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ORIGINAL ARTICLE

Numerical analysis of heat transfer enhancement in grooved vertical Multi-Cylinders at various groove geometries

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KEYWORDS

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Multi cylinders, Natural convection;
Vertical cylinder

Abstract In this study, the natural convection enhancement in grooved vertical multi-cylinders at various groove geometries is investigated. The effects of several grooves ranging from 3 to 7, groove thickness ranging from 0.25 to 1 mm, and cylinder surface temperature ranging from 350 to 500 K at different Rayleigh numbers are examined. The current study was simulated using the finite volume method using CFD with a laminar steady-state condition. The SIMPLE scheme is used for the pressure–velocity coupling discretization and the second-order upwind method is utilized to discretize the momentum and energy equations. The results obtained from the present research show that the presence of grooves on the cylinders will increase the heat transfer surface, create and intensify the secondary flow and mixing, and ultimately increase the heat transfer. Moreover, by increasing the number of grooves and its thickness, the amount of heat transfer increases dramatically. It's also found that the groove thickness parameter's effectiveness on heat transfer is more than the groove number parameter. Ultimately, it's demonstrated that using grooved cylinders leads to a 14 % augmentation in Nusselt number in comparison with employing plain cylinders.

1. Introduction

In recent decades, due to the increase in energy consumption and the necessity for energy saving, researchers have paid more attention to the design and optimization of engineering systems [1,2]. Heat transfer systems are systems that have

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Nomenclature

D	Diameter [m]
e	Groove size [m]
F	fixed coefficient
g	Gravity acceleration [m/s^2]
Gr	Grashof number
h	Heat transfer coefficient [W/m^{2}K]
K	Thermal conductivity [W/mK]
L	Length [m]
N	Groove number
Nu	Nusselt number
Pr	Prandtl number
p	Pressure [Pa]
q	Heat transfer rate [W]
Ra	Rayleigh number
T	Temperature [K]
u	Velocity magnitude in x-direction [m/s]
v	Velocity magnitude in y-direction [m/s]
w	Velocity magnitude in z-direction [m/s]

Greek symbols

α	Thermal diffusivity [m^2/s]
β	Thermal expansion coefficient [1/K]
ϑ	Kinematics viscosity [m^2/s]
ρ	Density [kg/m^3]

Subscripts

∞	ambient
s	Wall surface

attracted the attention of researchers due to their inexpensive [3,4]. But the inherent shortcoming of these systems is the low rate of heat transfer compared to forced convection systems [5,6]. One of the ways to fulfill this shortcoming can be the use of fins that were extensively studied by [7–9]. Through these studies, some were focusing on heat transfer enhancement over the arrays of hot pipes, which were widely used in the heat exchanger industry. It is important to investigate heat transfer behavior in many industrial applications, such as cavity [10], sinusoidal parallel-plate heat exchanger [11], vertically oriented plate fins [12], helically grooved shell and tube heat exchangers [13], vertical cylinders with branched fins [14] and Nanofluid flow over a cone [15].

Some studies were performed on natural convection from the pipes. Ganesan and Rani [16] have studied transient natural convection on a vertical cylinder with varying surface temperatures. They showed the profiles of temperature and velocity and calculated the skin friction factor and Nusselt number. Jarall and Campo [17] studied natural convection from a vertical cylinder heated by electricity and immersed inside air. They concluded three correlations for variation of Nusselt number with Rayleigh number and position to diameter y/d. Jin et al. [18] studied the effect of pulsating flow agitation on heat transfer in a triangular grooved channel. They found heat transfer enhancement will increase as the Reynolds number decreases. Kim et al. [19] studied natural convection in a square enclosure with a circular cylinder at different vertical locations. Their study focused on natural convection between a cold outer square and an inner hot cylinder at different Rayleigh numbers. Bilen et al. [20] studied the effect of groove geometry on heat transfer from internally grooved tubes. They found out that grooved tubes are thermodynamically beneficial for Reynolds numbers up to $Re = 30000$ for circular and trapezoidal grooves and $Re = 28000$ for rectangular grooves. Heo and Chung [21] experimentally studied natural convection heat transfer over inclined cylinders for Ra $= 1.69\times 10^8 - 5.07\times 10^{10}$and angle inclination between $0^\circ - 90^\circ$. Results show that heat transfer for cylinders with 30° angle of inclination were only 5 % lower than horizontal cylinders. Chamkha et al. [22–28] investigated the effects of different parameters such as nanoparticles, magnetic field, local thermal non-equilibrium and nano-encapsulated Phase Change Materials (PCMs) on heat transfer rate in the cavity. Their results show that all the mentioned parameters have significant impacts on the fluid flow, the temperature distributions and heat transfer enhancement. The influences of Prandtl number, length and diameter on the natural convection inside open vertical pipes numerically investigated by Ohk et al. [29]. Ibrahim et al. [30], in a natural convection on a vertical plate variable with temperature by presence of magnetic field, deduced that by intensifying the magnetic field temperature at thermal boundary layer enhances and velocity decreases at its boundary layer. Ouaf [31] reported a similar result by solving radiation effects in a magnetic field on a porous plate. Ali et al. [32] investigated the natural convection on a vertical plate and radiation effects in the presence of a magnetic field applying heat generation and viscose heating. Alijani et al. [33] examined the effect of design and operating parameters such as groove widths, filling ratio, and different heat flux values on the thermal performance of four aluminum flat-grooved heat pipes. Quantify and compare the natural convection heat transfer enhancement of perforated fin array with various perforation angles, fin spacing, the pitch of perforation, perforation diameter, and heater inputs experimentally investigated by Pankaj et al.[34]. They present optimum fin for maximum heat transfer rate. Senapati et al. [35] investigated the natural convection heat transfer over an annular finned cylinder for the laminar range of Ra number. They perform optimization studies to find the best fin spacing and fin-to-tube diameter ratio for maximum heat transfer.

In the present study, the effects of natural convection around array of grooved cylinders is investigated numerically. The effect of using helical grooves on the array of vertical cylinders in improving the heat transfer rate of natural convection was not much considered in previous studies as it has many industrial applications such as the stack of fire heaters in the refinery. Then the effect of different numbers and thicknesses of the grooves and different cylinder surface temperatures on the heat transfer rate and flow field were investigated and discussed.

