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RESEARCH PAPER

Free Convection from a One End Closed Vertical Pipe with Annular Fins: A Computational Study

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Abstract

Cylindrical objects in heat transfer applications have always taken the spotlight due to their compact nature and better surface area-to-volume ratio over their cuboid counterparts. Despite this advantage, fins have been used to enhance the heat transfer properties of such geometries. This study involves one such geometry where the top of the cylinder is open to the surrounding atmosphere and the cylinder is fitted with annular fins of different sizes and spacing. The k-ε model is used to simulate the turbulent cases along with a density-based solver. The cylinder is modeled to be a heat source with a constant temperature of 350 K and the surrounding atmosphere being air at 300 K. Nusselt number and effectiveness of the fins are calculated and analyzed. The heat flow rate experienced a maximum increase of 50% when the number of fins is raised from S/d 0.9 to S/d 0.3. The maximum effectiveness occurs at Ra = 1011, with a value of 4.44 observed for S/d of 0.3 and d/D of 0.25. The maximum values for effectiveness are observed at lower d/D and S/d values.

Keywords: Natural convection, heat transfer, open cavity, extended surfaces.;

1. Introduction

Research on natural convection heat transfer in cylindrical geometries has been going on for several decades due to its advantageous properties, namely the absence of noise and moving parts, cost-effectiveness, and economical operation. However, a drawback limits its use in real-life applications, which is its minimum rate of thermal dissipation. Here comes the role of fins to tackle this problem, which explains the extensive research that has been going on in this field of engineering. The literature review studies are classified into four parts to make it more readable. The first part is for the studies dedicated to plain cylinders that have no fins [1-11]. The second section is for the cylinders fitted with fins [12-22]. The third part is for finned cylinders with special geometries [23-26], and the fourth part is for special geometries with fins [27-31].

Rana [1] explored the influence of the inclination angle on the Nusselt number on the inner and outer surfaces of the cylinder submerged in motionless air for the Rayleigh number in the laminar regime. The study showed the Nusselt number reached its peak value when the cylinder was in a level position and then it declined steadily as the inclination angle increased for given values of Ra and *L/D*. Dash and Dash [9] executed a numerical study to

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determine the natural convection thermal dissipation in cylindrical geometry for two situations: the first one the cylinder was grounded and the second was elevated from the ground. The study showed that the thermal dissipation got better when the cylinder was elevated, however; the longer the cylinder gets, the less the thermal dissipation becomes. The study suggested a formula to predict thermal dissipation based on the above-mentioned factors.

2. Nomenclature

A_t	Surface area of the hollow cylinder (m^2)
C_f	Skin friction coefficient
D	Maximum diameter of the cylinder (m)
Gr	Grashof number
g	Gravitational acceleration (m/s^2)
h	Heat transfer coefficient ($W/m^2.K$)
K	Thermal conductivity (W/m.K)
L	Cylinder length (m)
L/D	Dimensionless ratio of the cylinder length to the maximum diameter
Nu	Nusselt number
P	Pressure (N/m^2)
P_{atm}	Atmospheric pressure (N/m^2)
Pr	Prandtl number
Q	Total heat transfer rate (W)
Ra	Rayleigh number
T_w	Surface temperature of the cylinder (K)
T_∞	Ambient temperature (K) = 300 K
u	velocity

Greek Symbols

α	Thermal diffusivity (m^2/s)
β	Volume Coefficient of thermal expansion (1/K)
ϑ	Kinematic viscosity (m^2/s)
μ	Dynamic Viscosity (Pa-s)
ρ	Density of air (kg/m^3)

Shah and Rana [11] performed a computational analysis to analyze the performance of natural convection thermal dissipation for two cylinders aligned in an upright direction. The outcome of this study shows that the larger the space between the two cylinders, the higher the rate of thermal dissipation. Rana and Senapati [4] analyzed the natural convection thermal dissipation in a tube standing upright in a still liquid. The study shows that the rate of thermal dissipation depends on the viscosity of the liquid. Thus, the less the viscosity is, the higher the thermal dissipation gets. Furthermore, the inner surface of the tube loses heat faster than the outer surface of longer tubes. Mukherjee et al [5] investigated the best way to cool the exhaust gas from ships and planes to make them less noticeable to heat-seeking missiles. The study shows that the louvers in the funnel system play a vital role in cooling the exhaust gas by sucking cold air to the hot gas stream, thus it makes it harder to detect by infrared sensors.

Mohamad et al [2, 3, 6] conducted several computational studies to estimate the effect of geometrical ratio, temperature differences, and thickness on the rates of entropy formation and natural convection thermal dissipation in cylindrical objects in the laminar regime. The studies show how geometrical ratio affects the performance of thermal dissipation heavily due to the change of surface area. The studies suggested two formulas to calculate the Nusselt number and predict the cooling time of the cylinder. Rana et al [7] explored the influence of Reynolds number, geometrical ratio, and Rayleigh number on the rate of thermal dissipation in an upright tube in two situations: motionless and rotating tube. The outcomes of this study are that the spinning tube has a higher rate of thermal dissipation compared to a still tube due to the higher value of Reynolds number. In addition, a smaller geometrical ratio and higher Rayleigh number both induce a higher rate of natural convection thermal dissipation. Dash and Dash [8] evaluated the effect of geometrical ratio and wall thickness ratio on the rate of natural convection thermal dissipation in an upright tube hanging in the air. The study shows that a taller tube with a low thickness ratio has better thermal dissipation. Liu et al [10] carried out a numerical study to analyze natural convection thermal dissipation in two tubes stacked vertically on top of each other. The study shows that the more distance between the tubes, the better the thermal dissipation becomes.

Li and Byon [21] performed a computational and empirical study to evaluate the influence of the direction of a circular thermal sink with rectangular fins on the rate of natural convection thermal dissipation. The study shows that horizontal orientation improves the rate of natural convection thermal dissipation. Furthermore, adding more fins and increasing their dimensions affect the thermal dissipation positively. Shen et al [17] researched the natural convection thermal dissipation in a thermal sink consisting of an upright cylinder with lengthwise fins. This study suggests a correlation to predict the performance of the thermal sink based on the design of the fins. Hosseini et al [22] explored the natural convection thermal dissipation in a double-pipe heat exchanger in which the inside pipe is fitted with lengthwise fins and surrounded with phase change material (PCM). The outcomes of this study are that fins have a vital role in improving the efficiency of thermal storage in the PCM in terms of time and thermal penetration. Joo and Kim [20] proposed a new correlation to estimate the rate of thermal dissipation in an upright cylinder with internal fins. This correlation helps in optimizing the fins' dimensions to achieve the best thermal dissipation performance. Kumar et al [19] emphasized the importance of optimizing the fins' dimensions and spacing to achieve the highest possible rate of thermal dissipation.

Senapati et al [16, 18] analyzed thermal dissipation in cylindrical-finned geometries that have level and upright orientations. The two studies suggested empirical equations to help optimize the fin's dimensions. One important finding was that the fins have limited influence in turbulent air in case they are too close due to air blocking. Acharya and Dash [15] proved that long cylinders with level orientation do not have optimized fins' dimensions and spacing, especially for internal lengthwise fins. However, they came up with a correlation to estimate the rate of thermal dissipation based on the fins' dimensions and the temperature difference between the cylinder and surrounding air. Wong and Lee [13] compared the thermal dissipation rates for cylindrical-finned geometries and their fins are made of different materials. Their study showed that aluminum fins have a uniform temperature distribution compared to the fins made of stainless steel. Vogel and Johnson [14] explored the role of airflow in finned settings that are utilized for storing thermal energy. The study demonstrated that it is important to optimize the fins' geometry to ensure efficient airflow, which in turn helps improve the storing process. Chen et al [12] proposed an equation to estimate the thermal dissipation rate in a tube heat exchanger that has finned pipes in it. This equation takes into account the size of tubes and temperature differences.

