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Characteristics of Air Flow and Heat Transfer in Serpentine Condenser Pipes with Attached Convection Plates in Open Channel

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Abstract. Many efforts have been made to reduce the energy consumption of household refrigerators. One method is to place the pipe condenser to increase the rate of heat release. This numerical study examines the effect of changing the gap ratio on the flow characteristics and natural convection heat transfer of 20 pipes attached vertically to the convection plate with aluminum foil coating. The gap ratio was varied between 1.05 and 4.20 using ANSYS-FLUENT software to obtain velocity vector, temperature contour, tangential velocity, and local Nusselt number, both inside and outside the channel. With the change in gap ratios from 1.05 to 2.10, the rate of heat transfer increased significantly, reaching 2.2% while tangential velocity also increased considerably. At gap ratios of 3.15 and 4.20, the rate of heat transfer increased more gradually, local Nusselt number increased slightly where the influence of convection walls was smaller, and tangential velocity showed a very small increase. Flow characteristics were similar, with air flowing upward across the inner and outer channels.

Keywords: Gap ratio; Household refrigerator; Natural Convection; Nusselt Number; Vertical Channel

1. Introduction

Household refrigerators are an important part of daily life, and researchers have made many attempts to make them more energy-efficient. Several factors contribute to the energy-efficiency of household refrigerators, including consumer behavior and heat dissipation. One factor in consumer behavior that affects refrigerator energy consumption has been demonstrated by Susanto et al. (2018), who tested the relationship between thermostat regulation and increased energy consumption of household refrigerators. The relationship between refrigerator energy consumption and heat transfer factors was investigated by Sefcik et al. (1991) conducted experiments on natural convection behavior in enclosures that were ventilated at the lower and upper ends to provide air flow to improve heat transfer. In his experiment, the average Nusselt number changed according to the size of the two ventilation holes.

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Research conducted by Davies et al. (2000) found that changes in free air temperature affect the Nusselt number:the higher the free air temperature the lower the Nusselt number. Chouikh et al. (1998) conducted a numerical study for natural convection flow along a range of two isothermal horizontal pipes by varying the Rayleigh number (Ra)and pipe spacing (Ra was used to determine the laminar to turbulent transition from the flow of natural convection boundary layers). The results showed that Ra increased and the air temperature gradient in the pipes was steeper so that the rate of heat transfer increased, and vice versa. Manca et al. (2002) studied natural convection heat-transfer characteristicsusing of discrete heated plates parallel slope variations; they tested the hypothesis that at an angle smaller than 85^{o} air flow wouldmove outside thereby increasing the temperature inside the channel. Manca et al. (2002) studied the characteristics of natural convection heat transfer using; it is stated that at a slope angle <85o it causes an inflow of air from the upper side which prevents the outflow of air thereby increasing the temperature in the channel. Buonomo et al. (2017) studied two horizontal parallel walls by filling the stem with porous media, findingthat the use of porous media resulted in an increase in heat transfer. Dehghandokht et al. (2011) performed numerical analysis on a multiport serpentine meso-channel heat exchanger. Their simulation showed that the effect of serpentine bends on heat exchangers will increase average heat transfer by almost 20% compared withthose using straight plates.