2. Mathematical modeling

2.1. Governing equations

To simulate the array of cylinders, the governing equations need to be solved for the laminar steady-state flow. These equations are as follows:

Continuity equation [36]

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

Momentum equation in ×, y and z directions

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

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

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

Energy equation [37]

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

where in Eq. (3),β is the thermal expansion coefficient

$$\beta = -\frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right) \approx -\frac{1}{\rho}\left(\frac{\rho - \rho_\infty}{T - T_\infty}\right) \tag{6}$$

The dimensionless parameters are the following [38–41]:

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\vartheta\alpha} \tag{7}$$

$$Pr = \frac{\vartheta}{\alpha} \tag{8}$$

$$Gr = \frac{Ra}{Pr} = \frac{g\beta(T_s - T_\infty)D^3}{v^2} \tag{9}$$

and the Nu number of the natural convection around the cylinders can be defined as [40,41]

$$Nu = \frac{hD}{K} \tag{10}$$

The heat transfer coefficient h can be calculated with Newton's law of cooling:

$$Q = \pi D L h(T_s - T_\infty) \tag{11}$$

2.2. Geometry

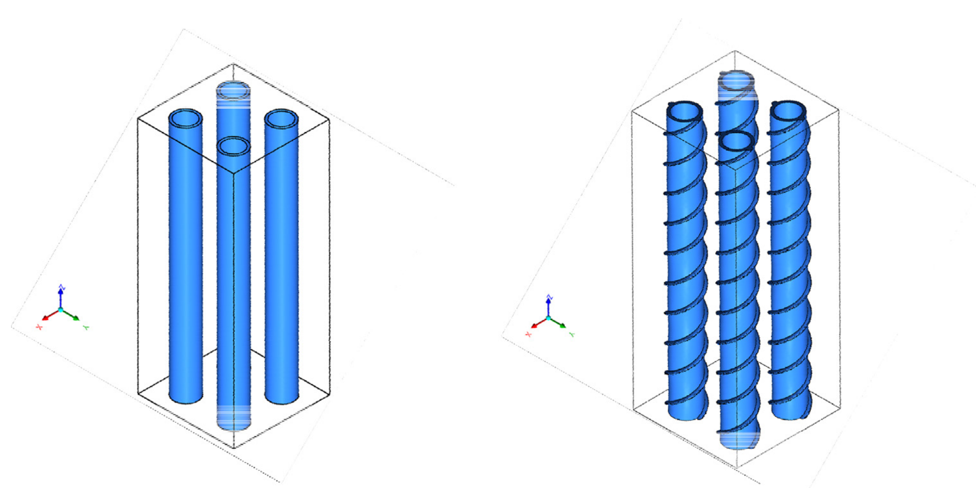
Four vertical cylinders were placed in a box with dimensions of $20 \times 20 \times 50mm$. The outer diameter and an inner diameter of each cylinder are $5mm$ and $4mm$ which makes the thickness of each cylinder to be $0.5mm$. Fig. 1 shows plain cylinders and spiral-grooved cylinders in an isometric view.

2.3. Operating conditions and numerical procedure

For the present study, a variation of several spiral grooves and the thickness of the grooves was considered. To perform analysis, three grooves of 3, 5, and 7 were chosen as well as three grooves with thickness of 0.25, 0.5, and 1 mm. the surface temperature of each grooved cylinder was selected to be constant and equal to 350, 400, 450, and 500 K. Table 1 shows the operating conditions of analysis:

Fig. 2 shows all three grooved cylinders with different numbers of grooves and Fig. 3 shows the cylinders with different groove sizes:

For temperatures discussed in Table 1, Rayleigh number lies between 0.5×10^6 and 2.2×10^6 which indicates that the regime flow of analysis is laminar all over the tests. The working fluid of analysis is air with a thermal expansion coefficient of $0.0033\frac{1}{K}$. Fig. 4 shows the boundary conditions applied to the investigation. The current study was simulated using the finite volume method using CFD with a laminar steady-state condition. The SIMPLE scheme is used for the pressure–velocity coupling discretization and the second-order upwind method is utilized to discretize the momentum and energy equations. To apply the weight term in Navier-Stokes equations, the Boussinesq approximation was used. The solutions converged when the residual values of the momentum and energy were 10^{-6}.



(a) (b)

Fig. 1 Isometric view of geometry cylinders for (a) plain cylinders, (b) spiral grooved cylinders.

Table 1 Operating conditions and characteristics of cylinders for the analysis.

No	cylinder type	groove thickness (mm)	number of grooves	surface temperature	number of calculations
1	smooth	–	–	350–500	4
2	grooved	0.5	3	350–500	4
3	grooved	0.5	5	350–500	4
4	grooved	0.5	7	350–500	4
5	grooved	0.25	5	350–500	4
6	grooved	1	5	350–500	4

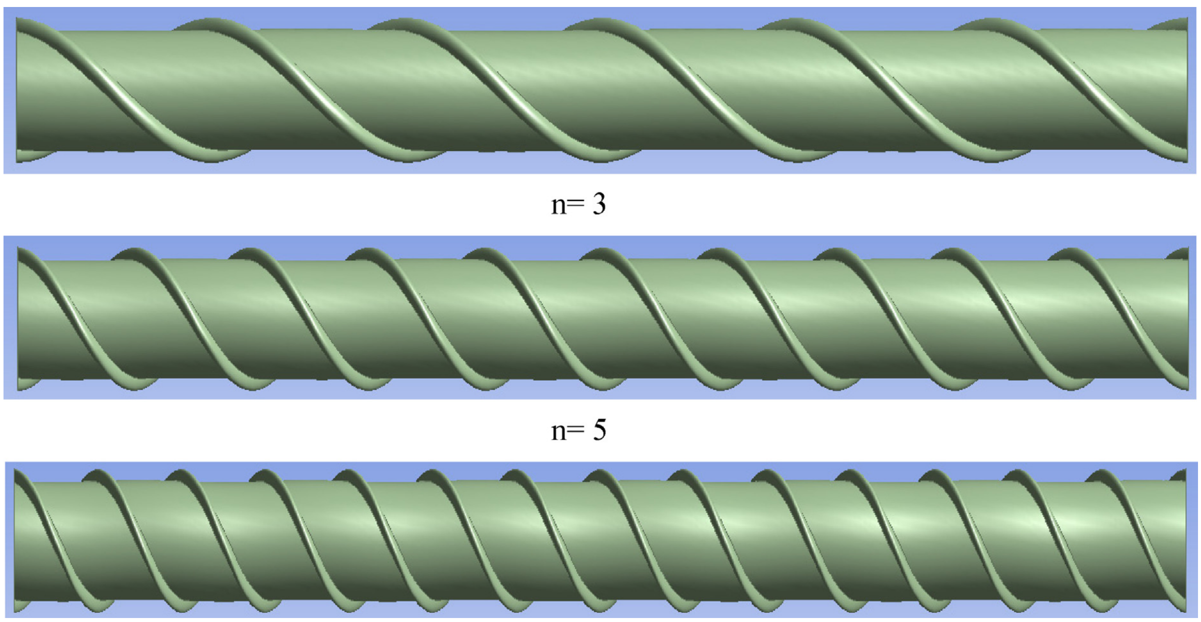


Fig. 2 Cylinders with different groove number.