Park et al [26] proved that lengthwise fins with branched endings have a better thermal dissipation rate by 36% compared to conventional fins in upright cylinders. Senapati et al [25] examined the performance of oval fins in level cylinders, and their analysis concluded that this type of fin has a lower rate of thermal dissipation compared to the rounded type. However, Oval fins are a better choice in case the space dedicated to the fins is confined. Kim and Kim [24] proved that for specially designed fins that have a branched geometry, the thermal dissipation rate is enhanced by 20%, furthermore, this type has a better volumetric efficiency compared to flat fins. The study developed a formula to help optimize the design of these fins for a cylindrical setting. Kim [23] examined the performance of different types of fins, namely flat straight fins, circular fins, linearly increasing and decreasing fins separately, and a mix of the last two. The study proved that the last type has the best thermal performance.

Liu et al [30] explored the effect of lengthwise fins within a rectangular geometry. Their findings show that fins help mix the cold and hot air inside the cavity, which improves thermal dissipation. Furthermore, the study proposed a formula to help optimize the fins' design. Singh and Dash [28] researched the influence of fins in spherical geometry. The outcomes of the study show that the positive impact of fins in terms of thermal dissipation is connected to the material of the fins, the fins' dimensions, and the type of airflow. Park et al [29] proved that shaking flat fins help improve the thermal dissipation significantly. Huang et al [31] evaluated the effect of adding small holes in a rectangular geometry with flat fins. The findings show that adding several small holes is better than adding one big one as they help circulate the air efficiently. Singh and Patil [27] studied the effect of adding embossed fins to a rectangular surface, and the experimental analysis proved that the thermal dissipation increased by 3 times after adding this type of fin.

The above-mentioned literature shows that there is a plethora of empirical and computational studies that have been performed to evaluate the influence of fins' design in natural convection thermal dissipation. However, evaluating the natural convection thermal dissipation in an upright cylinder with a closed lower base has not been reported before. Therefore, computational analysis is conducted on an axisymmetric model of an upright cylindrical geometry with flat fins by solving the fundamental equations of fluid dynamics in Ansys-Fluent 18.1. The goal of this analysis is to find the optimum values of fin spacing that achieve the highest natural convection thermal dissipation. An empirical formula is developed to help in the designing process of such a type of fin, which will be beneficial in the academic and industry sectors.

2. Methodology

The geometry is created in 2D and axisymmetric to optimize the computational resources. The inner tube and the fins are drawn from the axis as per the dimensions shown in Fig. 1. The model is then placed in a domain that is created to contain air at atmospheric pressure. This geometry is then meshed for the solver. The meshing cannot be fine for all the parts of the geometry. The domain has very little changes once it is above the cylinder and the fin and slightly far away. This dimension of the model and domain has been adapted from Jnana et al. [16] since they have already optimized the size of the domain. The meshing is therefore concentrated on the fins and spaces near and around the fins, and the remaining domain had a much coarser mesh. A sample of the mesh is shown in Fig. 2. Grid independence is carried out based on the element size on and around the fins. Since the fins are 1 mm thick, having an edge sizing of 2 mm would not make much sense. Hence the least mesh size is 1 mm around the fins and it is varied further down to find the optimum number of elements. The grid independence results are shown in Fig. 3. It is evident that using the 1 mm edge sizing is perfectly fine for the tests to be conducted.

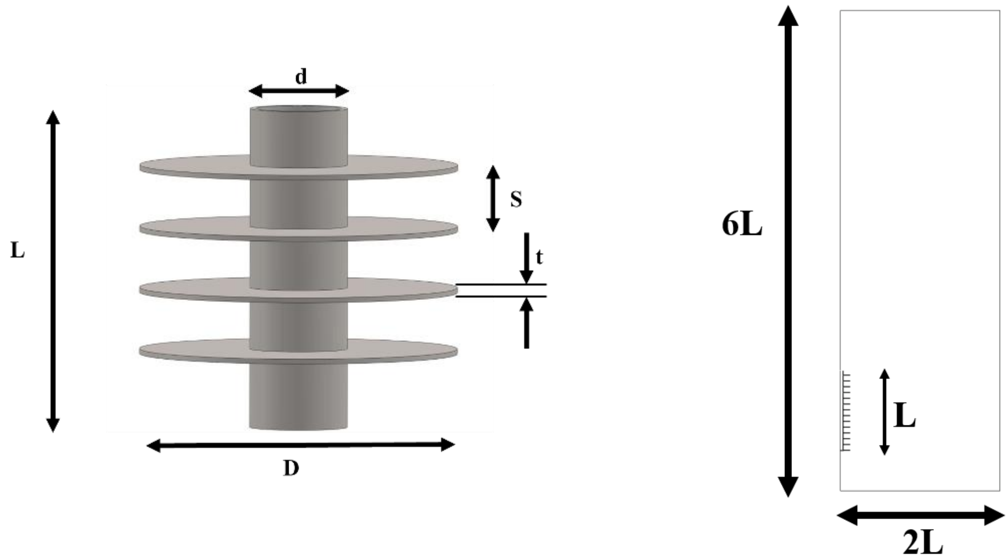


Fig. 1: Schematic of the geometry in 3D (left)and 2D domain (right)

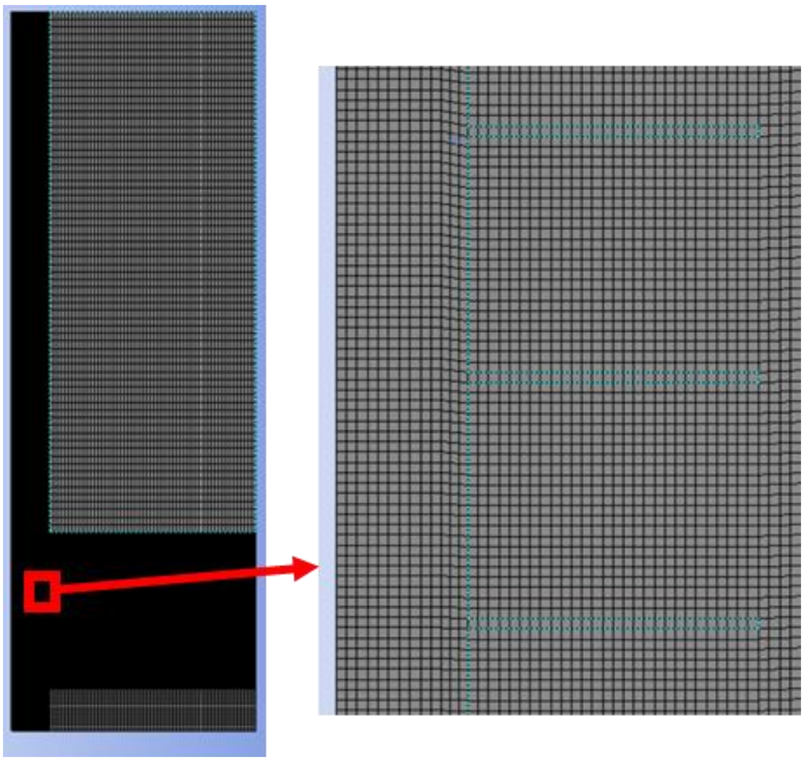


Fig. 2: Mesh geometry within the model and at the fins (zoomed in).

The solver should be density-based as it is intended to investigate natural convection phenomena. The viscous models are chosen to be laminar and k-ε Standard [16]. The discretization schemes are initially set to first-order convergence to get a converged solution and then second-order convergence is set to get a better and more accurate result. The convergence criteria for the residuals is set to 10^{-4} as explained and implemented by Senapati et al. [17]. The solutions are more stable under the mentioned criteria. The boundary conditions are set to be coupled for the fin wall and the base wall. The inner wall is selected to be the source of heat with a constant temperature of 350 K.

