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Natural Convection in a Vertical Annulus Enclosure with Longitudinal Fins

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Abstract. Heat transfer by natural convection can be found in many thermal engineering applications. In order to improve the heat transfer, fins are used to increase the contact surface and thus the heat transfer rate. In this project, natural convection heat transfer performance in the vertical annulus enclosure was studied numerically. Software Ansys fluent was used to solve the governing equations to arrive to velocity and temperature distribution. A grid test was performed, and the model was validated with previous experimental and numerical studies. The results showed that the fins have a significant effect on the heat transfer enhancement compared to the model without fins. Afterward, a parametric study was performed to study the effects of fins number, length, thickness, and shape. Increasing fins number increases the heat transfer rate by 46%. The effect of the number of fins on the heat transfer rate corresponds with the results in the literature. Increasing fins length increases the heat transfer rate by 27%. The fin thickness has a little effect on the amount of the convection heat transfer. It was also found that square fins have the best results over circular and triangular fins. Regarding the Rayleigh number effect, increasing the Rayleigh number increases the heat transfer rate, which corresponds with the results reported in the literature.

Keywords: Keywords: Natural convection; Vertical annulus; Longitudinal fins

1. Introduction

There is an increase in energy demand worldwide for industrial use. The rapid development of electronic devices for personal and industrial use and thermochemical processes led to the rise of technical and financial challenges. One of these challenges is the ability to enhance the heat transfer in these applications to reduce energy consumption and cost.

Sometimes the heat is not transferred adequately which can cause serious problems in the thermal systems. For example, when the heat generated from nuclear reactor rods is needed to be transferred regularly to maintain the temperature under control, therefore, enhancing the heat transfer is needed. Bergman et al. [1].

The heat transfer enhancement techniques are mainly classified into two groups. The first one is the active technique where an external source of power is required to maintain the heat transfer enhancement such as mechanical aids, surface vibration, fluid vibration, electrostatic fields, injection, suction, spray, and jet impingement. The second one is the passive technique where the external power source is not needed such as treated surfaces, rough surfaces, smooth surfaces, use of additives such as nanofluids, and extended surfaces such as fins, Rohsenow et al. [2].

The fins are used to enlarge the surface area, hence increasing the contact area resulting in increasing the heat transfer rate. It's important to mention that the effect of fins on the natural convection depends on several parameters. Some of the parameters are fins number, length, thickness, material, and shape, Bergman et al. [1].

2. Literature Review

In this study, we conduct a study on natural convection in annulus due to its important application such as in heat exchangers, chemical reactors, and nuclear reactors. Therefore the literature review will focus on natural convection in annulus shape.

2.1. Natural convection in the annulus

2.1.1. Horizontal annulus

Many studies were conducted on the natural convection in the horizontal annulus. For example, Nada [3] experimentally studied the effect of the annulus gap widths and the annulus inclination angle on the natural convection in an annulus and the results showed the heat transfer increases with the increase of the annulus gap width and slightly decreases with the increase of the annulus inclination angle. Yeh [4] numerically studied the natural convection in a horizontal annulus with open ends for the adiabatic outer cylinder surface case and isothermal outer cylinder surface case. He found that the maximum inner cylinder surface temperature reported was at the inner cylinder top and that the inner surface cylinder temperature decreased toward the outlet plan in the adiabatic case and remain relatively constant in the isothermal case. Kumari and Nath [5] numerically studied the unsteady natural convection in an annulus filled with a non-Darcy porous medium. The results showed that the annulus completely filled with a porous medium has the best insulating effectiveness. He also found that most of the natural convection in the horizontal annulus is confined at the top and bottom places, therefore, only these places should be insulated. Yoo [6] numerically studied the bifurcation and dual solutions for natural convection in the horizontal annulus. He found that when the value of the Prandtl number is ranged from about 0.3 to 1.0, two different fluid motion shape types were noticed. The first one is the crescent-shaped eddy in the annulus top. The second one is two counter-rotating eddies in a half annulus. Wang et al. [7] numerically studied the Magnetohydrodynamics effect on the natural convection in a horizontal annulus in terms of the uniform axial magnetic field impacts. It was noticed that a spiral flow arises and the symmetry breaking occurs under a weak magnetic field when Hartmann's number is less than ten. Yang and Kong [8] numerically studied the natural convection in a horizontal annulus. The smoothed particle hydrodynamics method was used and the results showed that the flow stability is increased by increasing the Rayleigh number. Belabid and Allali [9] numerically studied the temperature modulation effect on the natural convection in a horizontal porous annulus, the study is time-dependent and the time-dependent modulation is periodic with frequency and amplitude. It was found that the system became stable at moderate frequency values.

2.1.2. Vertical annulus

Regarding the natural convection in the vertical annulus, several studies were conducted in that area. For instance, Reddy and Narasimham [10] numerically studied the natural convection in a vertical annulus driven by a central heat generating rod. The results showed that with the increase of the Grashof number, the average Nusselt numbers of the inner and outer boundaries increased. Sankaret at.[11] numerically studied the natural convection in a vertical porous annulus with discrete heating for the inner wall, the results indicate that the heat transfer increases with the increase in the modified Rayleigh number, and decreases with the increase in the heater length. Chen et al. [12] studied the effect of inner-wall motion on the linear stability of natural convection in a tall vertical annulus using linear theory, the results reveal that the inner wall motion has a significant effect on the stability characteristics. Sankar et al. [13] numerically studied the effect of the axial and radial magnetic field on the natural convection in a vertical annulus. The results indicated that for the shallow cavity, the flow and heat transfer in suppressed is more by the axial magnetic field, while in the tall cavity the radial is more effective. Husain and Siddiqui [14] experimentally and numerically studied the transient natural convection of water in a high aspect ratio narrow vertical annulus. The results showed that the transient period increases gradually as the annulus height increases while decreasing as the heat input increases. Afrand et al. [15] numerically studied the natural convection in a vertical annulus filled with molten gallium in the presence of the horizontal magnetic field. The results revealed that the magnetic field is generating the Lorentz force in the opposite direction of the buoyancy force. Abhilash and Lal [16] numerically studied natural convection in a narrow vertical annulus which is closed at the top and opened at the bottom. They noticed two different modes of flow which are the radial mode and the circumferential mode. Also, the Rayleigh number has an important effect on these modes' stability. Wang et al. [17] numerically studied the natural convection in a vertical annulus with an inner wall covered by a porous layer. The results showed that wrapping with a porous layer is a good tool for both increasing or reducing heat transfer. Char and Lee [18] numerically studied the natural convection in a porous medium saturated with cold water under density inversion. The results revealed that the inversion parameter and the radius ratio have an important effect on the flow and heat transfer, where the radius ratio is the difference between the inner and outer radius divided by the inner radius. Jha et al. [19] analytically studied the effects of suction/injection and wall surface curvature on natural convection flow in a vertical micro-porous annulus. It is found that when the suction/injection on the cylinder walls is increased, the velocity and temperature are increased.

2.2. Natural convection in the annulus with fins

2.2.1. Horizontal annulus

The effect of the fins on the natural convection in the horizontal annulus was studied from many aspects. For example, Rahnama and Farhadi [20] numerically studied the radial fins effect on the turbulent natural convection in a horizontal annulus. It was found that the fins arrangement has no significant effect on the heat transfer while its effect on the flow and temperature fields is noticeable in the case of four fins arrangement. Kiwan and Zeitoun [21] numerically studied the laminar natural convection in a horizontal annulus with porous fins. The results revealed that the porous fins provided higher heat transfer rates than solid fins for similar configurationsNada and Said [22] numerically studied the effect of the longitudinal and annular fins on the natural convection in a horizontal annulus. The results showed that longitudinal fins enhance heat transfer better than annular fins. Alshahrani and Zeitoun [23] numerically studied the fin's inclination angle effect on the natural conduction in the horizontal annulus. It was found that the inclination angle has a weak effect on the thermal conductivity ratio, where the effective thermal conductivity of the fluid is defined as the thermal conductivity of the fluid at which stagnant fluid in the annulus can give the same rate of heat transfer, and the effective thermal conductivity ratio is the effective thermal conductivity of the fluid divided by the thermal conductivity of the same fluid. Shadlaghani et al.[24] numerically studied the natural convection in a horizontal concentric and eccentric annulus with serrated fins. The results showed that the heat transferred is better in the eccentric annulus case compared to the concentric annulus one.

2.2.2. Vertical annulus

A few studies were performed to study the effect of fins on the natural convection in the vertical annulus. Ramezanpour and Hosseini [25] experimentally studied the natural convection in a vertical annulus with a helical fin and the results show that the helical fin enhances the heat transfer. Solomon and Velraj [26] experimentally studied the natural convection in a vertical annulus with longitudinal fins during the phase change material, where the phase change material is a substance that absorbs or releases large amounts of latent heat when they go through a change in their physical state, the results of the study showed that with the fins used an enhancement occurs in the heat transfer. Kumar [27] numerically studied the natural convection in a vertical annulus with longitudinal fins for various parameters. The results show that the heat transfer rate increases with the increase of the fin width and radius ratio and decreases as the aspect ratio increases beyond one, where the aspect ratio is the annulus height divided by the annulus gap width.

The literature review revealed that adequate studies on natural convection in the annulus were done, but when it's come to natural convection in the annulus with fins, especially on the vertical annulus, few studies were done. Therefore, more studies on vertical annulus with fins are needed.

3. Study Goal

This project will study parameters: fins number, length, thickness, and shape. Their effect on heat transfer rate, temperature distribution, and velocity profile will be studied and presented. The model of a 3-D vertical annulus enclosure was established. It was studied numerically using Ansys fluent.

4. Problem Formulation

4.1 Model Specifications

The mathematical model is a 3-D vertical annulus enclosure with a hot inner wall and cold outer wall, Schematic of the vertical annulus is shown in Fig.1. Boundary conditions of the top and bottom wall are insulated, the inner vertical wall has a hot temperature T_H of 401.65 ℃ to 416.5 ℃ and the outer vertical wall has a cold temperature T_C of 383.5 ℃ to 398.35 ℃. Temperatures of vertical walls were chosen so that R_L values range from 10^4 to 10^5.

Radiation heat transfer becomes important at high temperatures (above 1000 K) [28] and our present study did not reach this limit. Therefore, we can neglect the radiation effect.