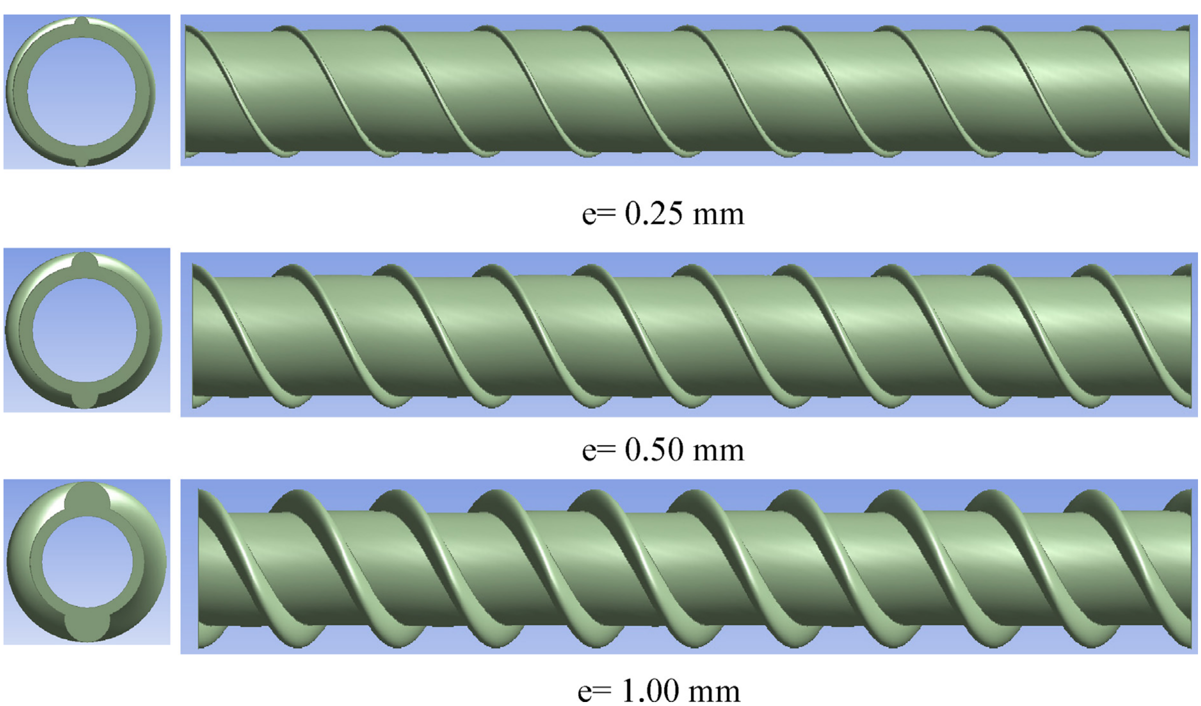


Fig. 3 Cylinders with different groove sizes.

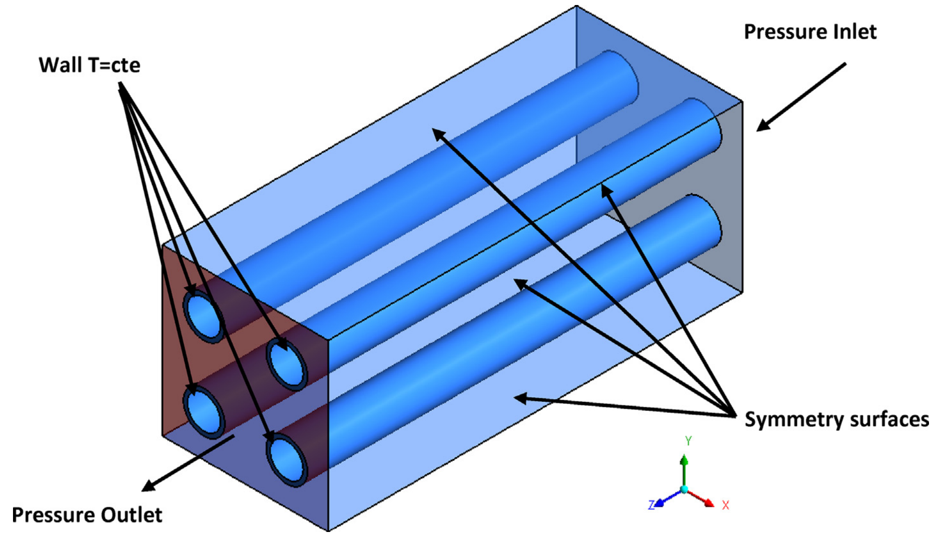


Fig. 4 Boundary conditions applied to the analysis.