Rayleigh number is considered to be the variable for natural convection and it is given by Eq. 1. The general consideration of Rayleigh number for laminar flow on a vertical surface is around 10^9. So, the Rayleigh number is set to be varied as given in Table 1. The main issue with varying the Rayleigh number is to vary the length of the surface to increase the characteristic length. Creating and meshing a new geometry for each Rayleigh number will be a tedious and inefficient way to do the study. Thus, gravity is varied [3] with the length being the constant in the definition of Rayleigh number. The equations solved by the solver and the boundary conditions are given by Eq. (2-9).

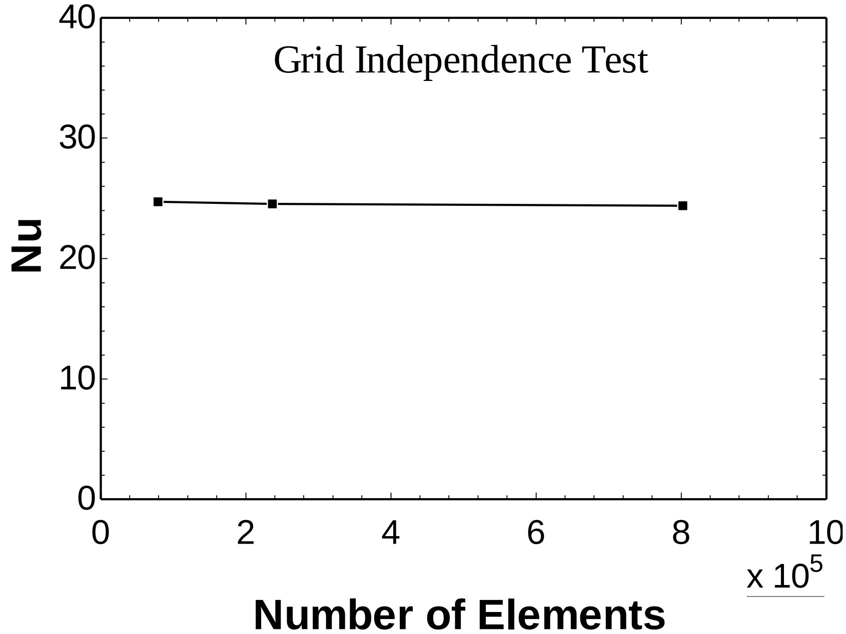


Fig. 3: Grid Independence test.

Table 1: Parameters considered for the study.

S. No.	Parameter	Range of Values
1	Length of the cylinder, L	325 mm
2	Outer diameter of the cylinder, d	12 mm
3	Outer diameter to Fin diameter ratio, d/D	0.625 to 0.25
4	Fin thickness	1 mm
5	Fin spacing to inner diameter ratio, S/d	0.3 to 3.6
6	Rayleigh Number	10^5 to 10^{13}

$$Ra = \frac{g\beta(T_s - T_\infty)L^3}{\vartheta\alpha} \tag{1}$$

Continuity Equation: $$\frac{D\rho}{Dt} + \rho\nabla.\vec{V} = 0 \tag{2}$$

Momentum Equation: $$\frac{\partial\rho u_i}{\partial t} + \frac{\partial}{\partial\lambda_j}(\rho u_j u_i - \tau_{ij}) = -\frac{\partial p}{\partial x_i} + S_i \tag{3}$$

Energy Equation: $$\rho\left[\frac{\partial\rho u_i}{\partial t} + \nabla\cdot(h\vec{V})\right] = -\frac{D\rho}{Dt} + \nabla\cdot(h(k\nabla T) + \phi \tag{4}$$

The turbulent model k-$\in$ is given by the following Eq. (5) and Eq. (6)

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial(\rho k u_i)}{\partial x_i}=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k+G_b-\rho\in-Y_m+S_k \quad (5)$$

$$\frac{\partial(\rho\in)}{\partial t}+\frac{\partial(\rho\in u_i)}{\partial x_i}=\frac{\partial}{\partial x_i}\left[\left(\mu+\frac{\mu_t}{\sigma_\in}\right)\frac{\partial\in}{\partial x_j}\right]+C_{1\in}\frac{\in}{k}(G_k+C_{1\in}G_b)-C_{2\in}\rho\frac{\in^2}{k}+S_\in \quad (6)$$

where, $\mu_t=\rho C_\mu \dfrac{k^2}{\in}$ (7)

At the solid -fluid interface, the boundary condition is given as

$$k_s\left(\frac{\partial T}{\partial n}\right)=k_{eff}\left(\frac{\partial T_f}{\partial n}\right) \quad (8)$$

where, $k_{eff}=k_f+\dfrac{c_p\mu_t}{\Pr_t}$ (9)

The heat transfer coefficient and Nusselt number is calculated using the formula as given in Eq. (10) and Eq. (11). Another set of simulations have been run without the fin to analyze the effectiveness of the various cases. This would eventually help in determining the optimum fin spacing and fin length. The effectiveness of the fin is calculated by equation Eq. (12).

$$h=\frac{Q}{A_s\times(T_s-T_\infty)} \quad (10)$$

$$Nu=\frac{hl}{k} \quad (11)$$

$$Effectiveness,\varepsilon=\frac{Heat\ transfer\ with\ fin}{Heat\ transfer\ without\ fin} \quad (12)$$

The validation of a sample set of results is conducted against correlations given by McAdams [32] for laminar flow and Eckert and Jackson [33] for turbulent flow. The turbulent flow data closely aligns with the correlation data, whereas laminar flow is consistently underestimated by nearly 20%. Given that the mesh quality and boundary conditions exhibit satisfactory agreement with established correlations in turbulent flow, the model is deemed acceptable for acceptance.

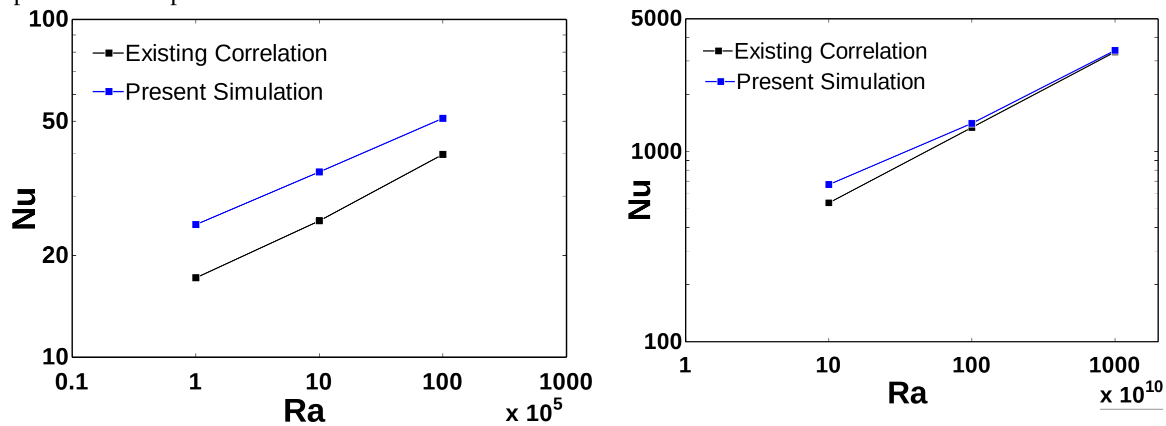


Fig. 4: Validation against correlations by McAdam [32] for laminar (left) and Eckert and Jackson [33] for turbulent (right) flows (for d/D = 0.4167 and S/d = 0.9)