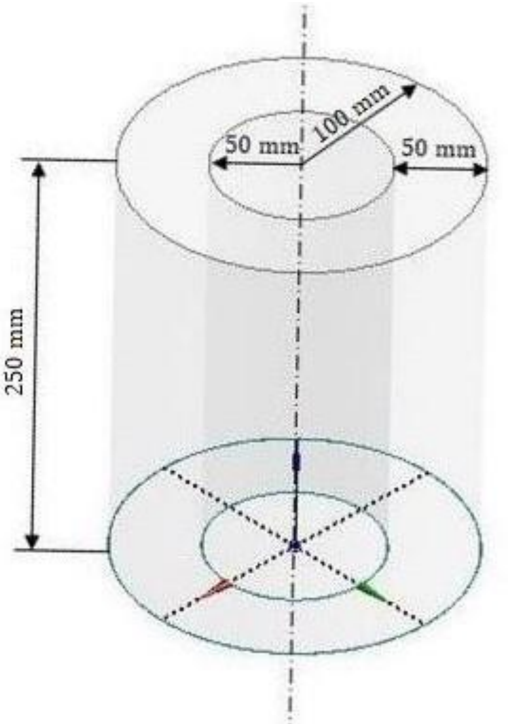

Fig.1 Schematic of the vertical annulus

Buoyancy force was modeled using Boussinesq approximation and taking gravity $g=-9.81$ m/s^2 in the negative vertical direction. In this project we assumed the following:

1- The variation of the density was taken into account only in the body forces (Boussinesq approximation).

2- Steady-state.

3- No heat generated.

4- No radiation in the enclosure between surface nor fluid particles.

4.2 Ansys Fluent

Ansys Fluent is industry-leading fluid simulation software used to predict fluid flow, heat and mass transfer, chemical reactions, and other related phenomena. Known for delivering the most accurate solutions in the industry, Ansys fluent is based on the finite volume method and works by the following steps:

1. The domain or the geometry in this study is discretized into a finite set of control volumes using a computational domain also known as mesh.

2. The general conservation equations such as mass, momentum, and energy are solved on this set of control volumes.

3. The partial differential equations are discretized into algebraic equations.

4. All algebraic equations are then solved numerically to obtain the solution.

It is important that the computational domain (mesh) is accurate enough to obtain accurate results therefore we decrease the control volumes (mesh size) and run the simulation and record the output results, we repeat this process until the output results become almost the same, this process called the mesh test also known as the grid test.

For the present study, we used Ansys fluent to solve the governing equation to obtain the effective thermal conductivity ratio, temperature distribution, and velocity profile. This is done by iterations through the discretization of the model. The governing equations were solved sequentially using a pressure-based solver. Momentum and energy equations were solved using a second-order implicit scheme. The governing equations for mass, momentum, and energy are defined as follows, Bergman et al. [1]:

$$\boldsymbol{\nabla \cdot V = 0} \quad (1)$$

$$\boldsymbol{\rho(V \cdot \nabla)V = -\nabla P + \mu\nabla^2 V + \rho g} \quad (2)$$

$$\boldsymbol{(V \cdot \nabla)T = \alpha\nabla^2 T} \quad (3)$$

The Rayleigh number is a dimensionless number associated with the buoyancy-driven flow and can be defined as the product of the Grashof number, which describes the relationship between the ratio of buoyancy forces to viscous forces, and the Prandtl number, which describes the relationship between the ratio of the momentum and thermal diffusivities.

Hence it may also be viewed as the ratio of buoyancy and viscosity forces multiplied by the ratio of momentum and thermal diffusivities, Bergman et al. [1].

In the present study, the Rayleigh number is dependent on the temperature difference between the hot and cold sides of the annulus which means that the other parameters such as r_i and r_o are fixed at 50 mm and 100 mm respectively while the hot inner side temperature and the cold outer side temperature are varied.

The dimensionless groups of the Rayleigh number and Prandtl number are defined as follows, Bergman et al. [1]:

$$Ra_L = \frac{g\beta(T_h - T_c)(r_o - r_i)^3}{\alpha\nu} \quad (4)$$

$$Pr = \frac{\nu}{\alpha} \quad (5)$$

In natural convection in an annulus, the heat transfer coefficient depends on the inner and outer radius of the annulus. Therefore, the free convection heat transfer in the annulus is normally calculated in terms of the effective thermal conductivity of the fluid in the annulus which is defined as the thermal conductivity of the fluid at which stagnant fluid in the annulus can give the same rate of heat transfer. In order to use the effective thermal conductivity in a dimensionless group, we divide it by the thermal conductivity to become the effective thermal conductivity ratio which is considered an indicator of the thermal conductivity enhancement ratio due to natural convection. Hence, when the effective thermal conductivity ratio increases, this means the natural convection is enhanced.

The ANSYS Fluent has many correlations which use to solve the problems. Based on the inputs and the geometry of the problem, the ANSYS Fluent selects a suitable correlation to obtain the results. In our problem, we input the following parameters: the outer diameter (D_o) and inner diameter (D_i) of the annulus, the thermal conductivity of the fluid (K), the temperature difference between the hot inner surface of the annulus (T_h) and the cold outer surface of the annulus (T_c). The numerical method used in ANSYS Fluent to obtain the heat transfer rate is the finite volume method. The ANSYS Fluent will compute the heat transfer rate and we use it in Eq. 3.6 to calculate the effective thermal conductivity ratio (K_{eff}/K).

The equation of the effective thermal conductivity ratio is given as follows, Nada and Said [22]:

$$K_{eff}/K = \frac{q' \ln\frac{Do}{Di}}{2\pi K\,(T_h - T_c)} \quad (6)$$

4.3 Fins Specifications And Parametric Study

Longitudinal fins are added to the inner hot vertical wall inside the 3D annulus enclosure. The fins were made of copper and we used copper because it is easier to maintain than aluminum fins.

A parametric study was conducted to study the effect of four parameters on the heat transfer rate performance. These four parameters are the number of fins, fins thickness, fins length, and fins shape. When we study these parameters, only one parameter was changed while the others remain constant, for example when we study the fins' shape we change the fins' shape while the fins' thickness, number, and length remain fixed. All cases were compared to the model without fins.

The study was done for the number of fins 4, 8, and 16. Fins' length was changed from 5 mm to 15 mm. Fins thickness of 2 mm, 7 mm, and 15 mm were used, and different shapes of square, circular, and triangular shapes were used. The four studied parameters are shown in Fig. 2 to 5 and their dimensions are shown in Table 1.

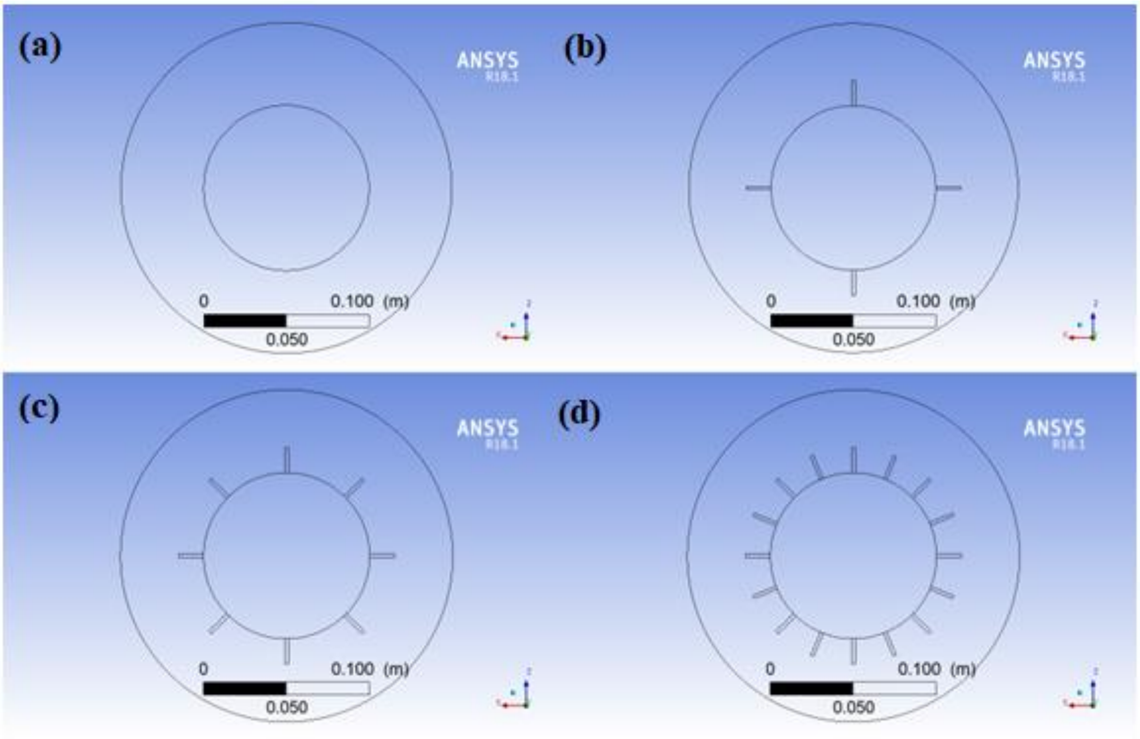

Fig. 2 Vertical annulus with (a) no fins, (b) four fins, (c) eighth fins, and (d) sixteenth fins.