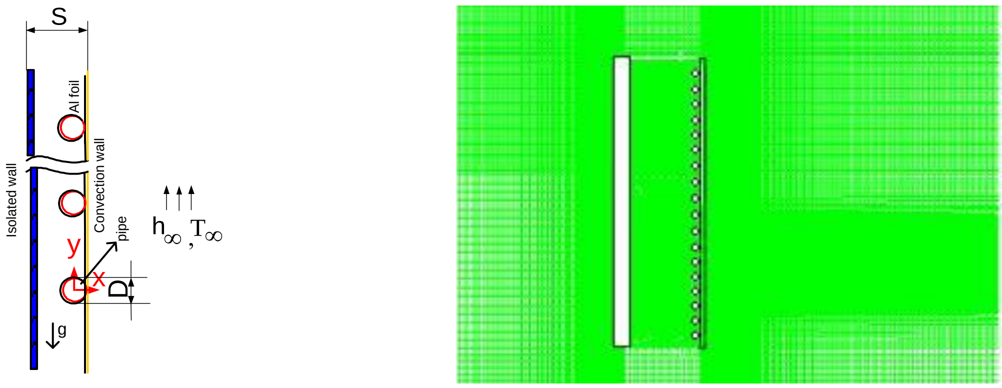
Lewandowski et al. (2018a) observed natural convection byvarying channel width and wall temperature, the results finding that wider air ducts caused significantly heat transfer. Ospir et al. (2012) investigated dynamic flow in vertical plate channels with Rayleigh numbers and gap ratiomodification;they used laser tomography to visualize flow, finding that a larger gap ratio causes the length of the upper cell to decrease. Alzwayiet al. (2014) performed numerical simulations to investigate the effect of channel width on transition flow under various plate temperatures, usingthe k-tur turbulence model to simulate flow and thermal fields in the channel;their results showed that the transition flow in isothermal cases is slower than adiabatic cases.

Lewandowski et al. (2018b) investigated the distribution and loss of heat from building walls using a Thermal Imaging Camera (TIC). they found that the use of infrared cameras made it possible to determine local heat loss. Lewandowski et al. (2017) also analyzed convection heat transfer to two parallel and vertical plates where the slit plate-width varies;infrared observation showed there was a relationship between the width of the gap and the rate of heat transfer.

The above studiesgenerally conclude that the heat transfer observedoccurs throughnatural convection. In natural convection, fluid flows due to differences in density caused by differences in temperature (buoyancy) and the absence of external influences such as fans. Serpentine pipe bends cause an increase in heat transfer. The novelty of this research is the combination of serpentine pipe bends that are attached to vertical plates, providing space for air flow. This is of interest because it causes a greater increase in heat transfer. The purpose of this research is to study the numerical effect of changing the gap ratio on the flow characteristics and natural convection heat transfer from 20 pipes that are mounted vertically to the convection plate with a layer of aluminum foil.

2. Methods

Numerical simulations can help save time and lower costswhen conducting experiments (Ramdlan et al., 2016; Pujowidodo et al., 2018). Here, modelling wasdone by attaching 20 pipes in the upright channel as shownin Figure 1a. Vertical walls are defined as insulating and convection walls. The gap distance (S) varies from 0.006–0.027 m, with gap ratios (S/D) of1.05, 2.10, 3.15, and 4.20. The air inside and outside of the upright channel is defined as outflow.



(a) (b)

Figure 1 Computational domain (a), andmesh generation (b) models

Mesh generation is used for quadrilateral map meshing type; mesh on the surface of the pipe is tighter to observe the fluid property changes near the pipe, while at longer distances from the pipe the mesh is looser to save memory and speed up computer convergence (Figure 1b). Table 1 details the amount of meshing.

Table 1 Meshing results

Nodes	452054
Elements	904108
Maximum Skewness	0.94

The meshing quality standard of the model produces a maximum skewness value of 0.94, which is acceptable based on ANSYS-FLUENT software standards (Figure 2).

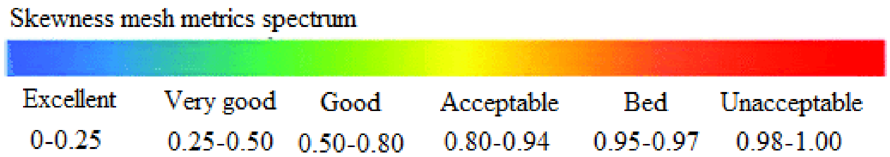


Figure 2 Standard Skewness

The experimental method comprises a condenser consisting of 20horizontal cylinders arranged in-line (Figure 3). Copper pipe diameter (D) is 6.35 mm (1/4 inch);the convection wall is made of aluminum plate with a height (H) of 695 mm and width (W) of 435 mm;the distance between convectionwalls(S) varies from 7–27 mm, andcan be expressed as variations of the gap ratio (S/D) from 1.05–4.20. The left wall is isolated, the right wall is a convectionwall, and the top and bottom are open so that the air in the gap is connectedto the outside air.