3. Results and discussion

The Nusselt number is clalculated and plotted against Rayleigh number for all the cases. For the first part of the analysis, the S/d is kept constant to observe the effects of the fin length (d/D) on the Nusselt number. The most known and common observation is the Nusselt number increased as the Rayleigh number increased. For all S/d values, the Nusselt number increased with the increase in d/D value. This shows that the larger fins have better heat transfer properties. This is also a very instinctively understood result. However, all the Nu values at the same Rayleigh number are very close to each other as seen in Fig. 5. There is a distinct difference only at the higher Rayleigh numbers. To conclude this behaviour, the next criteria had to be studied. The S/d values are varied for each d/D value in the Nu vs Ra plots. It is observed that the Nusselt numbers of the least S/d value (S/d = 0.3) are much smaller than the other three, especially at high d/D values. It can be observed from the Fig. 6. that the values are very close but not overlapping each other for the cases of d/D = 0.625. This behavior is reduced and distinct values of Nu are demonstrated as the d/D value is decreased to 0.25. At the lowest d/D value, the Nusselt number increased with the S/d value. It can be safely concluded that fin spacing does not having a significant impact after a certain value and having a higher number of fins doesn't always result in a higher value of Nu.

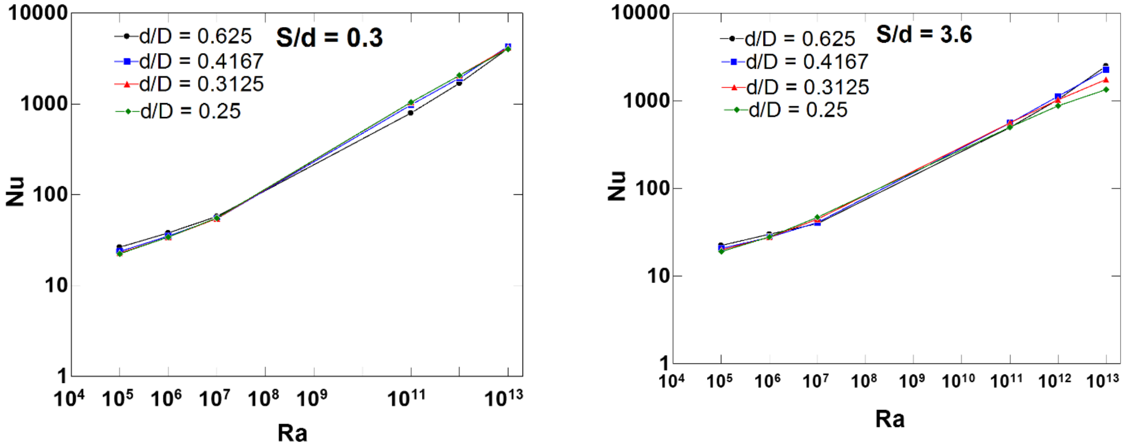
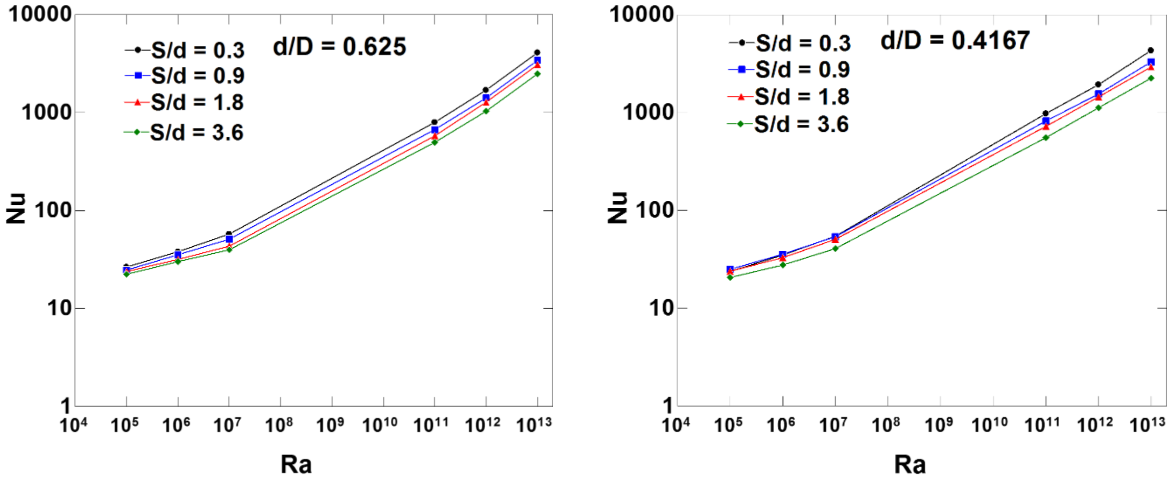


Fig. 5: Nu vs Ra plot for various d/D values at constant S/d value



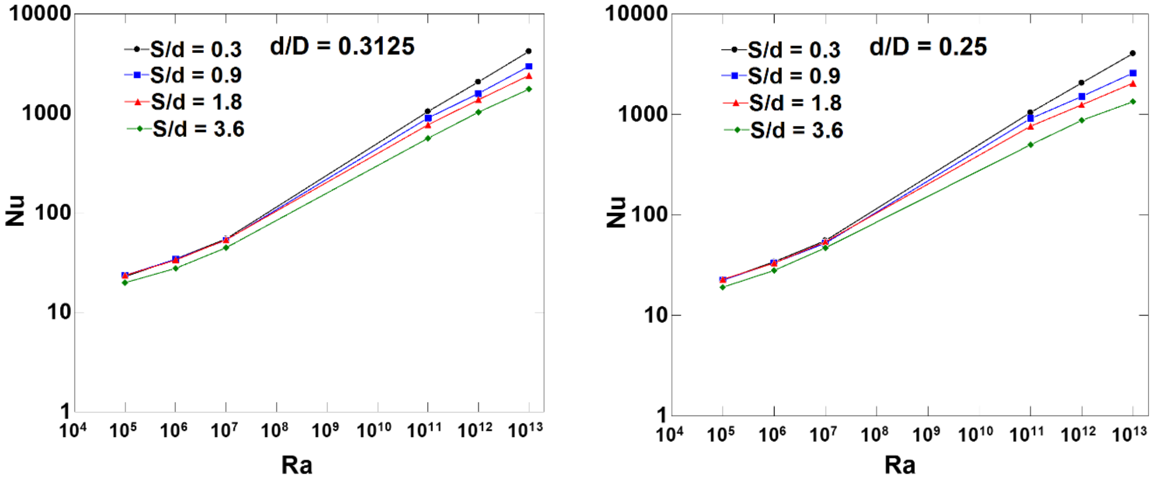


Fig. 6: Nu vs Ra plot for various S/d values at constant d/D value

The heat flow rate from the combined fin and base walls is also observed to find out the optimum length where the heat flux would be the maximum. No such trend of increasing to a point and then decreasing is observed since the range of Rayleigh numbers is too large. The highest value of Ra had the highest value of Nu. However, it is observed that fin spacing and fin length do not show any abnormal variations. The highest number of fins and the largest fins gave the highest Nusselt number. It's worth noting that the disparity in heat flow rate values fails to reflect the expected difference. As an illustration, when the number of fins triples (changing from S/d 0.9 to S/d 0.3), the heat flow rate demonstrates a maximum increase of 50%. The same trend is observed when looking at the graphs of Heat flow rate vs Ra for only the fin walls. It is obvious that the heat transfer from the base is dominant in the lower number of fins but that doesn't take away the fact that the overall heat flow rate is not increased by a great margin when the number of fins is considered.