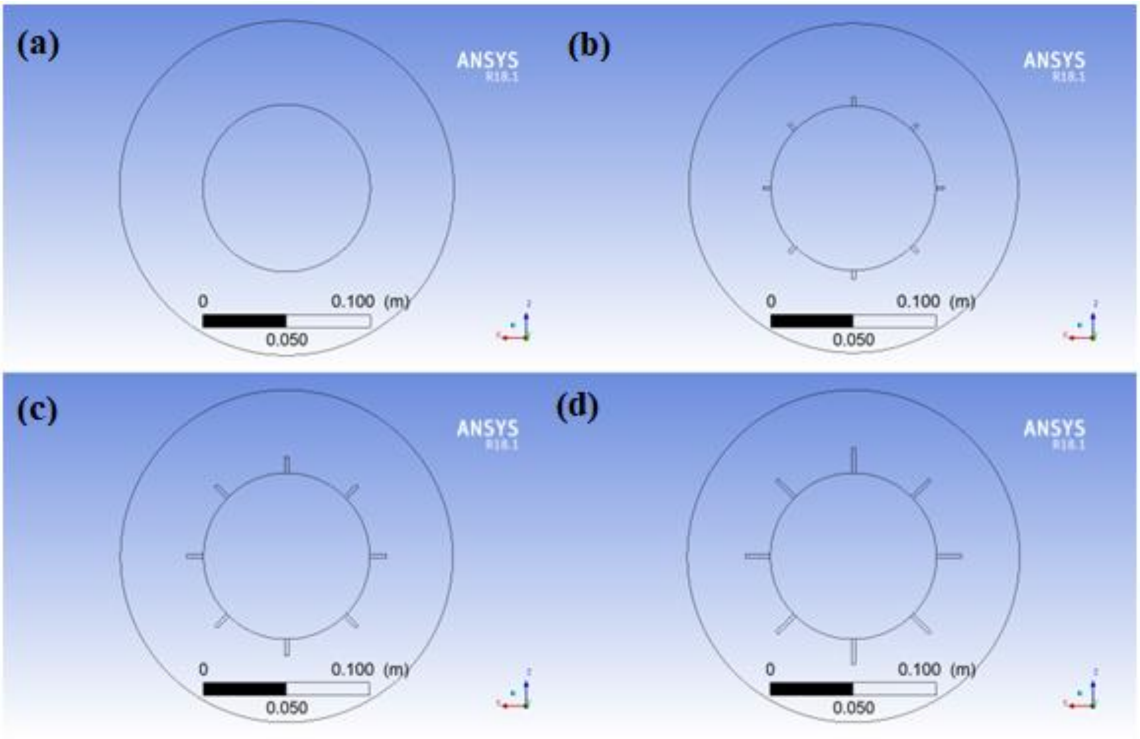

Fig. 3 Vertical annulus with (a) no fins, (b) L=5 mm, (c) L=10 mm, and (d) L=15 mm.

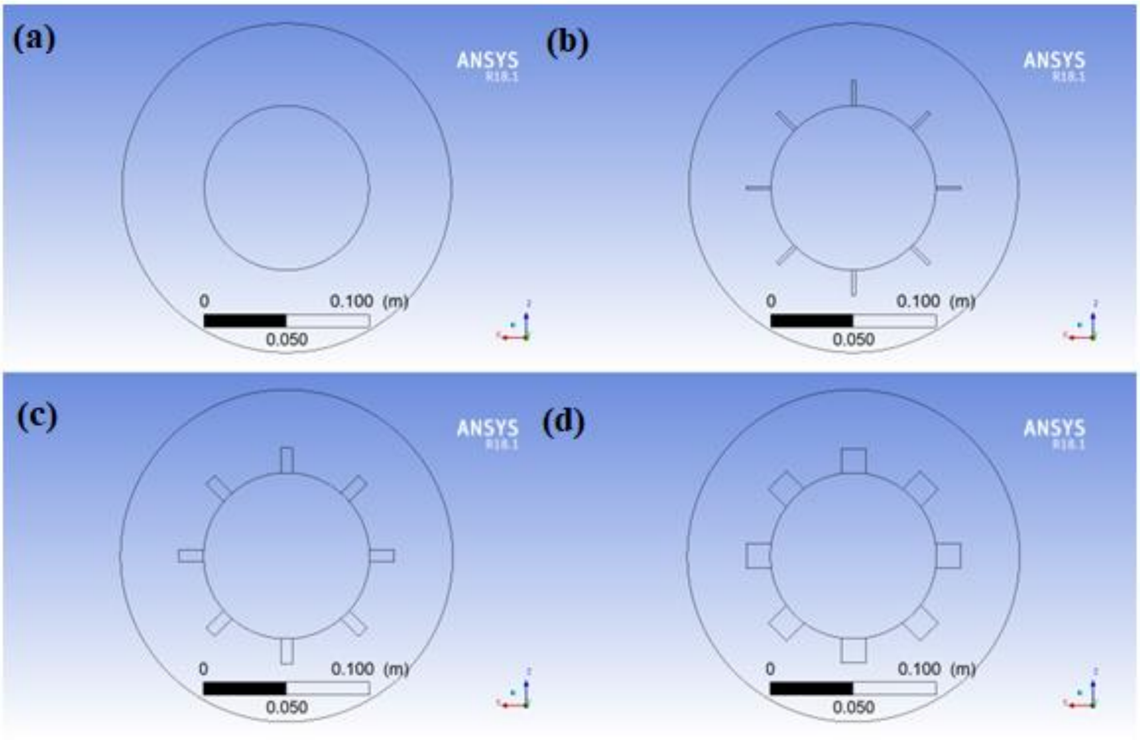

Fig. 4 Vertical annulus with (a) no fins, (b) W=2 mm, (c) W=7mm, and (d) W=15 mm.

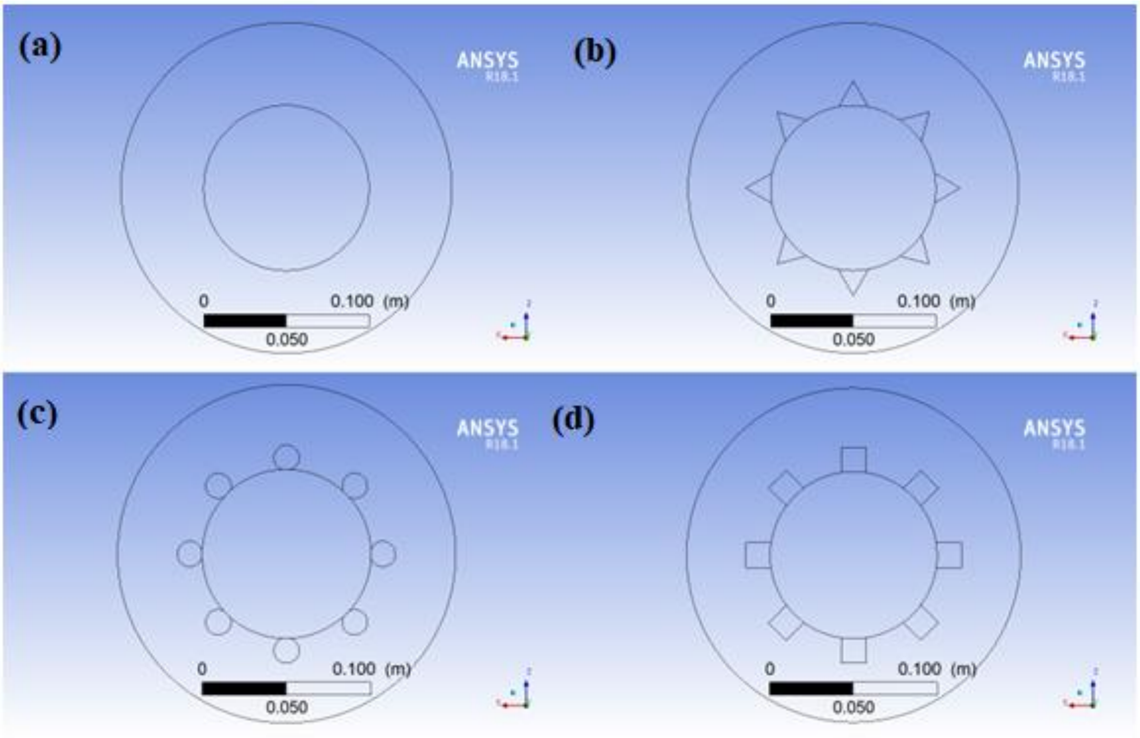

Fig. 5 Vertical annulus with (a) no fins, (b) triangular fins, (c) circular fins, and (d) square fins.

Table1 Fixed and changed values of the four parameters

Parameter	Original Value	Changed values
number of fins	8	4,8,16
fins length	15 mm	5 mm, 10 mm, 15 mm
fins thickness	2 mm	2 mm, 7mm, 15 mm
fins shape	rectangular	circular, triangular, square

5. Grid Test and Code Validation

5.1. Grid Test

One of the Ansys fluent steps to obtain the solution is to divide the geometry into discrete control volumes or elements. When the mesh size was reduced, the number of elements increased. and the output of the results is more accurate. The effective thermal conductivity ratio (K_{eff}/K) was used as the output results to test the grid accuracy.

The grid test was done at $Ra_L=10^5$ and started by increasing the number of elements from 194275 to 4054716 elements. Table 2 and Fig. 6 show the resulted K_{eff}/K for different numbers of elements. The number of elements Increased until the change in the effective thermal conductivity ratio became minimum. After the number of elements of 3,446,018, the effective thermal conductivity ratio value remains at 3.466 with less than 0.09 % error, hence, the number of elements of 3,446,018 is used in our current project.

Table 2 Effective thermal conductivity ratio for different numbers of elements

Number of elements	K_{eff}/K
194275	4.152
277962	3.544
541394	3.512
1032848	3.491
1421910	3.483
2005578	3.475
2586843	3.472
3009350	3.469
3446018	3.466
4054716	3.466

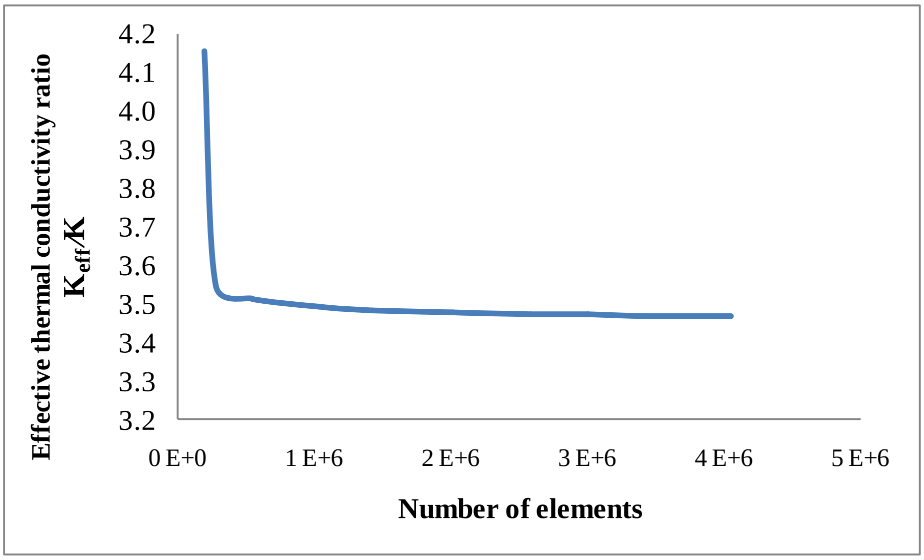

Fig. 6 Effective thermal conductivity ratio vs. number of elements

5.2. Code Validation

Due to the initial conditions and assumptions used in the vertical annulus studies, the validation was found easier with the horizontal annulus studies than with the vertical annulus studies. Therefore, the present study was validated by comparing its output results with four different horizontal annulus studies. Two of the four studies were performed experimentally [29] and [30] and the other two were performed numerically [21] and [23].