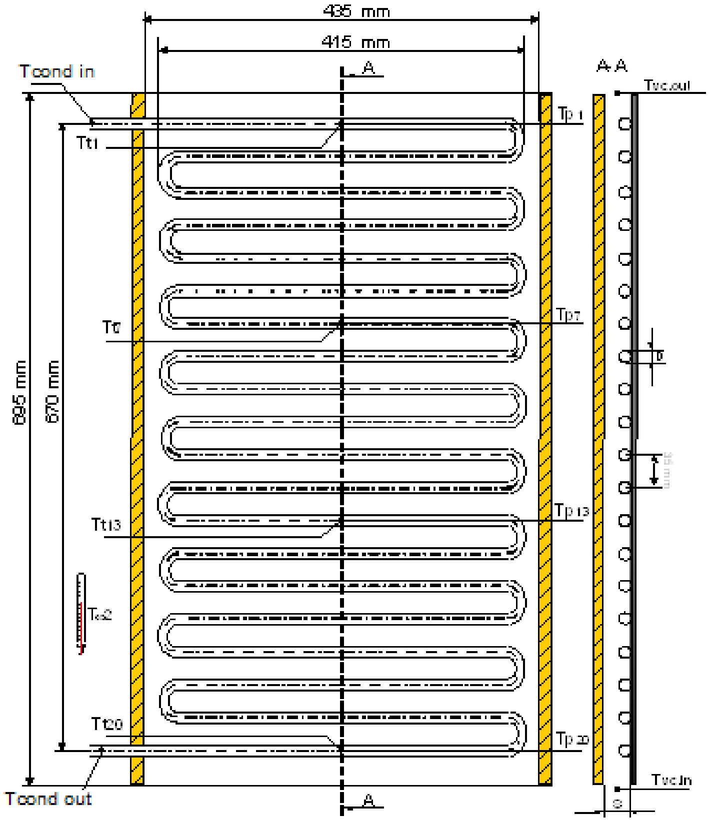


Figure 3 Layout scheme for condenser measuring devices

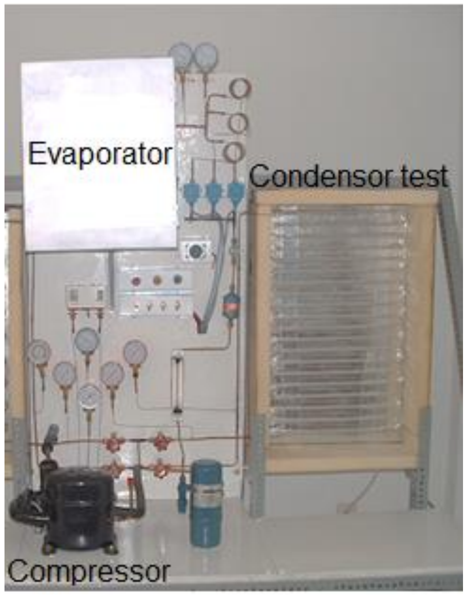


Figure 4 Schematic experiment

3. Results and Discussion

Figure 5 shows the increase in total heat transfer rate as being significant in the S/D range of 1.05–2.10, with a high gradient increase of 284.44 Watt to 290.77 Watt (about 2.2%).This indicatesthe stronger influence of the convection wall onthe heat transfer rate, which is caused by the small air flow space crossing the pipe or close to the enclosure conditions. In the S/D range of 2.10–4.20,theincrease is not significant (as shownby the gently sloping gradient increase), indicating the weaker influence of the convection wall onthe rate of heat transfer. This occurs because the air velocity across the pipe increases with a greater gab ratio, whereas the air temperature across the pipe is lower due to the greater air flow entering the channel. Table 2 gives the numerical Q_{cond} (heat transferred in the condenser) and experimentalQ_{cond} values for each S/D value.

Table 2 Comparison of numerical $Q_{cond}$and experimental Q_{cond} values

S/D	1.05	2.10	3.15	4.20
Q_{cond}Numeric (Watt)	284.44	290.77	292.28	293.87
Q_{cond}Experiment (Watt)	240.44	244.77	245.28	247.76

Average experimental $Q_{cond}$was lower than average numerical Q_{cond} because there is a greater loss of heat transfer from experimental$Q_{cond.}$ The average deviation of numerical Q_{cond} and experimental$Q_{cond}$was 19%.