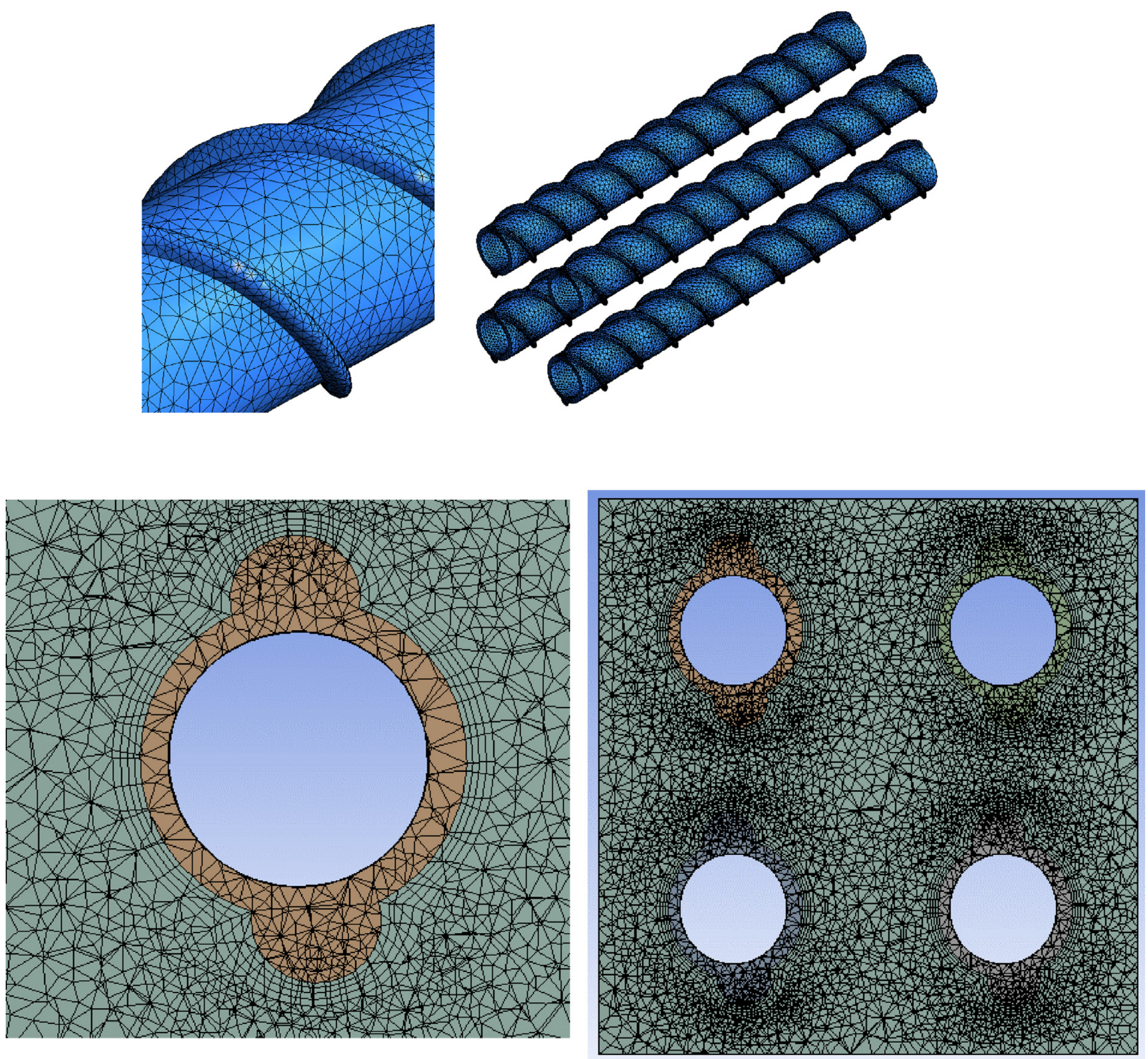


Fig. 5 Grid arrangement in the computational domain.

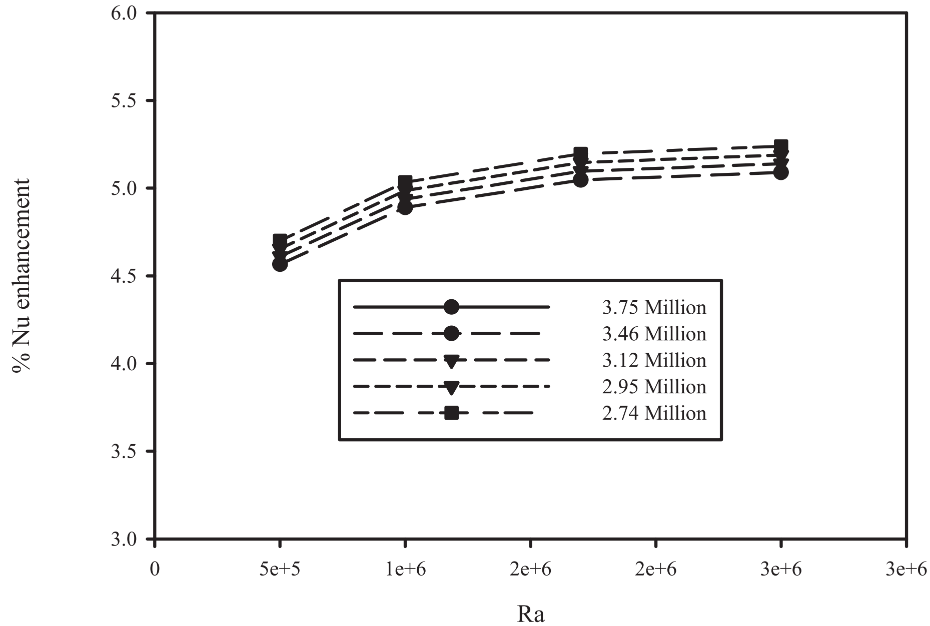


Fig. 6 Variation of *Nu* enhancement with the number of mesh.

2.4. Mesh analysis

As with numerical studies [42,43], grid studies can increase the accuracy of calculations and measured parameters. Due to the complexity of the present geometry, the tetrahedron meshes are chosen. Fig. 5 shows the generated mesh in this study. For more accuracy, five layers of the boundary layer mesh with a first-layer thickness of 0.1 mm were generated around the grooved cylinders. To obtain correct results from the present study, results were made to be independent of the grid used in the analysis. On the other hand, a minimum number of meshes must be used for analysis. Therefore, several cases were studied for the model with five grooves and a groove thickness of 0.5*mm*. Fig. 6 shows the plot of the *Nu* versus the mesh number for different case studies. As can be seen, the results of the *Nu* number are similar between 2.74 million and 3.75 million meshes used for two cases. Hence, the case with 2.74 million meshes was chosen for the present study.

2.5. Validation of results

Validation of the results in numerical and experimental work can guarantee the accuracy of the extracted results [44–47]. To validate the present study with other validated research, at first, an analysis was performed on a 2D vertical cylinder in the natural convection regime, and the results were compared with the results of Churchill [48] and Eckert [49]. Fig. 7 shows the geometry of the 2D vertical cylinder used for the validation and boundary conditions:

For this case study, the Rayleigh number was chosen as $10^4 \le Ra \le 10^8$. Working fluid is air with Pr = 0.72, and the temperature difference between cylinder and air is 10*K*. Using Churchill correlation [48], the Nusselt number is as in Eq. (12):

$$Nu = F\left[0.825 + \frac{0.387(Ra)^{\frac{1}{6}}}{\left[1+\left(\frac{0.492}{Pr}\right)^{9/16}\right]^{\frac{8}{27}}}\right]^2 10^{-1} \le Ra \le 10^{12} \tag{12}$$

Where F is as in Eq. (13):

$$F = 1.3\left[\frac{\frac{L}{D}}{Gr}\right]^{1/4} + 1 \tag{13}$$

Also, Eckert correlation for Nusselt number is as in Eq. (14):

$$Nu = 0.59F(Ra)^{0.25} 10^4 \le Ra \le 10^9 \tag{14}$$

Table 2 shows the comparison of results for a single vertical cylinder using methods of the present study with the results of Churchill and Eckert [48,49].