In order to compare our results to these four studies, our work was run under the same conditions used in those four studies. The present study results compared with the results of the four studies and showed a good agreement as shown in Fig. 7 We used the effective thermal conductivity ratio (K_e/K_f) to represent the enhancement of the natural convection in a dimensionless form and the modified Rayleigh number (Ra_m) to represent the temperature difference between the hot side and the cold side of the annulus in a dimensionless form.

The relationship between the K_e/K_f and Ra_m is a positive relationship. Hence, when the Ra_m increases, the K_e/K_f increases. We used the effective thermal conductivity ratio (K_e/K_f) vs. modified Rayleigh number (Ra_m) in the validation because it was used in the literature [21,23] Hence, we can compare our results to those in the literature.

Modified Rayleigh number (Ra_m) is given as:

$$Ra_m = Ra_i^{\frac{1}{4}}\,(0.1389(1-\frac{D_i}{D_o})+0.0927)\ln(\frac{D_o}{D_i}) \quad (7)$$

Where Rai is the Rayleigh number based on the inner diameter and given as:

$$Ra_i = \frac{g\beta(T_h-T_c)(D_i)^3}{\alpha\nu} \quad .(8)$$

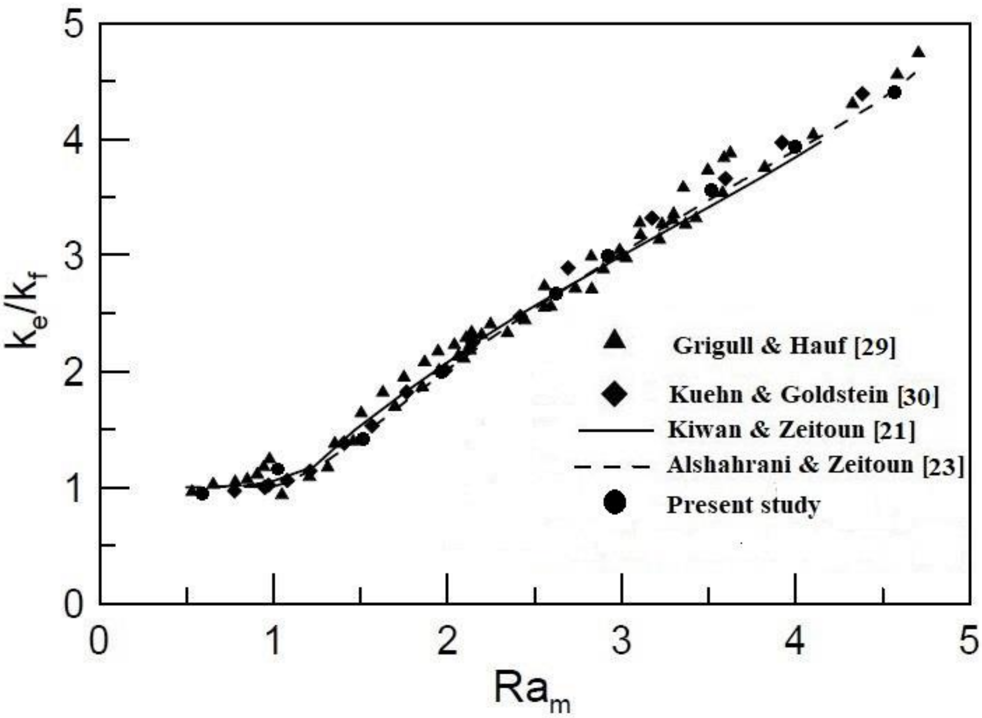

Fig. 7 Numerical model validation: K_e/K_f vs. Ra_m

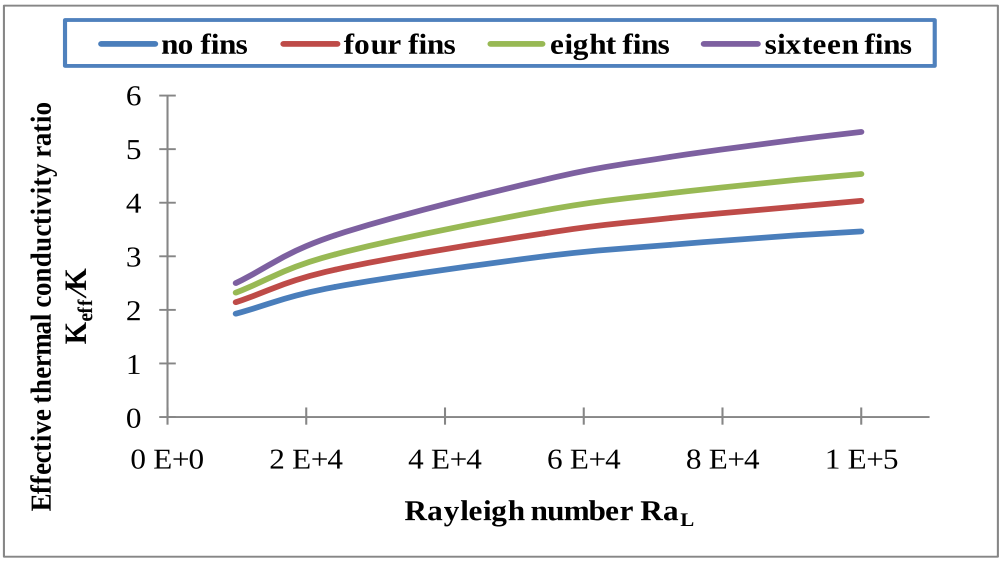

Fig. 8 Effective thermal conductivity ratio vs. Rayleigh number at different fins number

6. Results and Discussions

6.1. Effect of fins number

Fig. 8 shows the effect of fins number on heat transfer performance. When the fins number increases, the contact surface area of total fins exposed to the enclosed fluid increases, and thus, the heat transfer rate increases. When using four fins, the heat transfer is enhanced by 15% compared to no fins case. When using eight fins, the heat transfer is enhanced by 27% compared to no fins case. When using sixteen fins, the heat transfer is enhanced by 46% compared to no fins case.

The effect of fins number in these study correspond with the effect of fins number reported in Nada and Said study [22]. Despite their study being done on a horizontal annulus, the effect is the same which is the increase in natural convection with the increases of fins number which is the same effect in this study. The similarity of the results is because the principle of both studies is the same, which is the increase of fins number, causes an increase in the contact surface area of total fins exposed to the enclosed fluid hence, the rate of heat transfer increases.

The effect of the Rayleigh number on the heat transfer in Rahnama and Farhadi's study [20] shows that the heat transfer increases with the Rayleigh number increases. Although their study was done on turbulent natural convection, it has the same effect compared to our study. Also, Nada and Said's study [22] where their study was done on laminar natural convection in a horizontal annulus. It was found that the natural convection increases with the increase of Rayleigh number which is the same effect in this study.

It can be observed that the Rayleigh number has the same effect on the heat transfer rate in both laminar natural convection and turbulent natural convection, also the Rayleigh number has the same effect on the heat transfer rate in both the horizontal annulus and the vertical annulus.

Fig. 9 shows the effect of fins number on the temperature profile along the radial direction from the inner hot surface to the outer cold surface. It shows that the temperature profile generally increases with the increase of fins number.

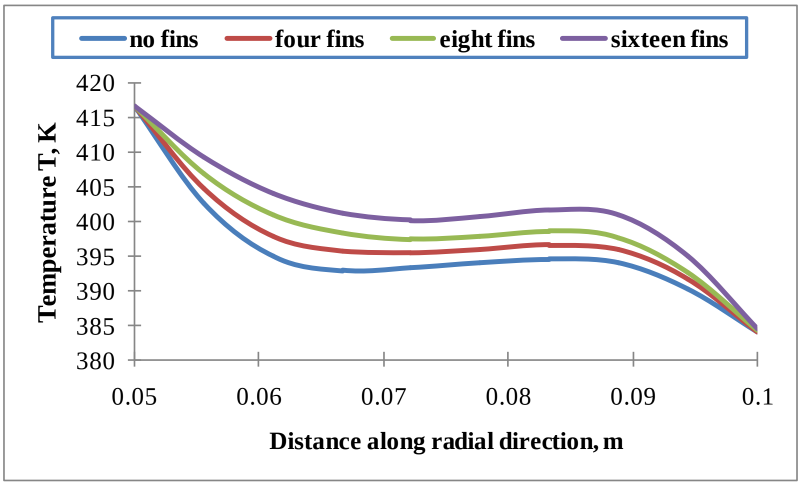

Fig. 9 Temperature profile along the radial direction at the vertical midpoint from the inner hot surface to outer cold surface at different fins number.

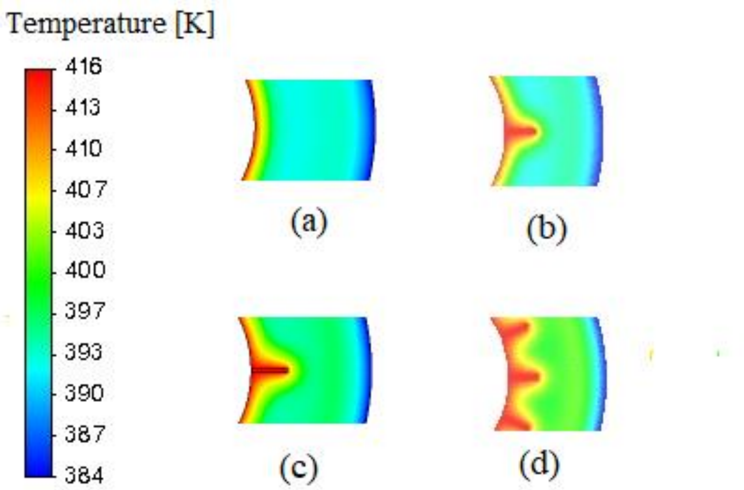

Fig.10 Cross-sectional horizontal view of temperature contour for (a) no fins, (b) four fins, (c) eight fins, and (d) sixteen fins.

Fig. 10 shows a cross-sectional view of the temperature contour plot with the different numbers of fins. This section was taken at the horizontal midsection. It is seen that with more fins, the hot temperature contour lines spread out away from the inner hot surface. This means a higher temperature gradient $(\frac{\partial T}{\partial x})$ and, thus, higher heat transfer.