The temperature boundary layer contours on the outside and inside of the channel expanded upward in the direction of hot air flow (Figure 6).

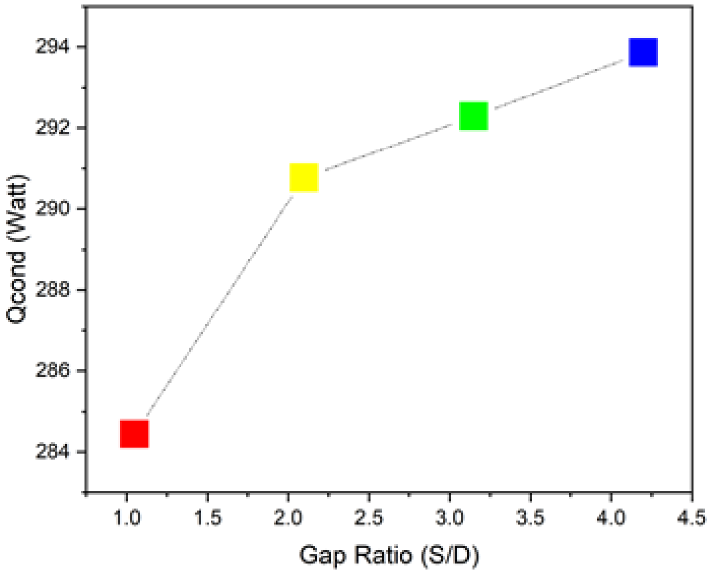


Figure 5 Effect of gap ratio on condenser heat transfer (numeric)

High temperatures result in less-dense air, which risesdue to buoyancy. The effect of capillary length change is not very influential on the air temperature contours that cross the open channel.

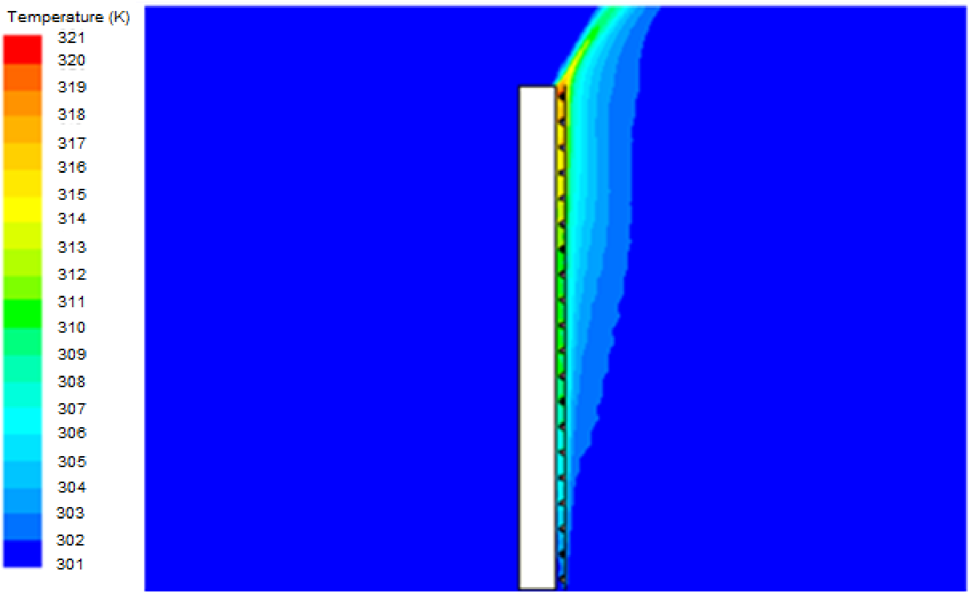


Figure 6 Statistics contour of air temperature atgap ratio (S/D) of2.10

3.1. Air Speed Across the Gap

Figure 7 shows the effect of gap ratio (S/D) on air velocity distribution.Larger values ofS/D cause increasedair velocity across the gap pipe, indicating greater rates of air mass entering the channel.