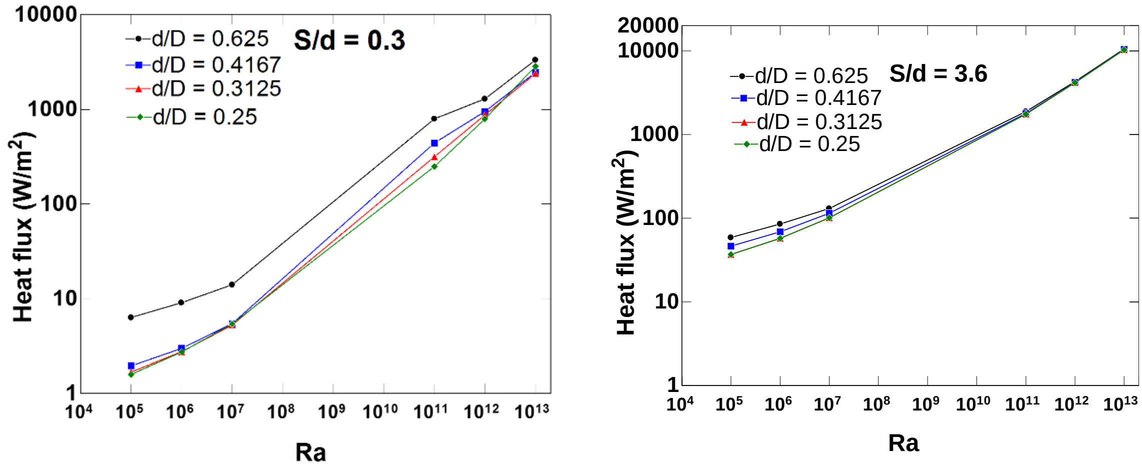


Fig. 7: Heat Flux vs Ra plot for various d/D values at constant S/d value

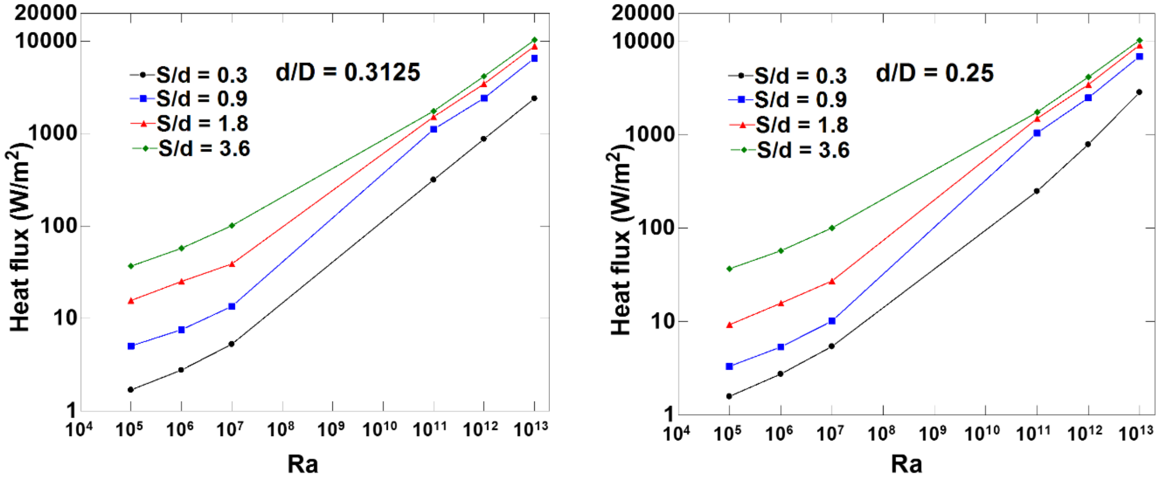


Fig. 8: Heat Flux vs Ra plot for various S/d values at constant d/D value

The effectiveness plots present a slightly more intriguing aspect. On observation, Ra of around 10^{11} seems to be the significant value of highest effectiveness for all cases. For a higher S/d value, the effectiveness do not change much between the Rayleigh numbers 10^9 and 10^{11}. This is because for the S/d value of 3.6 only 3 fins are available for the heat transfer. Hence the values are very close to 1 and not changing too much with the highest being around 2 for the largest fins. In some cases, at high Ra values, the effectiveness values are very close for different d/D values. It is still holding up the trend of having the highest value of effectiveness for the smallest d/D values but nonetheless, it seems to be an area of consideration if a 25% increase in the fin diameter is only increasing the effectiveness by 4%. Based on all of these results, S/d of 0.3 or 0.9 with a d/D of 0.25 to 0.3125 with a Ra of around 10^{11} seems to be the ideal option for the best heat transfer as well as effectiveness. The sample plots are as given in Fig. 9 and Fig. 10.