Also, Fig. 8 shows the plots of comparison between the results mentioned in Table 2:

3. Results and discussion

To estimate the effect of grooving on cylinders, at first, a grooved cylinder with 5 grooves and a groove thickness of 0.5*mm* was analyzed and compared with a cylinder without grooves. Fig. 9 shows the variation of heat flux from these two cylinders with a variation of Rayleigh number. As can be seen, heat flux increases with an increase in Rayleigh number for both grooved and plain cylinders, but the increase in the grooved cylinder is more than the plain cylinder due increase in heat transfer area in the grooved cylinder. As calculated, the plain cylinder has an area of 785mm^2 and the grooved cylinder has an area of 933mm^2 which a 20 % increase in the area can be observed.

Fig. 10 shows the variation of the Nusselt number with a variation of the Rayleigh number for both plain and grooved cylinders. It can be observed that grooving the cylinder causes an increase in the Nusselt number for all Rayleigh numbers calculated in this figure. The reason for this increase can be justified by observing Figs. 11 and 12. As shown in Figs. 11 and 12, the use of grooves on the cylinder increases the local velocity by approximately-two times, and also thermal diffusion into the fluid is better.

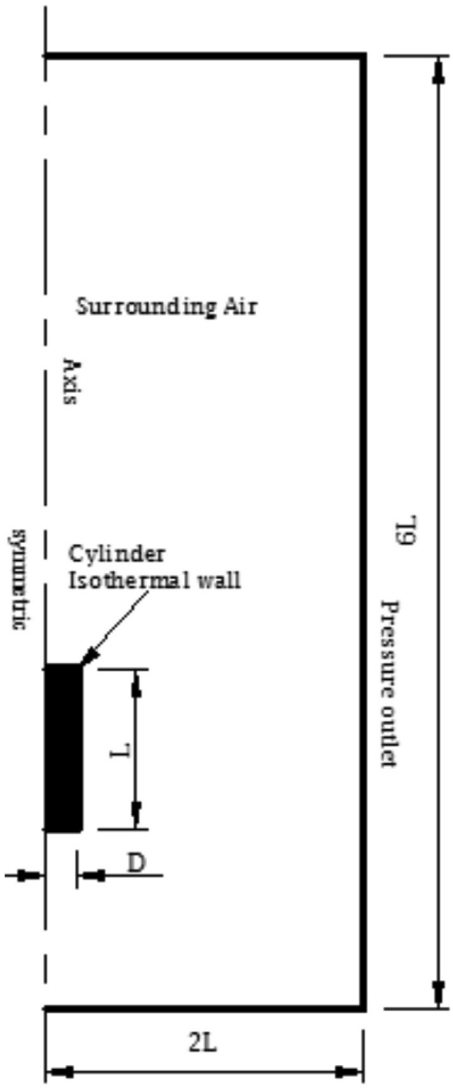


Fig. 7 Validation geometry.

Table 2 Comparison of the results of the present study with the results of Churchill [48] and Eckert [49].

Ra Number	Nu number of the present study	Nu number of Churchill	Nu number of Eckert
10^4	6.8	7.4	8
10^5	11.4	11.1	12.6
10^6	20	18.5	20.8
5×10^6	30.8	27.7	30
10^7	37.2	33.2	35.3
5×10^7	54.9	51.9	51.7
10^8	63	63.4	61.1

Fig. 13 shows the enhancement of the Nusselt number using a grooved cylinder relative to the plain cylinder for various Rayleigh numbers. As can be seen for this model of grooving enhancement of Nusselt number was in the range of 5 % and increases with an increase in Rayleigh number.

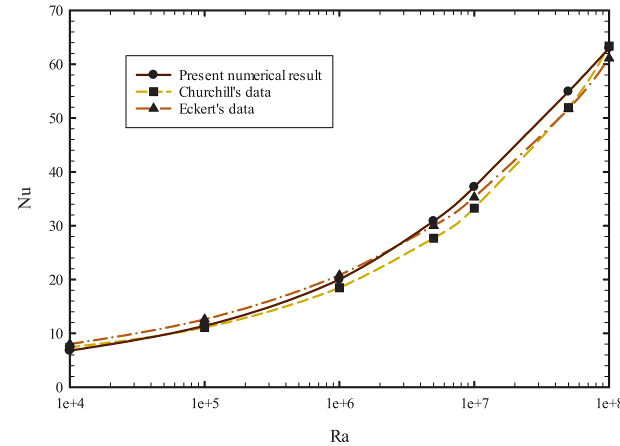


Fig. 8 Comparison of the present results with the studies of Churchill [48] and Eckert [49].

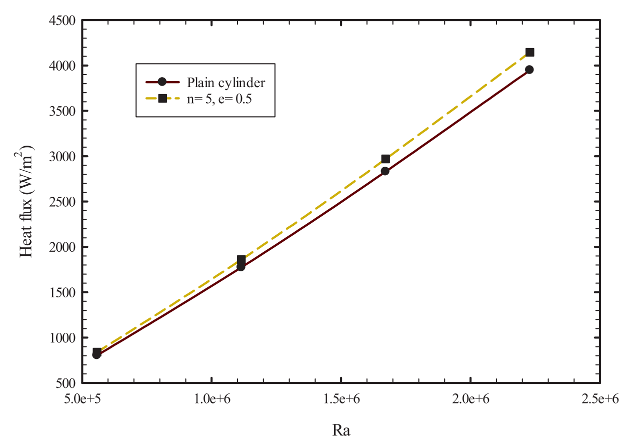


Fig. 9 Variation of heat flux with vs Rayleigh number for grooved and plain cylinders.

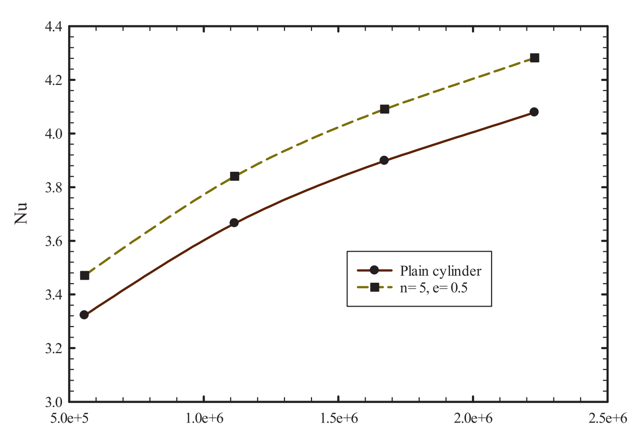


Fig. 10 The effect of groove on Nu number at different Rayleigh number.