Fig. 11 shows a vertical side view of the temperature contour plot with the different numbers of fins. This section was taken at the vertical midsection. It can be seen that with more fins, the fluid layers at the inner hot surface (left) become higher in temperature, which means higher heat transfer.

To understand the number of fins' effect on the velocity we should first discuss how the fluid velocity moves inside the annulus without fins. When the vertical inner hot surface (inner side) heats up the adjacent air it will rise up due to the buoyancy effect. When air reaches the top side and moves toward the vertical outer cold side it becomes cooler and drops down. Adding fins to the inner hot side increases the heat transfer and, hence the velocity in the vertical direction increases. This can be seen clearly in Fig. 12. When the velocity has a positive value, this means the fluid movies to the upper direction and when the velocity has a negative value this means the fluid movies to the lower direction. Note that for the case with no fins, the air is almost stagnant at the midpoint between the two vertical surfaces. This is because vertical fluid movement occurs near the walls. As fins are added, it disturbs the fluid movement at the midpoint and provides extra movement to the fluid.

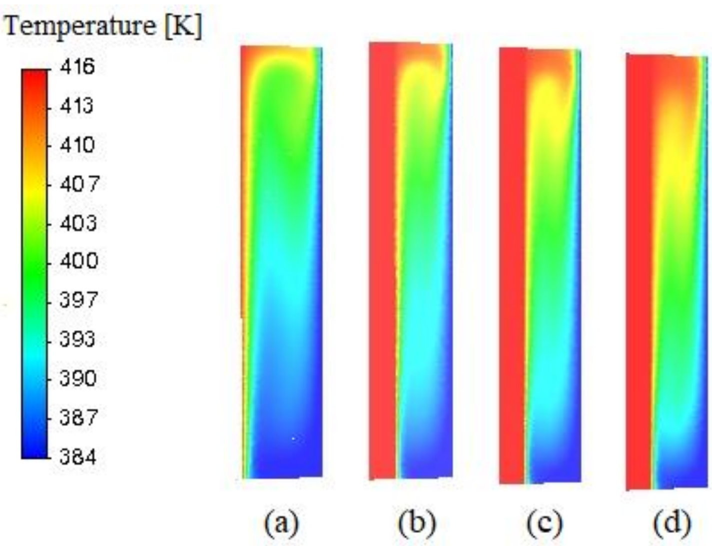

Fig. 11 Cross-sectional vertical view of temperature contour for (a) no fins, (b) four fins, (c) eight fins, and (d) sixteen fins.

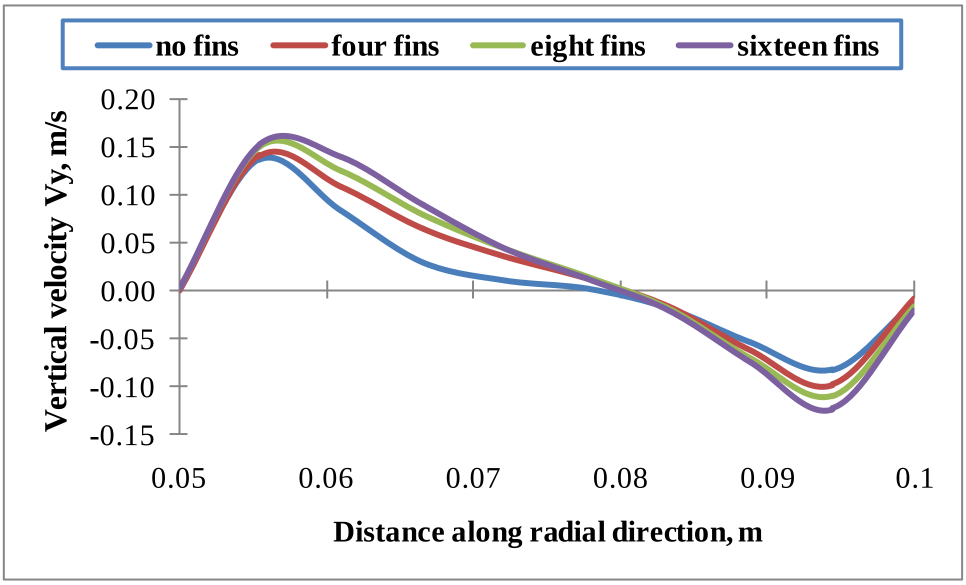

Fig. 12 Vertical velocity profile along the radial direction at the vertical midpoint from the inner hot surface to outer cold surface at different fins numbers.

6.2. Effect of fins length

Fig. 13 shows the effect of fins length on heat transfer performance. When fins length increases, the contact surface area of total fins exposed to the enclosed fluid is increased, and thus, the heat transfer rate increases. When using the fin length of 5 mm, the heat transfer is enhanced by 5% compared to no fins case. When using the fin length of 10 mm, the heat transfer is enhanced by 16% compared to no fins case. When using the fin length of 15 mm, the heat transfer is enhanced by 27% compared to no fins case.

Fig. 14 shows the effect of fins length on the temperature profile along the radial direction from the inner hot surface to the outer cold surface. It shows that the temperature profile generally increases with the increase of fins length. It also shows that the temperature gradient increases toward the inner hot surface while decreasing toward the outer cold surface. Figure 15 shows a cross-sectional view of the temperature contour plot with different fin lengths. This section was taken at the horizontal midsection. For longer fins, the fluid temperature at the inner hot wall (left), which is closer to the fins gets higher in temperature. This clearly enhances the heat transfer rate.

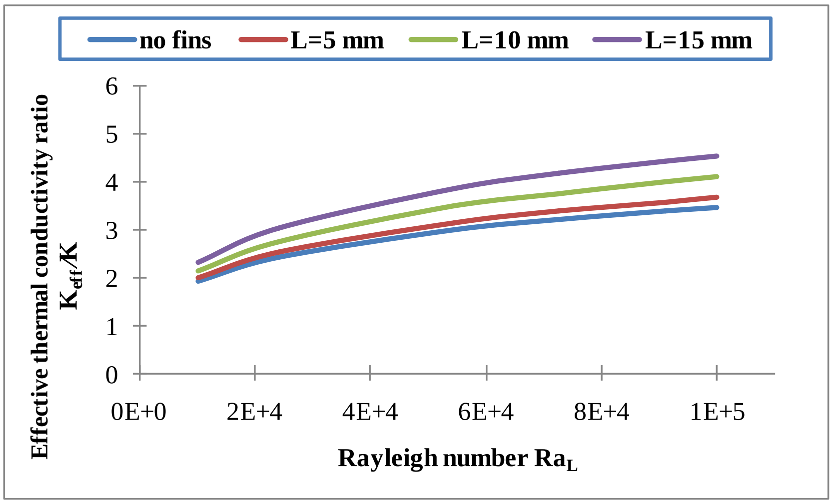

Fig. 13 Effective thermal conductivity ratio vs. Rayleigh number at different fins length.

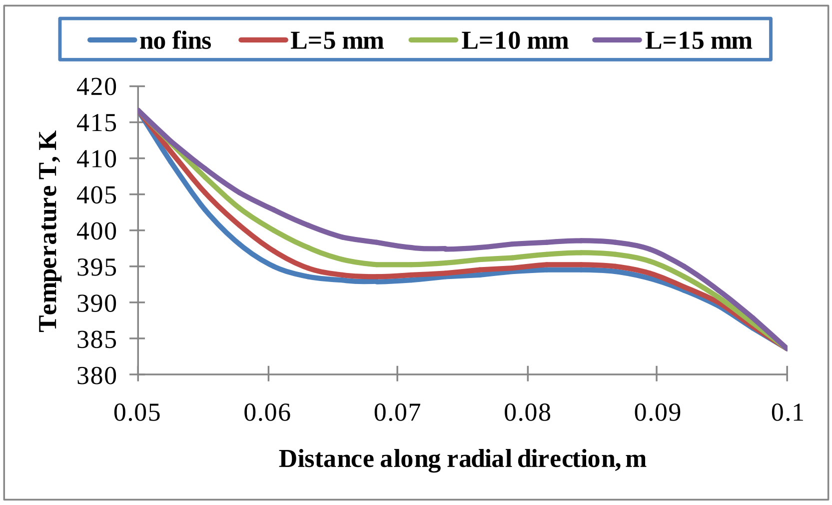

Fig. 14 Temperature profile along the radial direction at the vertical midpoint from the inner hot surface to the outer cold surface at different fins lengths.

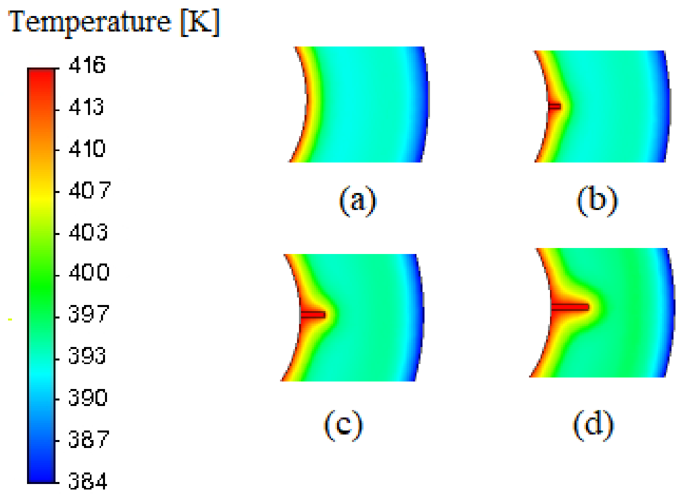

Fig.15 Cross-sectional horizontal view of temperature contour for (a) no fins, (b) L=5 mm, (c) L= 10 mm, and (d) L= 15mm.

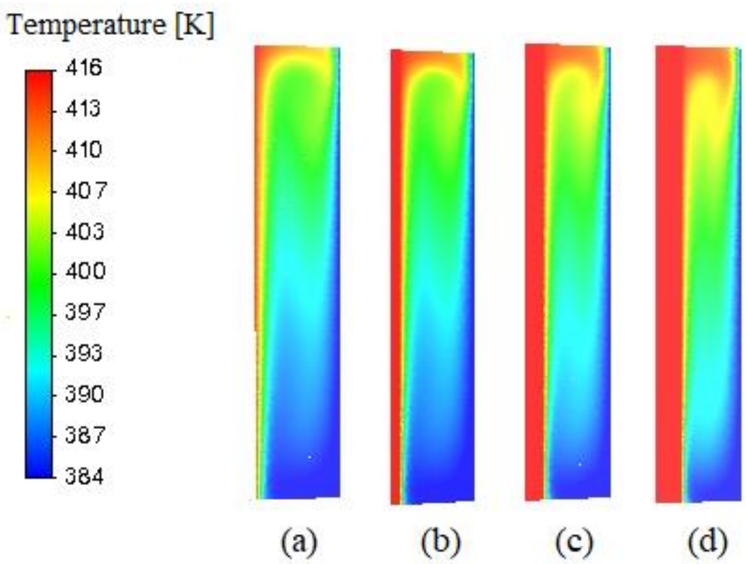

Fig.16 Cross-sectional vertical view of temperature contour for (a) no fins, (b) L=5 mm, (c) L= 10 mm, and (d) L= 15 mm.