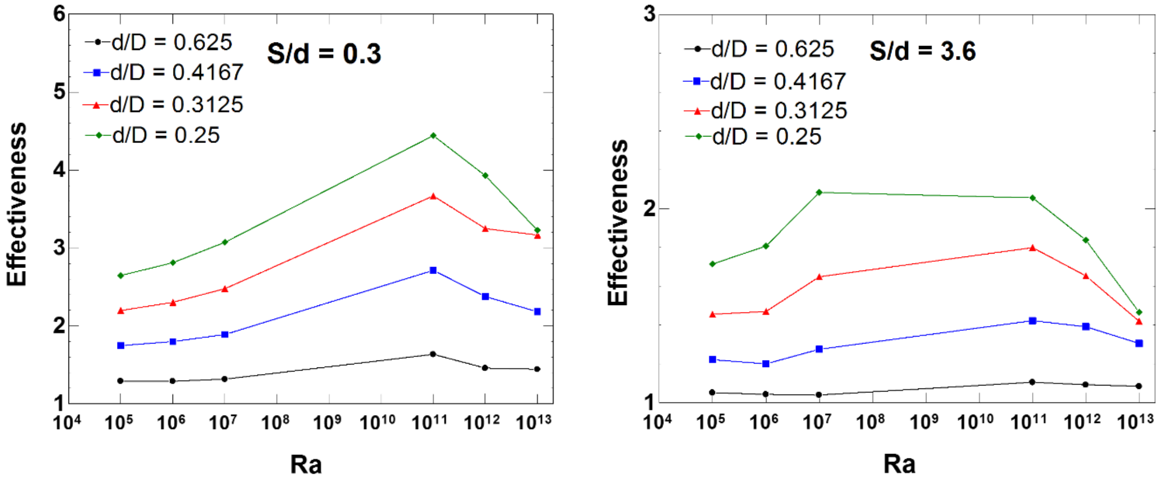


Figure 9: Effectiveness vs Ra plot for various d/D values at constant S/d value

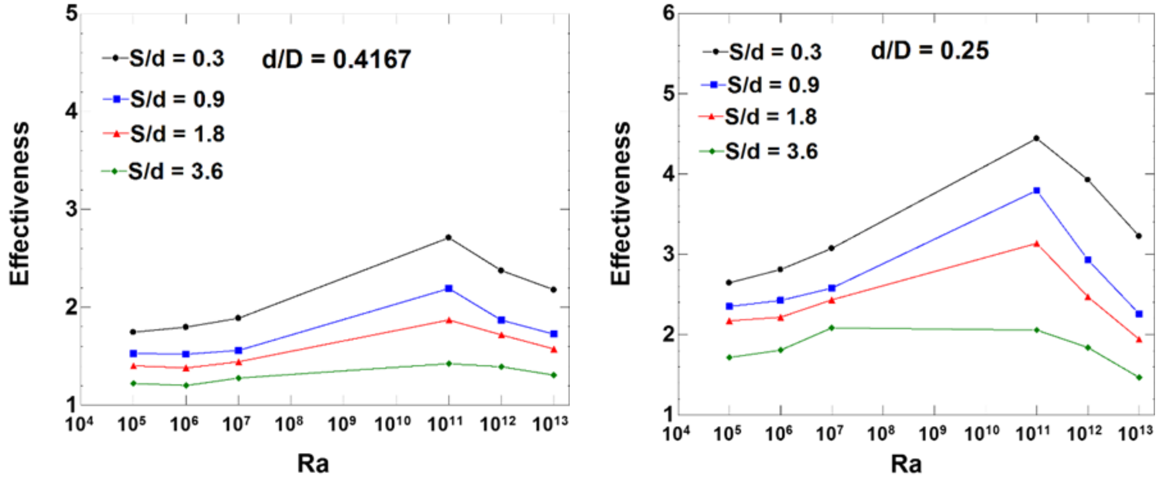
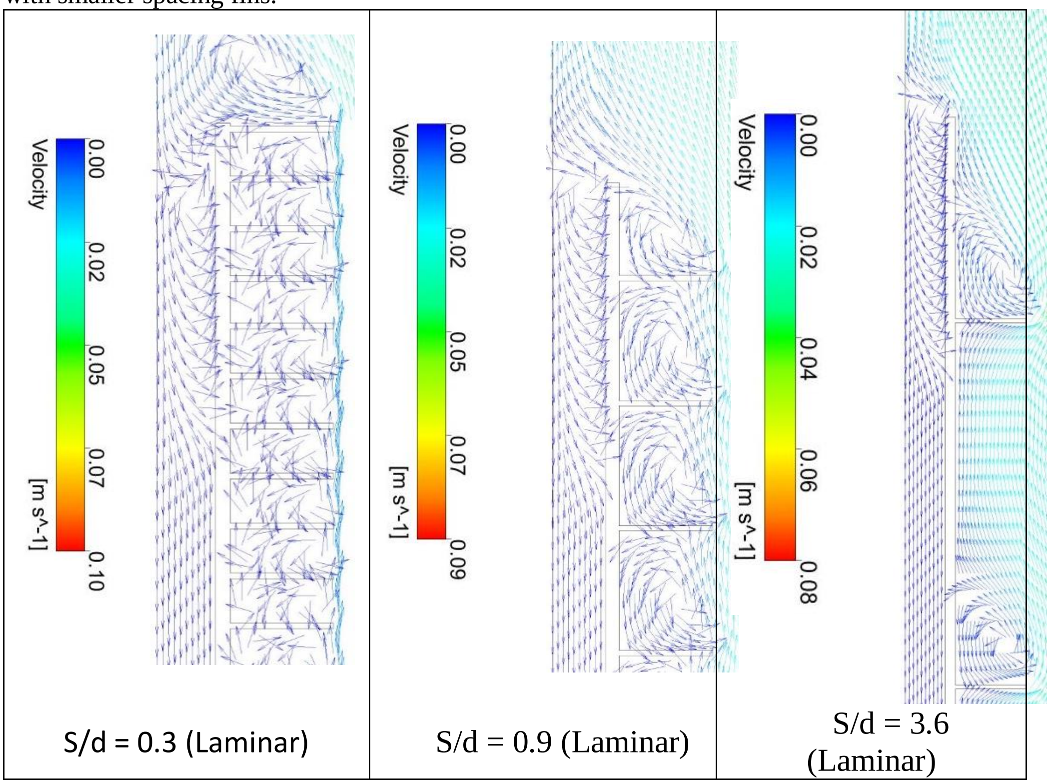


Fig. 10: Effectiveness vs Ra plot for various S/d values at constant d/D value

The graphs are one part of the study that helps in understanding the heat transfer characteristics. However, having contours of velocity and temperature will further aid in understanding the behavior of fluid under those different conditions. The velocity vectors in Fig. 11 point out that most of the flow is concentrated around the fins despite an opening available for flow behind the fins. The reason is that the high temperatures of that area are now allowing any fresh air to come into the space since it gets heated near the entrance and flows away. The laminar flow cases show that the flow doesn't go in between the fins for cases of S/d = 0.3 due to very tight gaps. However, in turbulent flow, it is seen that vortices are formed around the fin gaps thus enhancing the heat transfer in that region. The presence of turbulence is far less in cases for S/d > 0.9 due to the large spacing of the fin. Hence, the heat transfer is much lesser and is similar to the laminar flow cases as depicted by the values of effectiveness in Fig. 10. The velocity contours presented in Fig. 12 and Fig. 13, indicate the same explanation. The velocity is very low near and in between the fins. The fins with larger spacing can be seen to have a less distributed velocity but the vectors indicate that not a lot of turbulence is created in the space of two consecutive fins for those cases as compared with smaller spacing fins.



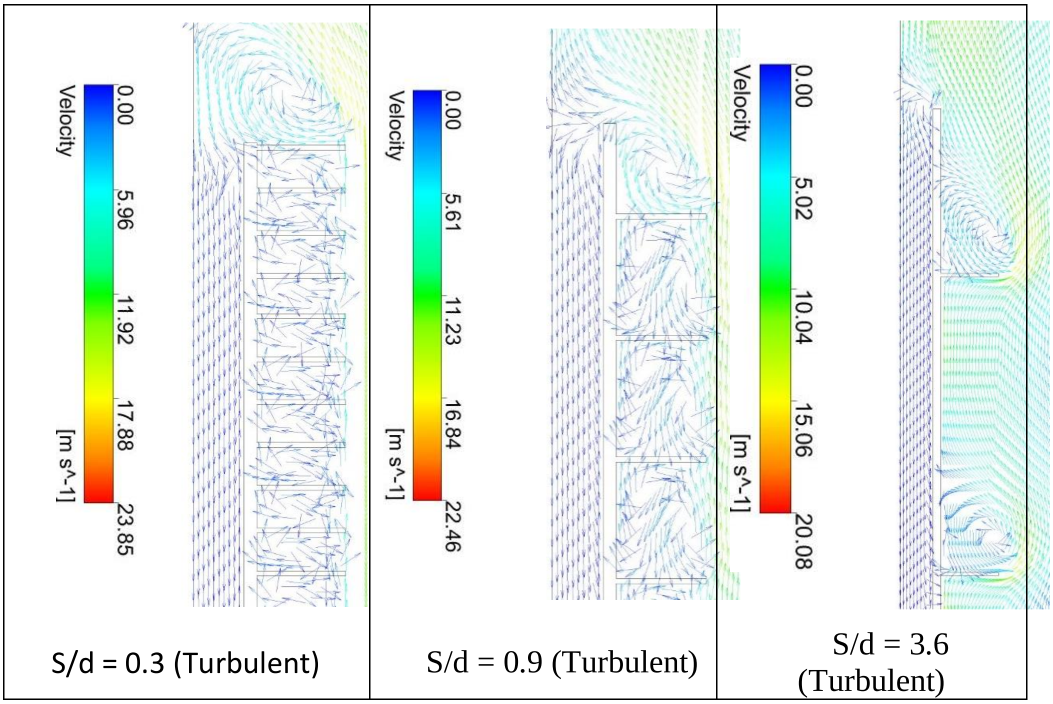
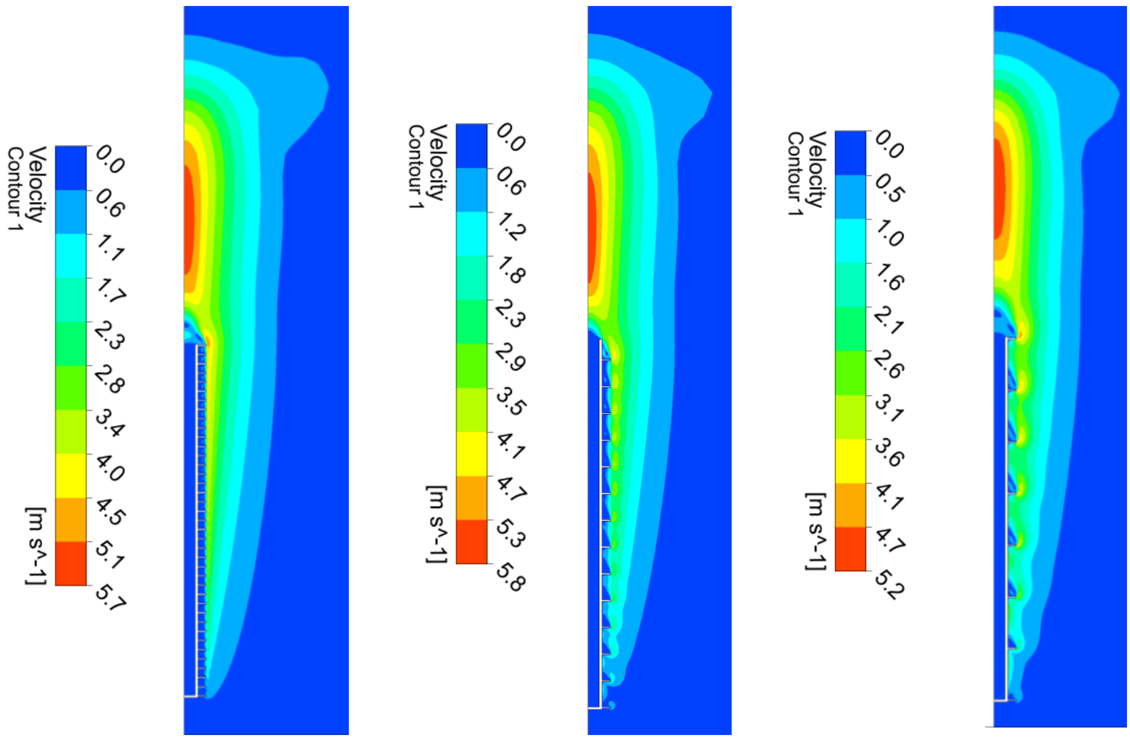