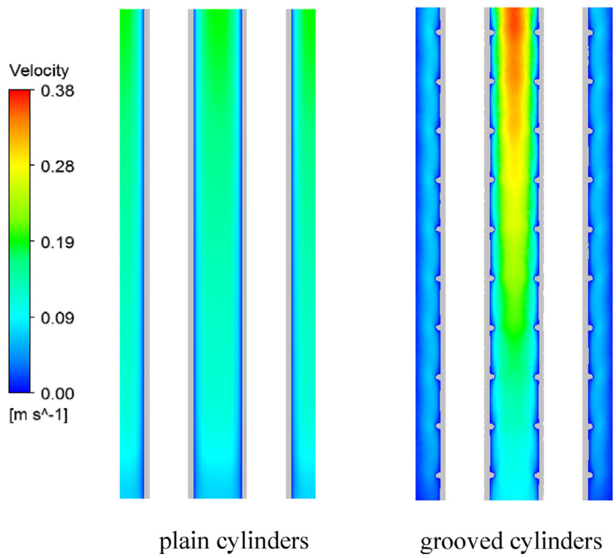


Fig. 11 Velocity distribution around the plain and grooved cylinders.

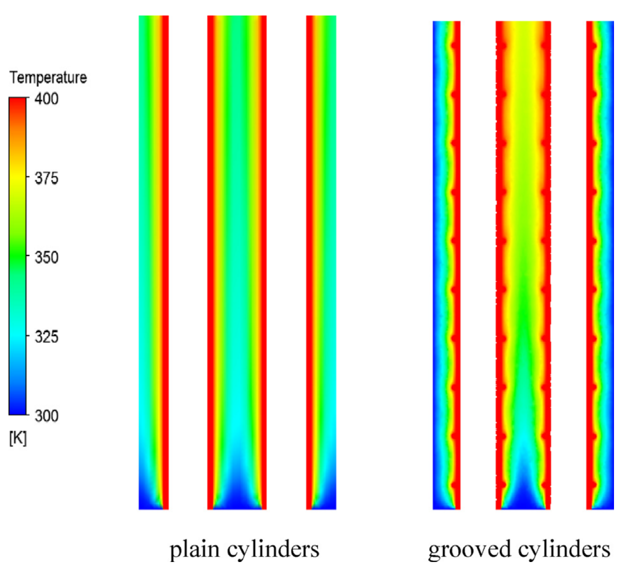


Fig. 12 Temperature distribution around the plain and grooved cylinders.

3.1. Effect of the number of grooves

In this section, the analysis of several grooves on the Nusselt number was performed. Three different numbers of grooves, 3, 5, and 7 were analyzed with a groove thickness of 0.5*mm* for different cylinder temperatures of 350 K to 500 K. Fig. 14 shows the variation Nusselt number with a variation of several grooves for each model. As can be seen, an increase in the number of grooves causes a gradual increase in the Nusselt number for the models. During the movement, the fluid creates secondary currents due to buoyancy by hitting the grooved surfaces, and this behavior can cause more collisions

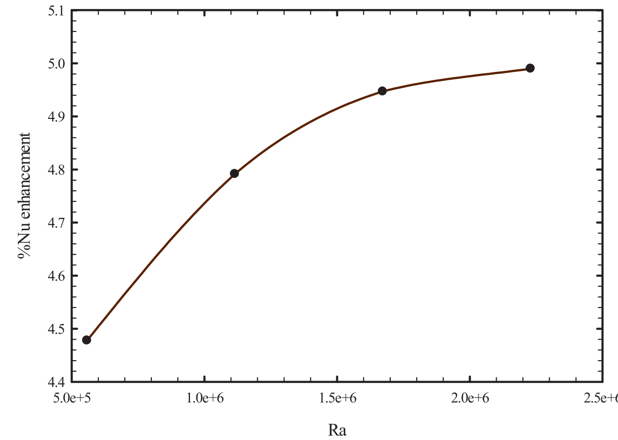


Fig. 13 Percentage of enhancement of Nusselt number for grooved cylinder relative to the plain cylinder.

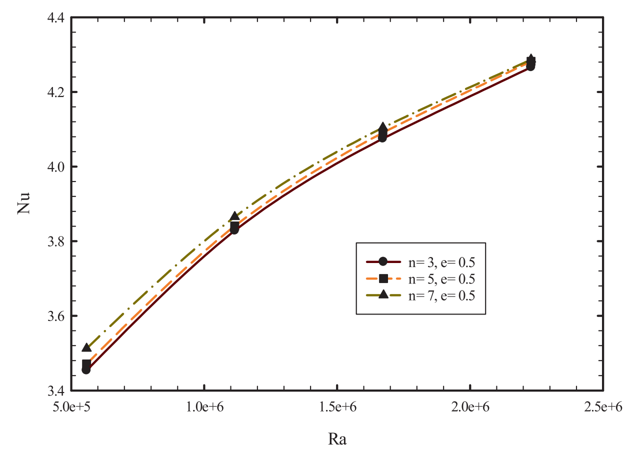


Fig. 14 Variation of Nusselt number vs Rayleigh number at different groove numbers.

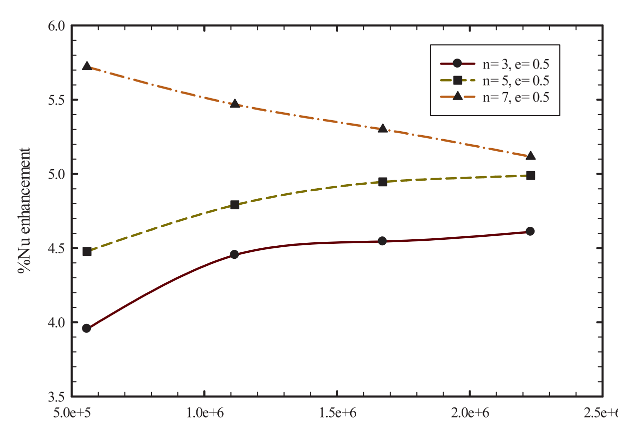


Fig. 15 Percentage enhancement of Nusselt number vs Rayleigh number at different groove numbers.