Fig. 16 shows a vertical side view of the temperature contour plot with different fin lengths. This section was taken at the vertical midsection. The figure shows wider layers of fluid at a specific range of temperature. This is explained by longer fins, which raise the fluid temperature on the left side.

When the fins length increases, the heat transfer enhances therefore, the velocity profile increases. Moreover, when fins are placed inside the annulus, the volume of the air domain is reduced compared to the annulus without fins hence, the cross-sectional area decreases causing the vertical movement of mass to increase. This is based on the conservation of mass which can be seen clearly in Fig. 17.

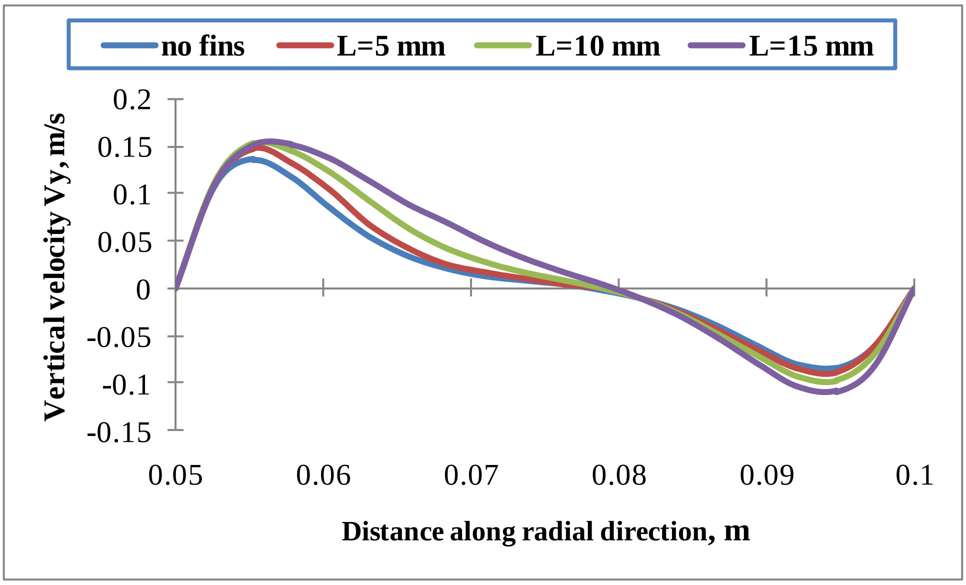

Fig. 17 Vertical velocity profile along the radial direction at vertical mid-point from the inner hot surface to outer cold surface at different fins lengths.

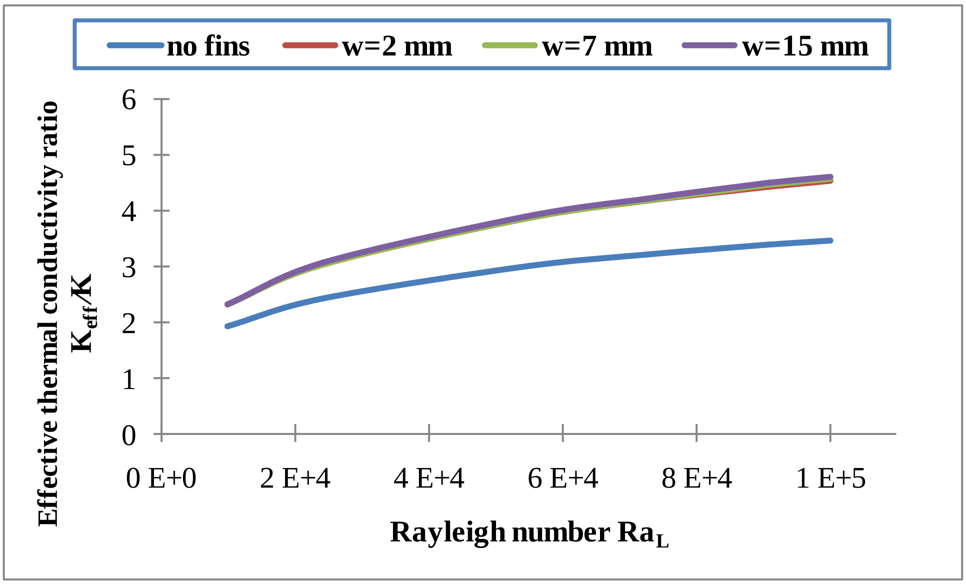

Fig. 18 Effective thermal conductivity ratio vs. Rayleigh number at different fins thicknesses

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6.3. Effect of fins thickness

Fig. 18 shows the effect of fins thickness on heat transfer performance. when the fins' thickness increases, the contact surface area of total fins exposed to the enclosed fluid increases, and thus, the heat transfer rate increases. When using fins thickness of 2 mm, the heat transfer enhances by about 27% compared to no fins case. When using fins thickness of 7 mm, the heat transfer is enhanced by 28%

compared to no fins case. When using fins thickness of 15 mm, the heat transfer is enhanced by 29% compared to no fins case. It was noticed that the effect of the increase in fins thickness is negligible. When we increase the fins thickness we increase the area of the fins tip only. The fins tip usually has a lower surface temperature than the base temperature. This means that the temperature gradient at the tip surface is very small which cannot contribute to the total heat transfer to the fluid. This explains why the effect is so small.

Fig. 19 shows the effect of fins thickness on the temperature profile along the radial direction from the inner hot surface to the outer cold surface. It shows that the profile for the three different thicknesses used is very close to each other. This is explained by the same argument addressed above about the small temperature gradient at the fin tip.

Fig. 20 shows a cross-sectional view of the temperature contour plot with different fin thicknesses. This section was taken at the horizontal midsection. With thicker fins, the temperature gradient spreads out in the angular direction. This has little effect on the heat transfer rate in the radial direction.

Fig. 21 shows a vertical side view of the temperature contour plot with different fin thicknesses. This section was taken at the vertical midsection. It can be observed that having fins has a great effect compared to the system without fins. Even though, changing the fins' thickness doesn't change the fluid hot layers much.

The vertical velocity profile is shown in Fig. 22. It can be seen that the increase of fins thickness has little to no effect on the vertical velocity profile. The reason is that increasing the fin thickness doesn't obstruct the movement of the fluid. It doesn't also reduce the cross-sectional area on a scale that will affect the mass flow. Therefore, the fins thickness has little to no effect on the vertical velocity profile.

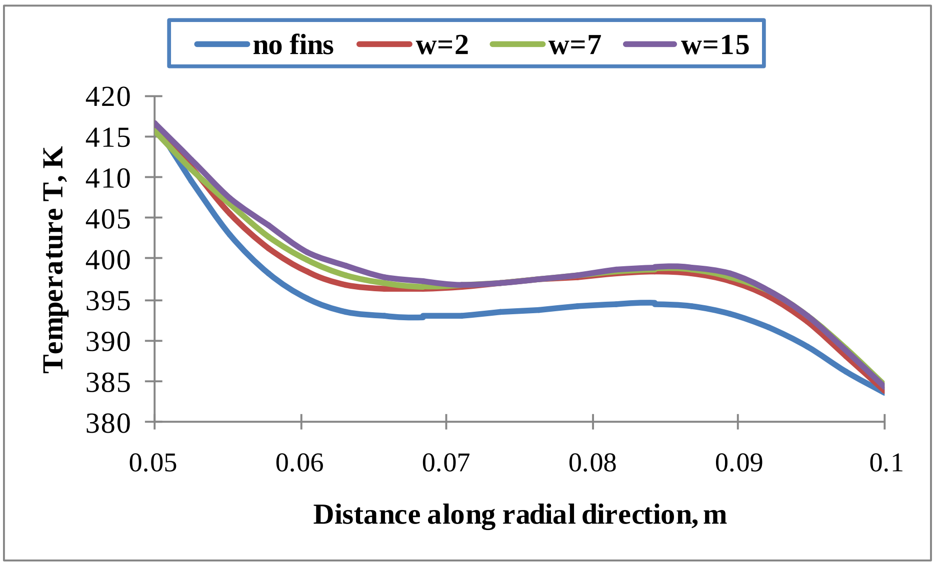

Fig. 19 Temperature profile along the radial direction at vertical mid-point from the inner hot surface to outer cold surface at different fins thickness.

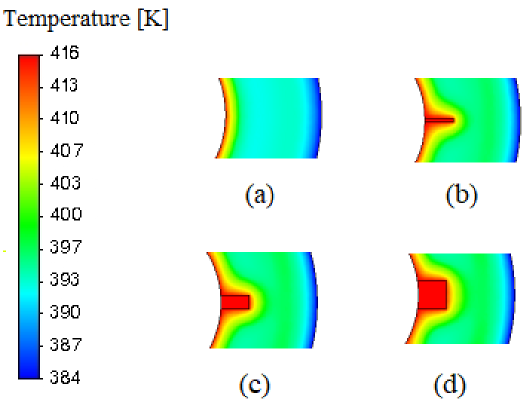

Fig. 20 Cross-sectional horizontal view of temperature contour for (a) no fins, (b) W=2 mm, (c) W=7 mm, and (d) W=15 mm.