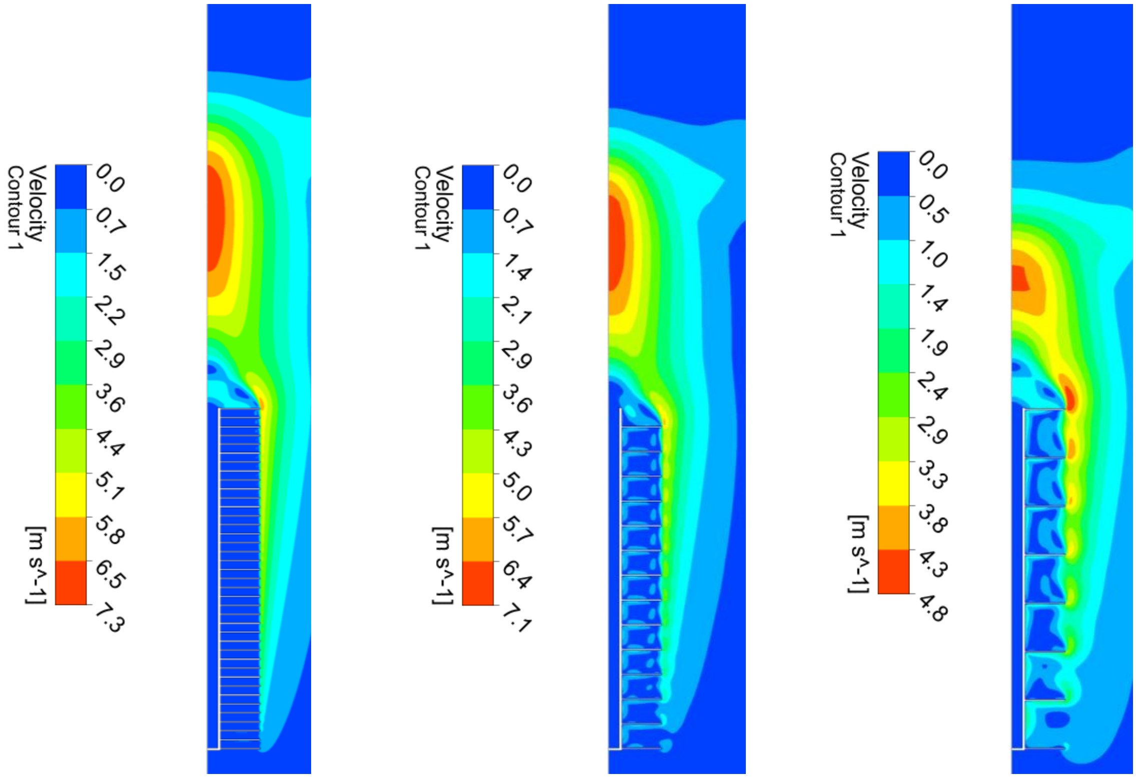
Fig. 11: Velocity Vectors at d/d = 0.3125 and Ra = 10^6 (Laminar) and Ra = 10^{12} (Turbulent)

The temperature contours are observed for all the cases and some sample contours are given in Fig. 14 – 16. The open space of the cylinder is predominantly at the highest temperature but in some cases, the temperature is slightly lesser at the opening due to turbulence created at higher Rayleigh numbers. This is confirmed by the vector plots for those cases. As the fin's diameter increased, the temperature around the fins is at a lesser value than the fins with a smaller diameter. The spacing of the fins also had an impact of creating a lot cooler zones beside the fins. Despite heat being dissipated in these scenarios, the data suggests that the quantity of heat transferred is significantly lower compared to configurations with a greater number of fins. The cases with large S/d values are also less effective and can be seen from the contours.



S/d = 0.3 S/d = 0.9 S/d = 1.8

Fig. 12: Velocity contours of various models with different S/d at Ra = 1011, d/D = 0.625



S/d = 0.3 S/d = 0.9 S/d = 1.8

Fig. 13: Velocity contours of various models with different S/d at Ra = 1011, d/D = 0.3125