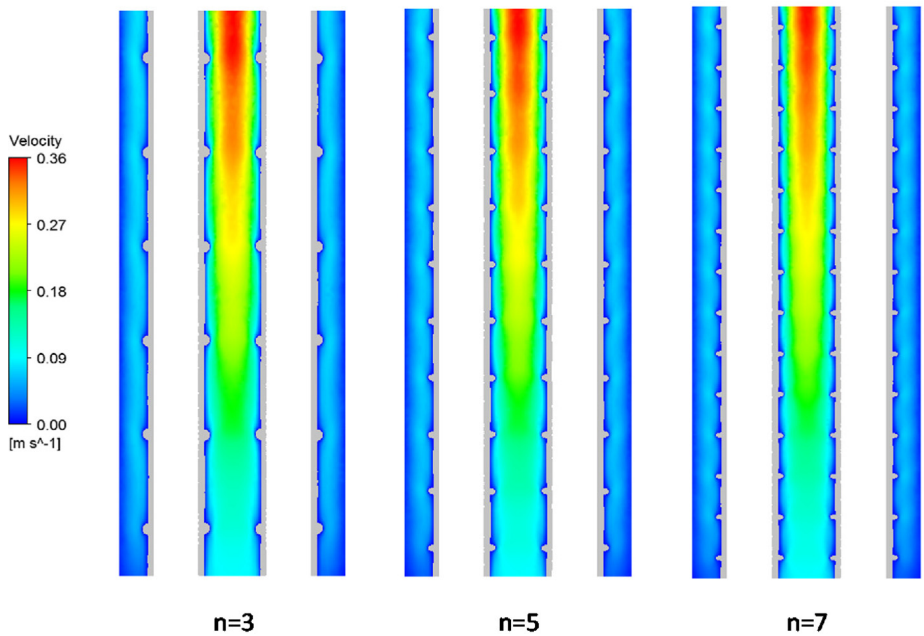


Fig. 16 The effect of groove number on velocity distribution around the grooved cylinders for $T_s = 400K$ and..$e = 0.5mm$

with hot surfaces. On the other hand, as the numbers of the groove increase, the mixing rate of the passing fluid and the fluid close to the hot wall will intensify, which will eventually increase the Nusselt number. By increasing the Rayleigh number, the enhancement of the buoyancy force will create the greatest amount of heat transfer. For this reason, the graph of Fig. 14 will have an upward trend with the increase of the Rayleigh number.

Fig. 15 shows the enhancement percentage of three models with various numbers of grooves relative to the plain cylinder. The plots shows that an increase in Rayleigh number as well as an increase in several grooves increases enhancement factor. This figure states that with the increase of the dimensions of the groove, the creation of secondary flow will be accompanied by large vortices. As the Rayleigh number increases, after the flow passes through the groove, the eddies created due to the stronger density gradients cannot contact the hot surface again and move towards the central areas of the channel, limiting the increase in heat transfer. In fact, increasing the number of grooves by 5 to limit the investigated Rayleigh number can increase the heat transfer. A further increase in the number of grooves decreases the thermal efficiency.

The slight improvement in Nusselt number due to the increase in the number of grooves can be interpreted by observing Figs. 16 and 17. As shown in these figures, as the number of grooves increased, the maximum velocity did not increase (It means the velocity of the center line of the flow), but the high- velocity area increased slightly; therefore, accord-

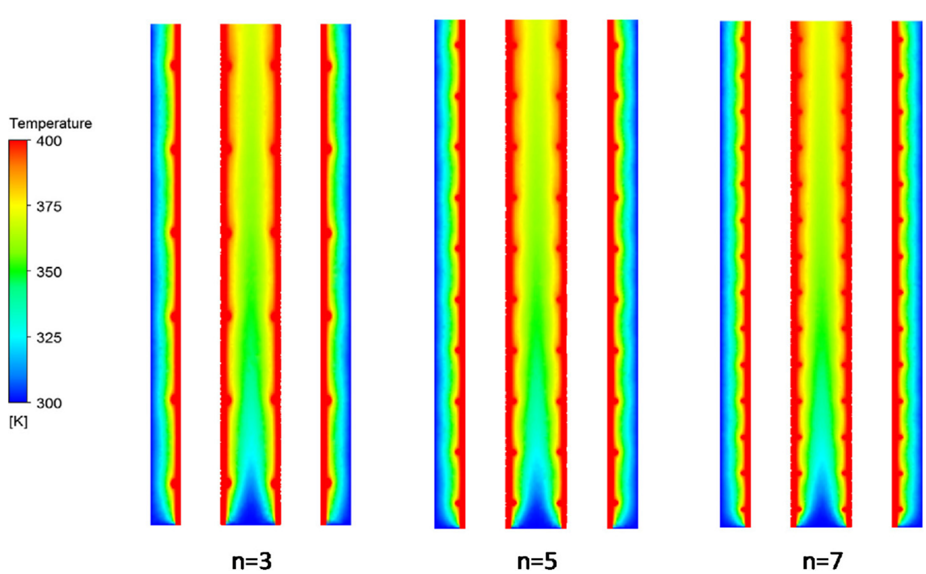


Fig. 17 The effect of groove number on temperature distribution around the grooved cylinders for $T_s = 400K$ and..$e = 0.5mm$

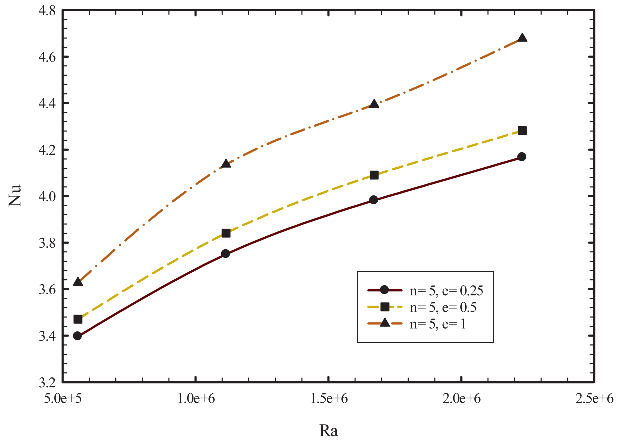


Fig. 18 Variation of Nusselt number vs Rayleigh number at different groove thicknesses.

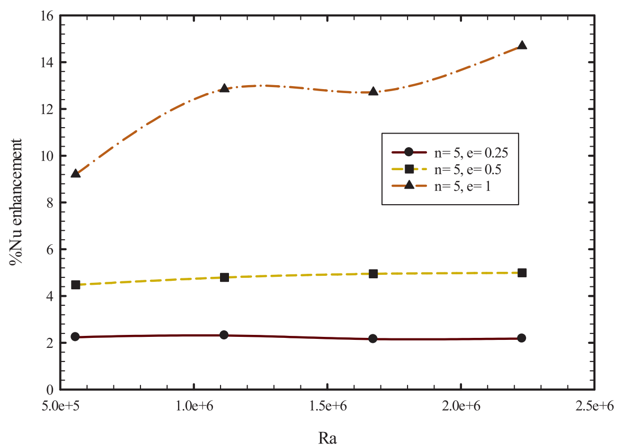


Fig. 19 Enhancement of Nusselt number vs Rayleigh number at different groove thicknesses.

ing to Fig. 17, in the outlet area, the local temperature of the passing fluid increases, and then the heat transfer can be improved slightly.