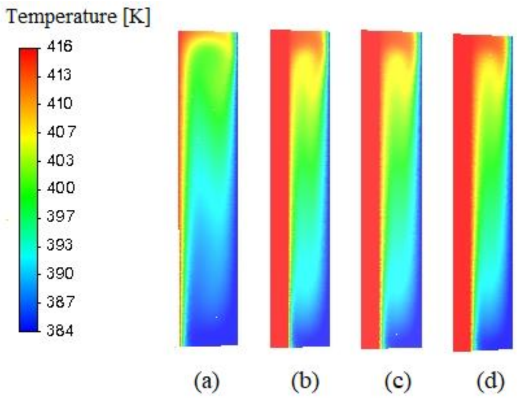

Fig. 21 Cross-sectional vertical view of temperature contour for (a) no fins, (b) W=2 mm, (c) W=7 mm, and (d) W=15 mm

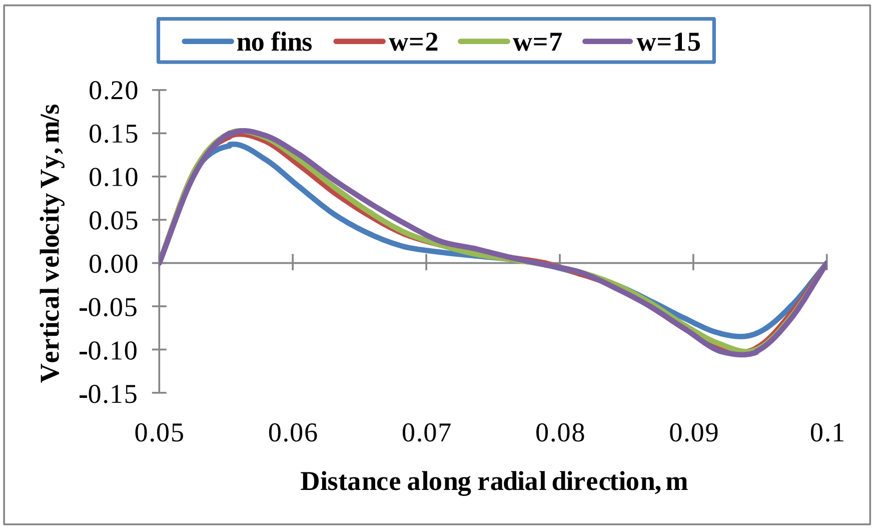

Fig. 22 Vertical velocity profile along the radial direction at the vertical midpoint from the inner hot surface to the outer cold surface at different fins thicknesses

6.4. Effect of fins shape

Fig. 23 shows the effect of fins shape on heat transfer performance. Three fins shapes were studied and compared to no fins case. These shapes are triangular, circular, and square fins. Results revealed that when using triangular fins, the heat transfer is enhanced by 21% compared to no fins case. When using circular fins, the heat transfer is enhanced by 25% compared to no fins case. When using square fins, the heat transfer is enhanced by 29% compared to no fins case. Square fins would have the largest contact surface area to the working fluid compared to triangular and circular ones. Fig. 24 shows the effect of fins shapes on the temperature profile along the radial direction from the inner hot surface to the outer cold surface. It shows that the temperature profile of the square fins has a slightly better temperature profile than triangular and circular ones.

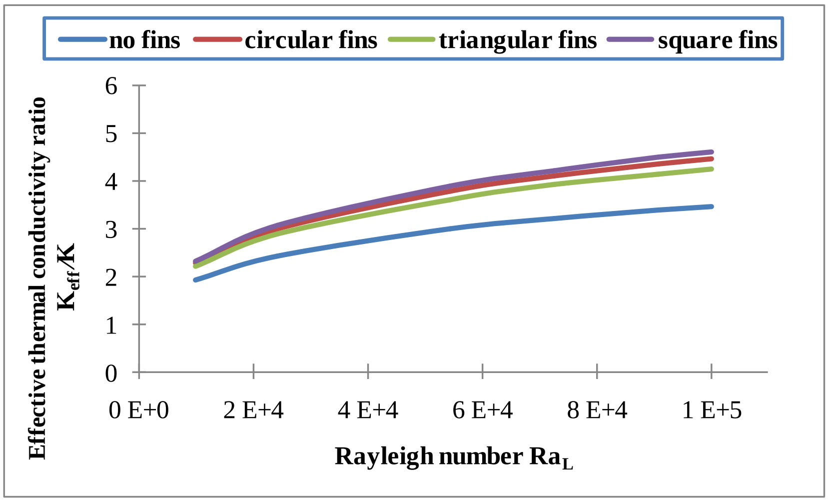

Fig. 23 Effective thermal conductivity ratio vs. Rayleigh number at different shapes of the fins.

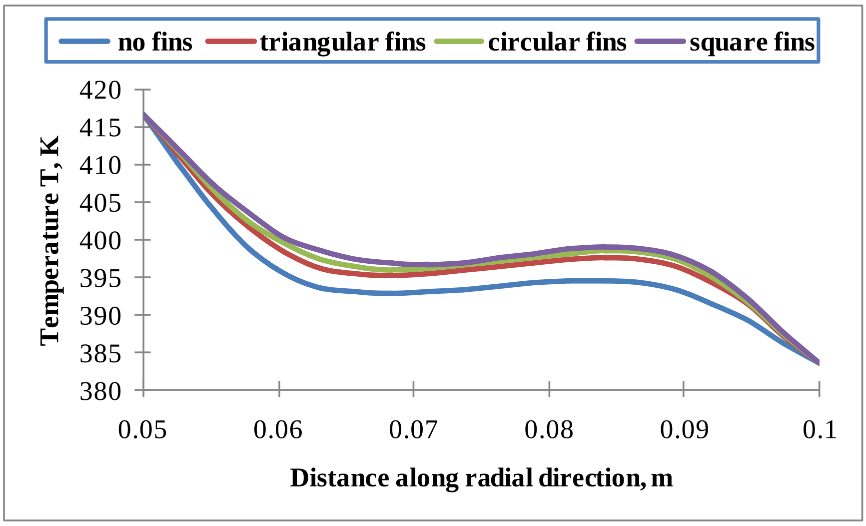

Fig. 24 Temperature profile along the radial direction at the vertical midpoint from the inner hot surface to the outer cold surface at different shapes of the fins.

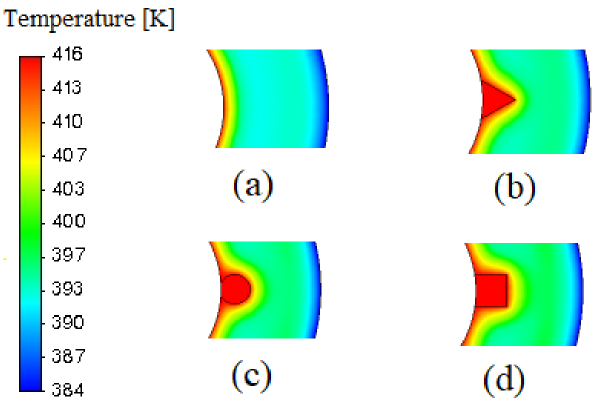

Fig. 25 Cross-sectional horizontal view of temperature contour for (a) no fins, (b) triangular fins, (c) circular fins, and (d) square fins.

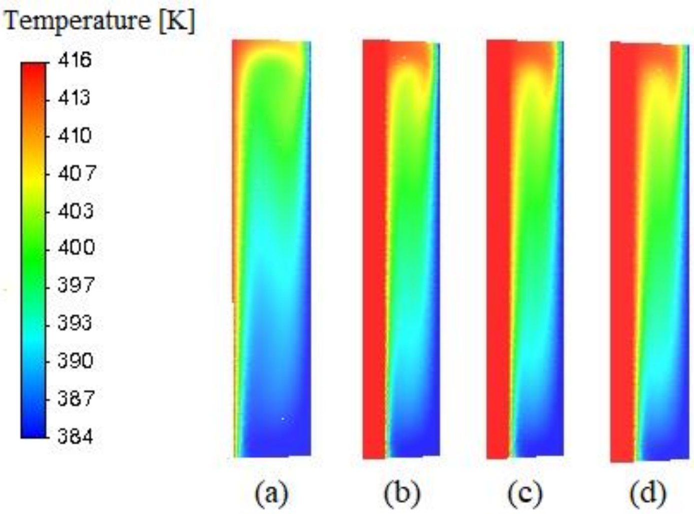

Fig. 26 Cross-sectional vertical view of temperature contour for (a) no fins, (b) triangular fins, (c) circular fins, and (d) square fins.

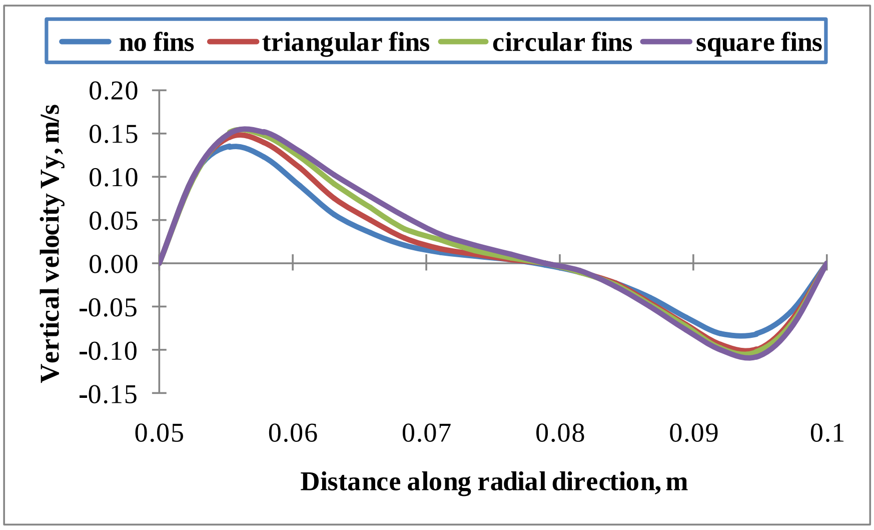

Fig. 27 Vertical velocity profile along the radial direction at the vertical midpoint from the inner hot surface to the outer cold surface at different shapes of the fins.

Fig. 25 shows a cross-sectional view of the temperature contour plot with different fin shapes. This section was taken at the horizontal midsection. Using different fin shapes spreads out the temperature profile in the angular direction rather than the radial direction. Moreover, the difference in contact surface is insignificant. Therefore, the improvement among different fin shapes is small.

Fig. 26 shows a vertical side view of the temperature contour plot with different fin shapes. This section was taken at the vertical midsection. The width of hot fluid layers closer to the inner hot wall has small changes when using different shapes of fins. However, using fins has a great improvement in heat transfer rate compared to cases without fins.

The velocity profile for different fins shapes shows that the velocity profile for the square fins is slightly more than other ones. Also, the velocity profile variations get larger towards the hot surface while closer toward the cold surface as shown in Fig. 27. This is because closer to the hot surface, we will have more dynamic and natural buoyancy movement while closer to the cold surface the fluid is denser and has less movement.