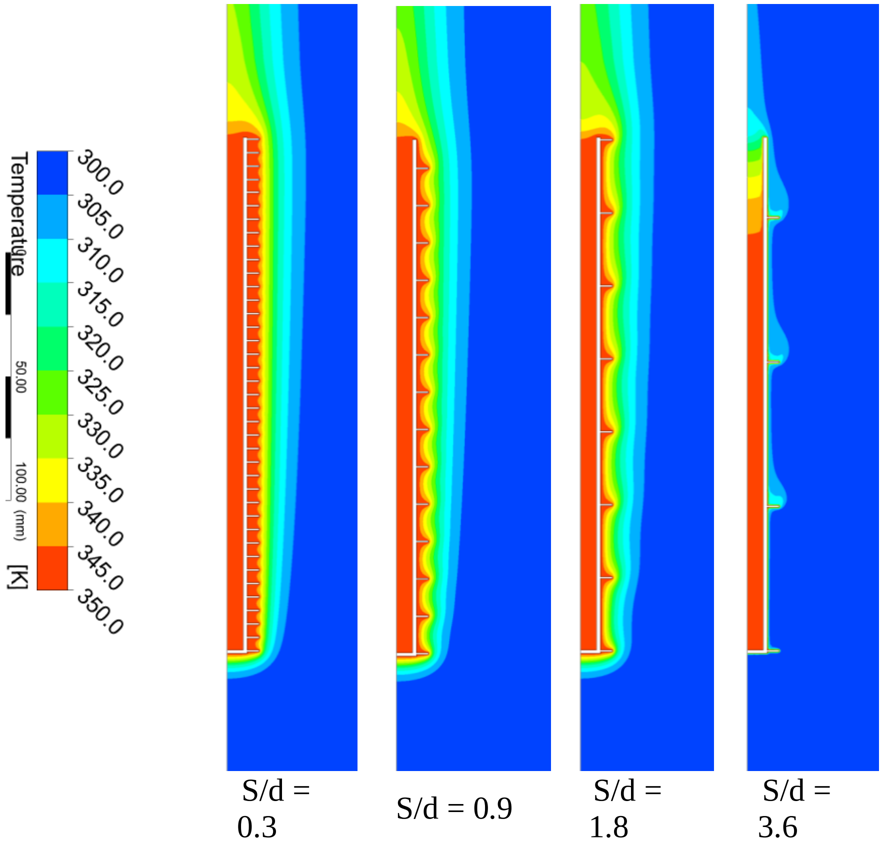


Fig. 14: Temperature contours with different S/d at Ra = 10^6, d/D = 0. 625

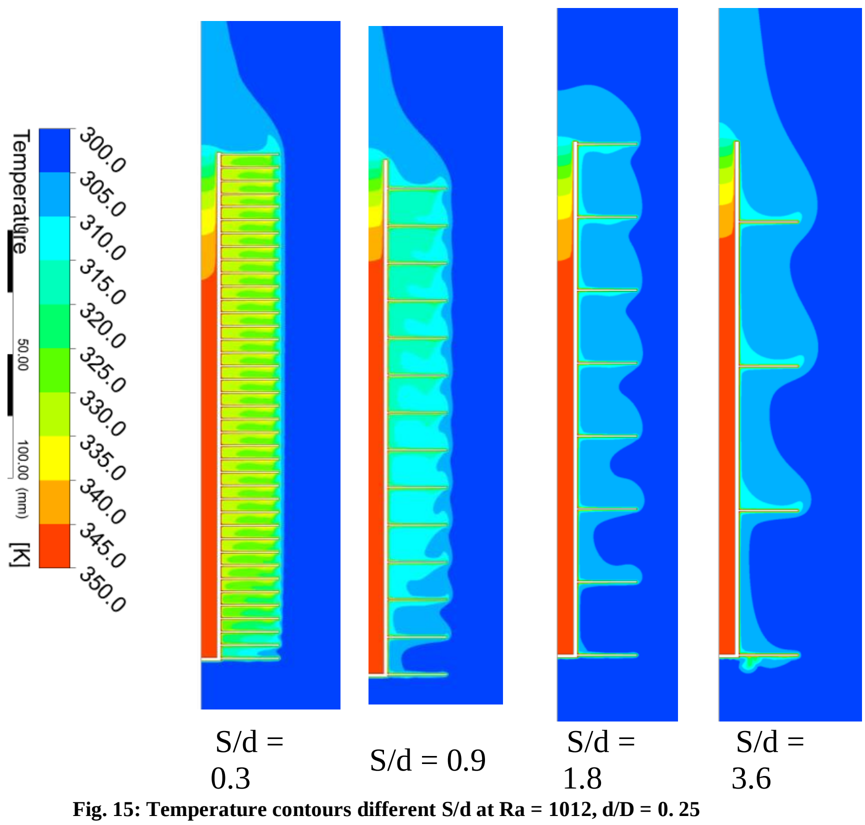


Fig. 15: Temperature contours different S/d at Ra = 1012, d/D = 0. 25

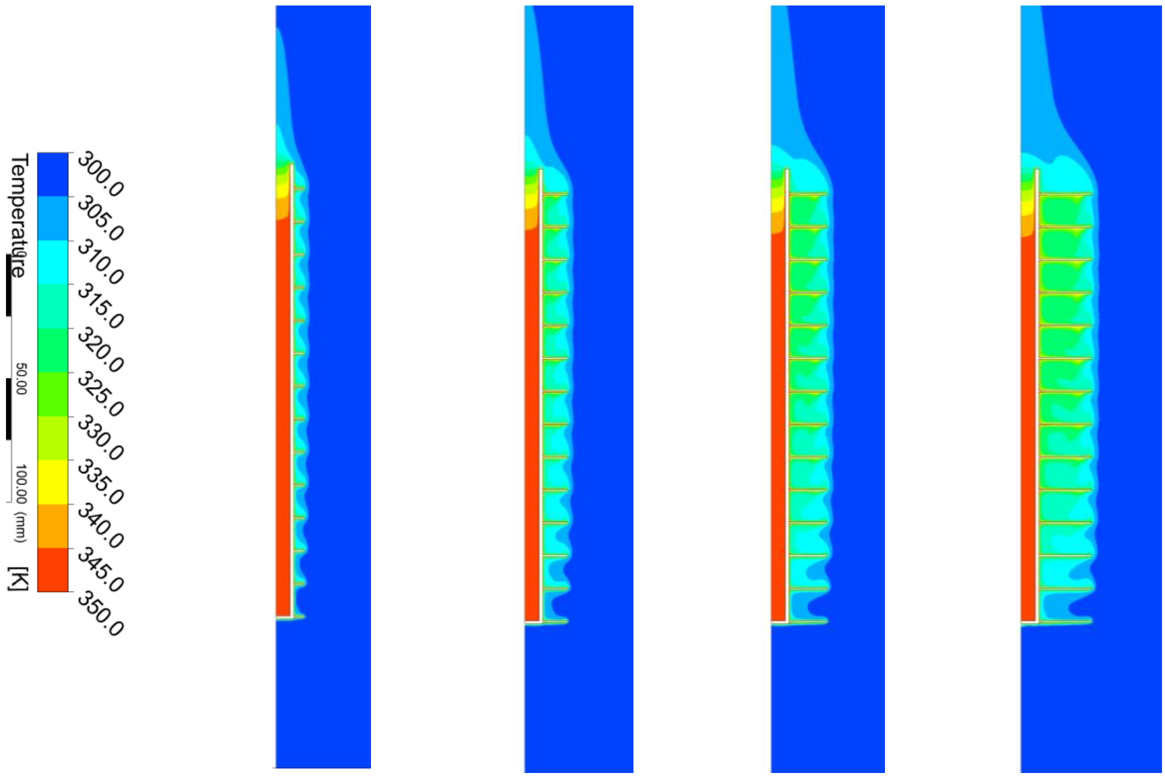


Fig. 16: Temperature contours with different d/D at Ra = 10^{11}, S/d = 0.9

4. Conclusion

To enhance realism, a cylindrical tube with one end exposed to the surrounding atmosphere and the other end closed is selected for analysis. Annular fins of various sizes and spacings are then attached to the cylinder, and the numerical solution of this configuration is validated against established correlations. Geometrical and flow parameters are systematically altered to analyze flow characteristics and determine an optimal combination of fin spacing and fin diameter for this specific application. The following key points summarize the conclusions drawn from this study.

- Increasing the number of fins threefold (from S/d 0.9 to S/d 0.3) resulted in a maximum 50% increase in the heat flow rate.
- In the laminar region as the Re is increased by a factor of 10 and the d/D value is increased by 50%, the Nu is increased by a maximum of 2% and in the turbulent region, the Nu is increased by a maximum of 35%
- The Nusselt number increased by a factor of 50 to 190 from the minimum to the maximum Rayleigh number.
- The effectiveness is maximum at Ra = 10^{11} at value of 4.44 for S/d of 0.3 and d/D of 0.25. The maximum values for effectiveness are observed at lower d/D and S/d values.
- The open cylinder does not encounter much heat transfer near its base on the inner side. Most of the heat transfer is carried out by the fins. It is also observed that the opening contributed to a small percentage of heat transfer in turbulent flows due to vortices forming from the last fin tip.

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