3.2. Effect of groove thickness

To calculate the effect of the thickness of the groove, 5 numbers of grooves were chosen with three thicknesses of 0.25, 0.5, and 1 mm for the grooved cylinders. Fig. 18 shows the variation of the Nusselt number with the Rayleigh number for three different groove sizes. As can be seen, an increase in the thickness of the groove results in an increase of the Nusselt number according to the corresponding Rayleigh number.

Fig. 19 shows the enhancement percentage of the Nusselt number with a variation of the Rayleigh number for three different groove sizes. As shown, with an increase in groove thickness, the enhancement percentage of the Nusselt number increases. Also, for a thickness of 0.25 and 0.5 mm increase in the Nusselt number was almost constant, but for a thickness of 1 mm, the enhancement percentage was increased monotonically with an increase in the Rayleigh number.

According to the contours of temperature and velocity in Figs. 20 and 21, increasing the thickness of the grooves has a significant effect on increasing the local velocity of the passing fluid and improving the heat transfer rate from the surface of the cylinders to the passing fluid.

4. conclusions

In this study, numerical analysis of natural convective from grooved cylinders was performed for three different numbers of grooves and three different groove thicknesses in four different cylinder temperatures and results were discussed and following conclusions were deduced:

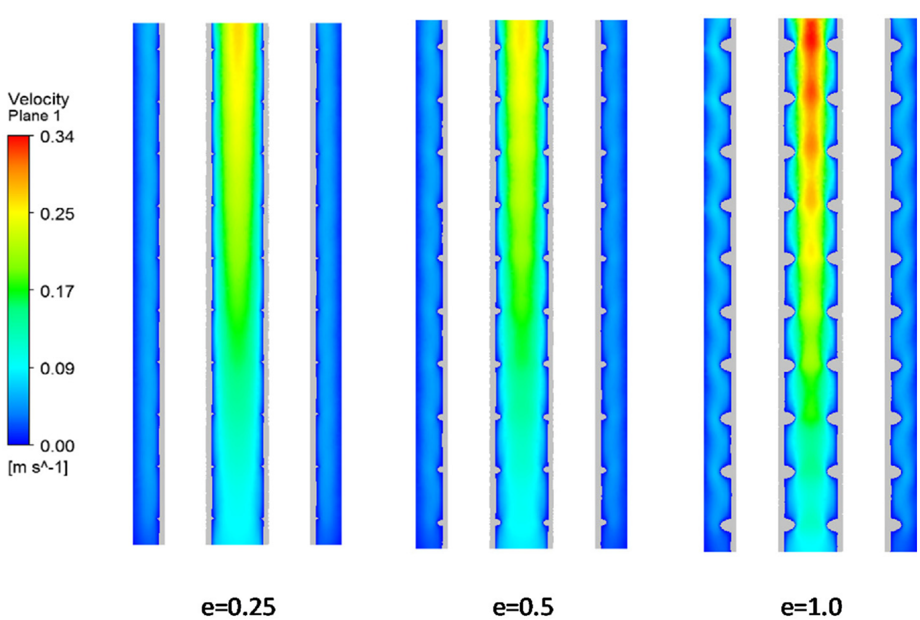


Fig. 20 The effect of groove thickness on velocity distribution around the grooved cylinders for $T_s = 400K$ and..$n = 5$

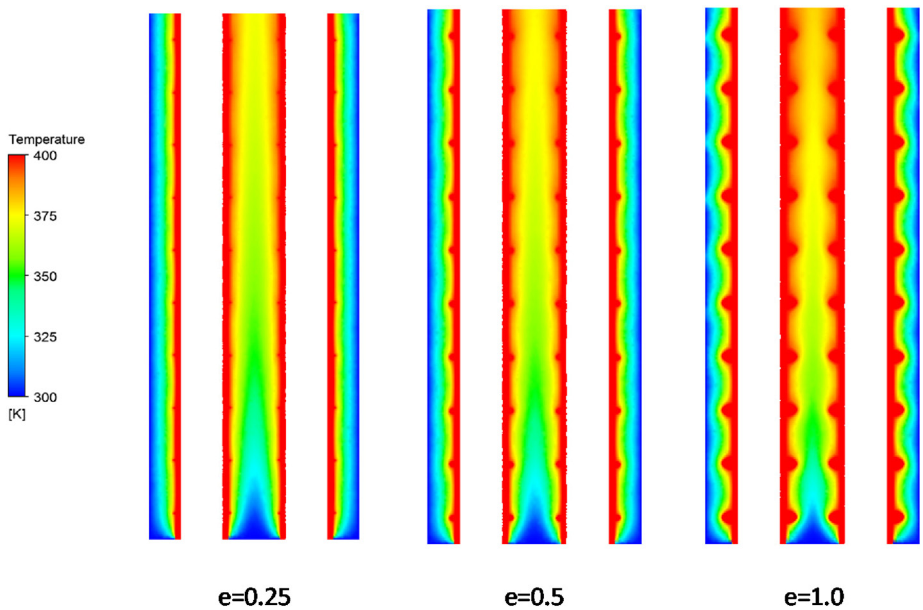


Fig. 21 The effect of groove thickness on temperature distribution around the grooved cylinders for $T_s = 400K$ and..$n = 5$

- The use of grooved cylinders instead of plain cylinders will cause an increase in heat transfer rate from the surface of the cylinder due to the increase in the area of heat transfer and formation of rotational flow
- The use of grooved cylinders can increase the heat transfer rate up to 14 % relative to plain cylinders.
- An increase in Rayleigh number will result in an increase in the Nusselt number as well as an increase in the percentage of Nusselt number enhancement.
- An increase in several grooves and groove thickness results in an increase of the Nusselt number and enhancement of the Nusselt number for a constant Rayleigh number. In this matter, the thickness of the groove is more effective than the number of the groove.
- By increasing the Rayleigh number, increasing the number of grooves up to 5 can only increase the heat transfer.
- The excessive increase of the groove causes the creation of large secondary currents and the separation of the current from the heated surface.

Declaration of Competing Interest

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 paper.

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