7. Conclusions

This study is for the natural convection of heat transfer in a vertical annulus enclosure where the inner surface is hot and the outer surface is cold. Fins were added to the inner surface to enhance the heat transfer performance. A numerical model was developed using ANSYS Fluent software to solve the governing equations for the model. The numerical model was grid tested and validated by comparing it to results in the literature. Afterward, a parametric study was carried out where one variable is changed while others are kept constants. Parameters chosen in our study are fins number, fins length, thickness, and shape. Results are presented for each parameter. The study results are concluded on the following points:

1- With the increase of fins number, the heat transfer enhanced up to 46% when using 16 fins. Also, it was found that this effect is similar to the one reported in the literature.

2- As the fins length increases, the heat transfer is enhanced up to 27% for 15mm fins length.

3- Fins did enhance the heat transfer by about 28%. Different fins thicknesses had a negligible effect on the heat transfer performance.

4- Square fins have the best results over circular and triangular fins by 5 to 10% respectively.

5- When the Rayleigh number increased, the heat transfer enhanced which is the same compared to the results reported in the literature. Also, when the Rayleigh number increased the temperature and velocity improved.

Nomenclature

g Gravitational acceleration (m/s^2)
Ha Hartmann number, the ratio of electromagnetic force to the viscous force.
K Thermal conductivity, W/(m.K)
k_{eff} Effective thermal conductivity, it is defined as the thermal conductivity of the air at which stagnant air in the annulus can give the same rate of heat transfer , W/(m.K).
L Fin length (mm)
P Pressure (N/m^2)
Pr Prandtl number

q' Heat transfer rate per unit length (W/m)
r Annulus radius (m)
Ra Rayleigh number
T Temperature (K)
Vy Flow velocity components in y-direction (m/s)
W Fin width (mm)

Greek Symbols
α Thermal diffusivity (m^2/s)
β Thermal expansion coefficient (K^{-1})
μ Dynamicviscosity (kg/ m.s)
ρ Density (kg/m^3)
ν Kinematic viscosity (m^2/s)
Subscripts
f Fluid
i Inner
m Modified
o Outer

Acknowledgment

I would like to express my sincere appreciation and my special thanks to my academic supervisor Dr. Abdullatif A. Gari for his great help and precious time, and effort that he provides to help me to upgrade my scientific knowledge and my research skills. Also, my sincere appreciation to King Abdulaziz University's High Performance Computing Center where my study was performed using their supercomputer (Aziz Supercomputer).

LIST OF REFERENCES

[1] Bergman, T. I., Lavine, A. S. and Incropera, F. P., *Fundamentals of Heat and Mass Transfer*, John Wiley & Sons, 7th, (2013).

[2] Rohsenow, W. M., Hartnett, J. P. and Cho, Y. I., *Heat Transfer*, 3rd, McGraw-Hill, N.Y., 11, (1998).

[3] Nada, S. A., Experimental investigation of natural convection heat transfer in horizontal and inclined annular fluid layers, *Heat and Mass Transfer*, 44, 929–936, (2008).

[4] Yeh, C. L., numerical investigation of the three-dimensional natural convection inside horizontal concentric annulus with specified wall temperature or heat flux, *International Journal of Heat and Mass Transfer*, 45, 775-784, (2002).

[5] Kumari, M. and Nath, G., Unsteady natural convection from a horizontal annulus filled with a porous medium, *International Journal of Heat and Mass Transfer*, 51, 5001–5007, (2008).

[6] Yoo, J. S., Prandtl number effect on bifurcation and dual solutions in natural convection in a horizontal annulus, *International Journal of Heat and Mass Transfer,* 42, 3279-3290, (1999).

[7] Wang, W., Li, B. W., Varghese, P. L., Leng, X. Y. and Tian, X. Y., Numerical analysis of three-dimensional MHD natural convection flow in a short horizontal cylindrical annulus, *International Communications in Heat and Mass Transfer*, 98, 273–285, (2018).

[8] Yang, X. and Kong, S. C., Numerical study of natural convection in a horizontal concentric annulus using smoothed particle hydrodynamics, *Engineering Analysis with Boundary Elements*, 102, 11–20, (2019).

[9] Belabid, J. and Allali, K., Effect of temperature modulation on natural convection in a horizontal porous annulus, *International Journal of Thermal Sciences*, 151, 106273, (2020).

[10] Reddy, P. V. and Narasimham, G. S., Natural convection in a vertical annulus driven by a central heat generating rod, *International Journal of Heat and Mass Transfer*, 51, 5024–5032, (2008).

[11] Sankar, M., Park, Y., Lopez, J. M. and Do, Y., Numerical study of natural convection in a vertical porous annulus with discrete heating, *International*

Journal of Heat and Mass Transfer, 54, 1493–1505, (2011).

[12] Chen, J. C., Chou, G. H. and Hsieh, C. K., The influence of inner-wall motion on the linear stability of natural convection in a tall vertical annulus, *International Journal of Heat and Mass Transfer*, 39, 193- 201, (1996).

[13] Sankar, M., Venkatachalappa, M. and Shivakumara, I. S., Effect of magnetic field on natural convection in a vertical cylindrical annulus, *International Journal of Engineering Science*, 44,1556–1570, (2006).

[14] Husain, S. and Siddiqui, M. A., Experimental and numerical analysis of transient natural convection of water in a high aspect ratio narrow vertical annulus, *Progress in Nuclear Energy*, 106, 1–10, (2018).

[15] Afrand, M., Toghraie, D., Karimipour, A. and Wongwises, S., A numerical study of natural convection in a vertical annulus filled with gallium in the presence of magnetic field, *Journal of Magnetism and Magnetic Materials*, 430, 22–28, (2017).

[16] Abhilash, R. and Lal, S. A., Three dimensional analysis of natural convection in a narrow vertical annulus closed at top and opened at bottom, *International Journal of Thermal Sciences*, 127, 277–287, (2018).

[17] Wang, Z., Liu, Y., Zhang, J. and Dang, N., Study of laminar natural convection in a vertical annulus with inner wall covered by a porous layer by using lattice Boltzmann method, *International Journal of Thermal Sciences*, 135, 386–397, (2019).

[18] Char, M. I. and Lee, G. C., Maximum density effects on natural convection in a vertical annulus filled with a non-Darcy porous medium, *Acta Mechanica*, 128, 217-231, (1998).

[19] Jha, B. K., Babatunde, A. and Muhammad, S., Combined effects of suction/injection and wall surface curvature on natural convection flow in a vertical micro-porous annulus, *Thermophysics and Aeromechanics*, 22, 217-228, (2015).

[20] Rahnama, M. and Farhadi, M., Effect of radial fins on two-dimensional turbulent natural convection in a horizontal annulus, *International Journal of Thermal Sciences*, 43, 255–264, (2004).

[21] Kiwan, S. and Zeitoun, O., Natural convection in a horizontal cylindrical annulus using porous fins, *International Journal of Numerical Methods for Heat and Fluid Flow*, 18, 618-634, (2008).

[22] Nada, S. A. and Said, M. A., Effects of fins geometries, arrangements, dimensions and numbers on natural convection heat transfer characteristics in finned-horizontal annulus, *International Journal of Thermal Sciences*, 137, 121–137, (2019).

[23] Alshahrani, D. and Zeitoun, O., Natural convection in horizontal cylindrical annuli with fins, *Alexandria Engineering Journal*, 44, 825-837, (2005).

[24] Shadlaghani, A., Farzaneh, M., Shahabadi, M., Tavakoli, M. R., Safaei, M. R. and Mazinani, I., Numerical investigation of serrated fins on natural convection from concentric and eccentric annuli with different cross sections, *Journal of Thermal Analysis and Calorimetry*,135, 1429–1442, (2019).

[25] Ramezanpour, M. and Hosseini, R., An experimental study of natural convection in vertical annulus with helical fin, *Experimental Heat Transfer*, 33, 226-244, (2019).

[26] Solomon, G. R. and Velraj, R., Analysis of the heat transfer mechanisms during energy storage in a Phase Change Material filled vertical

finned cylindrical unit for free cooling application, *Energy Conversion and Management*, 75, 466–473, (2013).

[27] Kumar, R., Three-dimensional natural convective flow in a vertical annulus with longitudinal fins, *International Journal of Heat and Mass Transfer*, 40, 3323-3334, (1997).

[28] Sehgal, B. R., *Nuclear Safety in Light Water Reactors*, Academic Press, (2012).

[29] Grigull, U. and Hauf, W., Natural Convection in Horizontal Cylindrical Annuli, *Proceeding of the 3rd International Heat Transfer Conference*, 2, 182-195, (1966),

[30] Kuehn, T.H. and Goldstein, R. J., An Experimental and Theoretical Study of Natural Convection in the Annulus between Horizontal Concentric Cylinders, *Journal of Fluid Mechanics*, 74, 695-719, (1976).

الحمل الحراري الطبيعي في حاوية حلقية العمودية ذات زعانف طولية

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مستخلص. يمكن العثور على انتقال الحرارة بالحمل الطبيعي في العديد من تطبيقات الهندسة الحرارية. من أجل تحسين انتقال الحرارة ، يتم استخدام الزعانف لزيادة سطح التلامس وبالتالي معدل انتقال الحرارة. في هذا المشروع ، تمت دراسة أداء انتقال الحرارة بالحمل الطبيعي في الغلاف الحلقي العمودي عدديًا. تم استخدام برنامج الأنسس فلوينت لحل المعادلات الحاكمة لحساب السرعة وتوزيع درجة الحرارة. تم إجراء اختبار الشبكة وتم التحقق من صحة النموذج مع الدراسات التجريبية والرقمية السابقة. أظهرت النتائج أن للزعانف تأثيراً بارزاً في تعزيز انتقال الحرارة مقارنة بالنموذج بدون زعانف. بعد ذلك تم إجراء دراسة بارامترية لدراسة تأثير عدد الزعانف وطولها وسمكها وشكلها. تؤدي زيادة عدد الزعانف إلى زيادة معدل انتقال الحرارة بنسبة ٤٦٪. يتوافق تأثير عدد الزعانف على معدل انتقال الحرارة مع النتائج المذكورة في الأدبيات. زيادة طول الزعانف يزيد من معدل انتقال الحرارة بنسبة ٢٧٪. سمك الزعنفة له تأثير ضئيل على كمية انتقال الحرارة بالحمل الحراري. كما وجد أن الزعانف المربعة لها أفضل النتائج على الزعانف الدائرية والمثلثة. فيما يتعلق بتأثير عدد رالي على انتقال الحرارة, يزداد معدل انتقال الحرارة مع زيادة عدد رالي وهو مايتطابق مع النتائج المذكورة في الأدبيات.