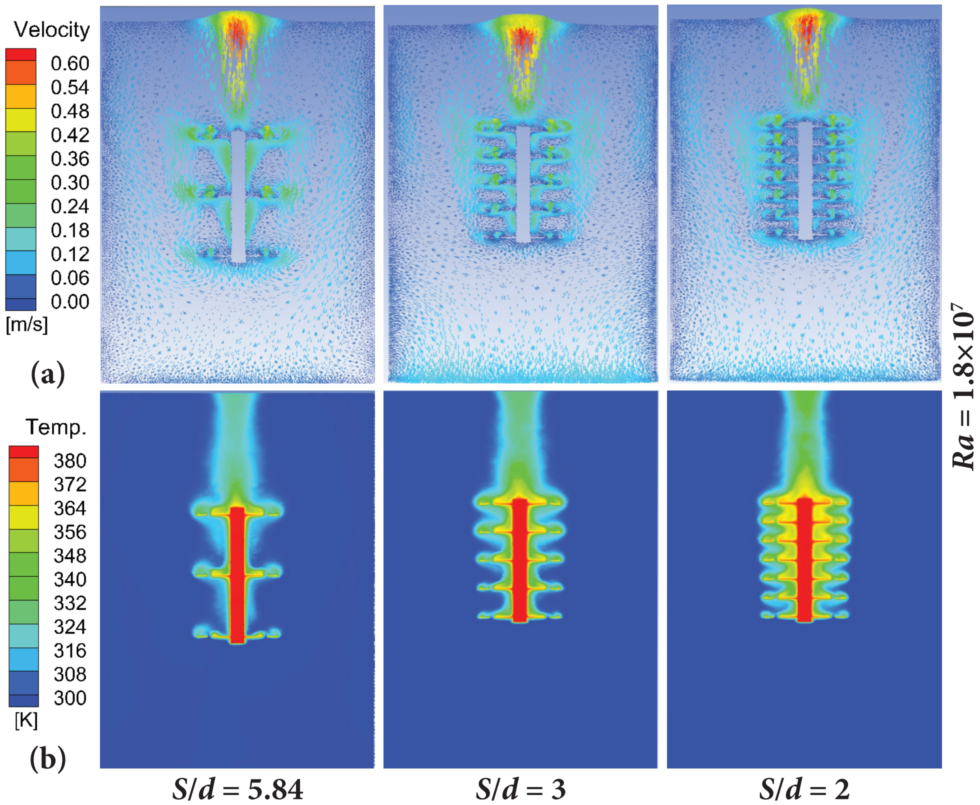

Figure 10. (a) Velocity vector field and (b) Temperature contours at the central x-y plane for S/d ratio 5.84, 3, and 2 corresponding to $Ra = 1.8 \times 10^7$.

Nu decreases with decreasing the S/d ratio or increasing the number of perforated fins (Fig. 8). However, Nu always increases with the increasing Rayleigh number for all S/d ratios (or the number of perforated fins). This phenomenon occurs because at high Rayleigh numbers, natural convection becomes dominant due to increased air velocity and perturbations around the perforated fins. This intensification in the convective currents leads to a higher Nu.

The augmented Nusselt number (Nu/Nu_o) is also calculated as a function of S/d ratio and Rayleigh number and shown in Figure 11 and Figure 12, where Nu denotes the Nusselt number in the case of perforated annular fins and Nu_o denotes the Nusselt number without perforations.

As the S/d ratio increases, the augmented Nusselt number decreases for any of the Rayleigh numbers considered in this study (Fig. 11). Nu/Nu_o is more than one for any

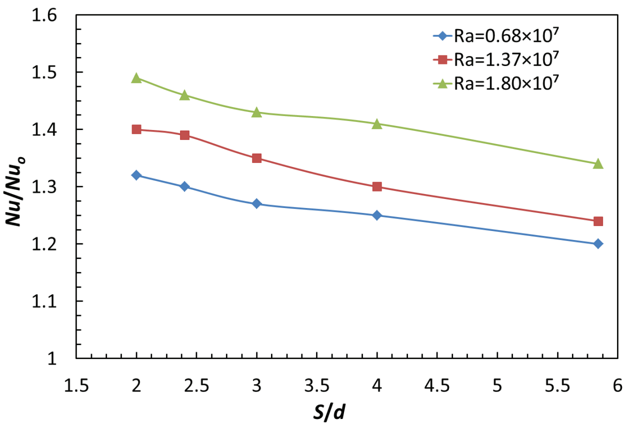

Figure 11. Effect of S/d ratio on augmented Nu for different Ra.

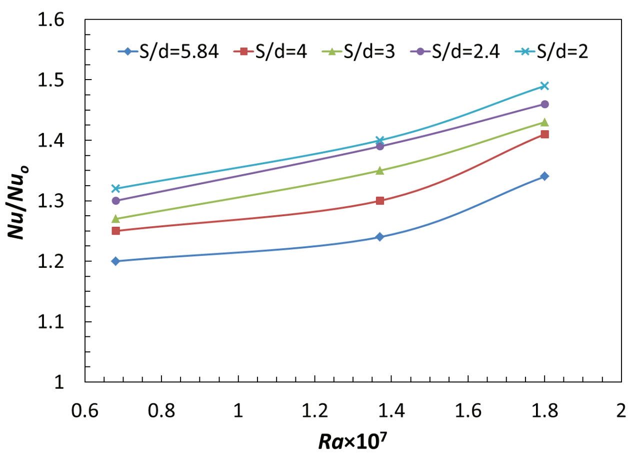

Figure 12. Effect of Ra on augmented Nu for different S/d ratio.

combination of S/d and Ra. Consequently, the introduction of perforations within the annular fins consistently enhances heat transfer, irrespective of the S/d ratio and Rayleigh number. Perforated annular fins outperform solid fins by offering an increased surface area, promoting turbulence through fluid coming out from the holes, which leads to improved mixing and convective exchange. Additionally, perforated fins attain a balance between surface enhancement and flow resistance. These factors collectively contribute to the augmented Nusselt number (Nu/Nuo) observed in perforated fins. The maximum enhancement in Nusselt number is 49% for an S/d ratio of 2 and Ra of 1.8×10^7 and minimum enhancement in Nusselt number is 20% for an S/d ratio of 5.84 and Ra of 0.68×10^7 (Fig. 11 and 12).

Temperature contours and Velocity vector field at the central *x-y* plane for S/d ratio 5.84, 3, and 2 corresponding to the minimum Ra of 0.68×10^7 and maximum Ra of 1.8×10^7 are shown in Figure 9 and Figure 10 respectively. Figure 9(a) and 10(a) show that the air is passing through perforations resulting in higher velocity and mixing in the inter-fin region which may be seen as a reason for enhancement in heat transfer due to perforated fins as compared to solid fins (Fig. 11 and 12). This phenomenon is more prominent at higher Ra and lower S/d ratio. It can be observed from Fig 9(b) and 10(b) that temperature of the rising plume increases with increasing number of fins (i.e., decreasing S/d ratio) and increasing Ra which verifies enhanced Q and Nu at higher Ra and lower S/d ratio shown in Figure 7 and 8.

Overall, the impact of varying the fin pitch-to-diameter ratio (S/d) on heat transfer augmentation is indeed significant because it affects the flow behavior around the fins. Closely spaced fins increase total surface area and enhance heat transfer, but also raise flow resistance. In contrast, widely spaced fins reduce flow resistance, but limit convective heat transfer and decrease the Nusselt number. Therefore, designers must have a trade-off between heat transfer enhancement and flow resistance. For designing the systems for cooling applications (e.g., electronics cooling) that require efficient heat dissipation, a smaller S/d may be preferred. Conversely, for low-pressure drop systems (e.g., HVAC systems), a larger (S/d) is advantageous.

CONCLUSION

This work investigates natural convection due to perforated annular fins, integrated on a hot vertical cylinder. Simulations were performed for various configurations of the fin pitch to cylinder diameter ratio and Rayleigh numbers. The results were presented in terms of heat transfer (Q), Nusselt number (Nu), augmented Nusselt number (Nu/Nu_o) quantitatively. Two-dimensional temperature contours and velocity vectors were also presented for visualization purpose. The outcomes are listed as follows.

1. The value of Q increases as the number of perforated annular fins attached to the hot vertical cylinder increases. Specifically, heat transfer is maximum at S/d = 2 (with 7 fins) and minimum at and S/d = 5.84 (with 3 fins). However, regardless of the S/d ratio, Q increases as the Ra increases.
2. The highest Q value, corresponding to $Ra = 1.8 \times 10^7$ and $S/d = 2$ is 53.3% higher than the lowest value, corresponding to Ra = 0.68×10^7 and $S/d = 5.84$.
3. Nusselt number increases as Ra increases, irrespective of the S/d ratio (or the number of fins). But, Nusselt number decreases as the number of fins increases (or as S/d ratio decreases) for any of the Ra.
4. The highest Nu value, corresponding to $Ra = 1.8 \times 10^7$ and $S/d = 5.84$, is 65.4% higher than the lowest value, corresponding to Ra = 0.68×10^7 and $S/d = 2$.
5. Augmented Nusselt number increases with increasing *Ra,* irrespective of the S/d ratio (or the number of fins). However, it decreases with an increase in S/d ratio.
6. The maximum augmentation in Nu is observed in the case of perforated fins compared to solid fins. Specifically, this augmentation is found to be 49%, corresponding to an $S/d = 2$ and $Ra = 1.8 \times 10^7$. Conversely, the minimum enhancement in augmented Nu is 20% for an $S/d = 5.84$ and $Ra = 0.68 \times 10^7$.

Present outcomes may be useful for various industrial applications, including cooling of machinery and equipment, where it contributes to effective heat dissipation in a cost-effective and environmentally friendly manner. This work may be further extended to investigate the effect of varying perforation size, annular fin diameter/thickness on enhancement of natural convection heat transfer.

NOMENCLATURE

A	Total area exposed to convective heat transfer [m^2]
A_b	Base area [m^2]
A_{fin}	Fin surface area and base area [m^2]
D	Fin diameter [m]
d	Cylinder diameter [m]
d'	Diameter of fin perforation [m]
g	Gravitational acceleration [m/s^2]
h	Convective heat transfer coefficient [W/m^2.K]
k	Thermal conductivity [W/m.K]
k_{air}	Thermal conductivity of air [W/m.K]
k_{fin}	Thermal conductivity of fin material [W/m.K]
L	Cylinder length [m]
N_{fin}	Number of fins
Nu	Nusselt Number for perforated fins
Nu_o	Nusselt number for fins without perforations
p	Pressure [N/m^2]
Pr	Prandtl number
Q	Heat transfer [W]
Ra	Rayleigh number
S	Fin pitch
T	Temperature [°C]
T_w	Wall temperature [°C]
T_∞	Air temperature [°C]
u,v,w	Velocity components [m/s]
x,y,z	Cartesian coordinates

Greek symbols

α Thermal diffusivity [m^2/s]

β Coefficient of volumetric expansion [K^{-1}]

ρ Density [kg/m^3]

ν Kinematic viscosity [m^2/s]

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 authors 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.

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