Figure 8 shows the velocity vectors. AtS/D = 1.05,there appears to be no airflowon the inside of the channel due to the small air flow space crossing the pipe or close to the enclosure condition, while at the top of the convection plate there is degradation of color indicating an increase in speed caused by the gradient of the plate temperature and the airgets bigger, causing the buoyant force to increase on the side of the convection plates outside the upright channel.

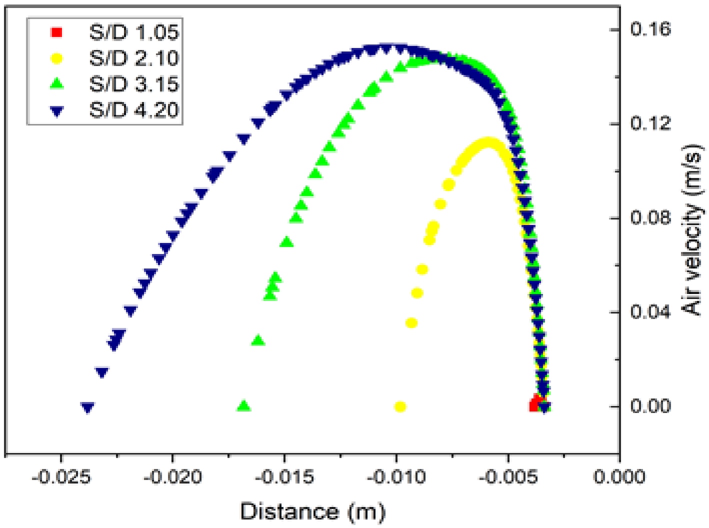


Figure 7 Effect of gap ratio (S/D) on air velocity

At S/D = 2.10, there is degradation of color in the area near the pipe indicating accelerated air flow; this is due to the air flow thatfollowing the cylindrical profile and the effect of widening the channel gap between the pipe and the adiabatic wall (polyurethane). At S / D values 3.15 and 4.20, the effect of widening the channel gap between the pipe and the adiabatic wall is not very influential so that the velocity contour is not much different which causes the air flow to look sloping.

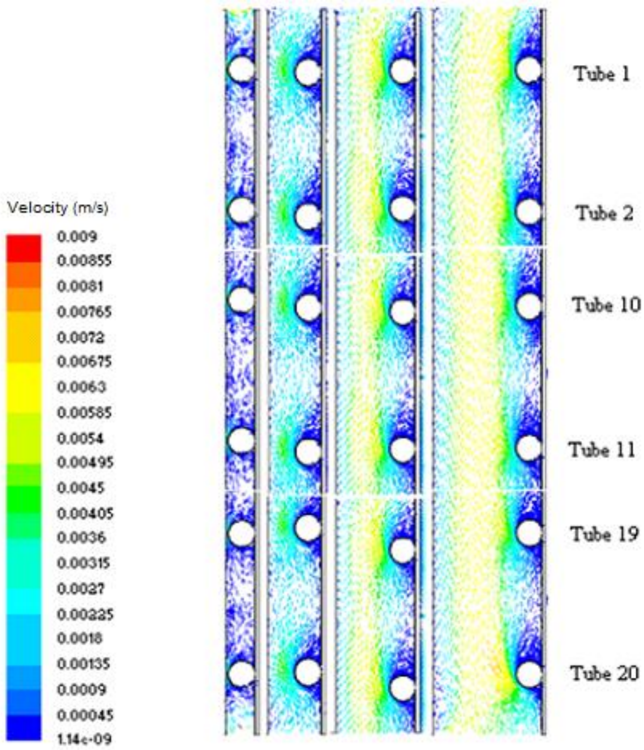


Figure 8 Velocity vectorsat gap ratios (S/D) of1.05, 2.10, 3.15, and 4.20

3.2. Air Temperature Across the Gap

Figure 9 shows the highly dominant effect of gap ratio.The larger the gap ratio the lower the air temperature across the gap pipe; this indicates the weak influence of buoyancy due to the greater rate of air mass coming into the channel.

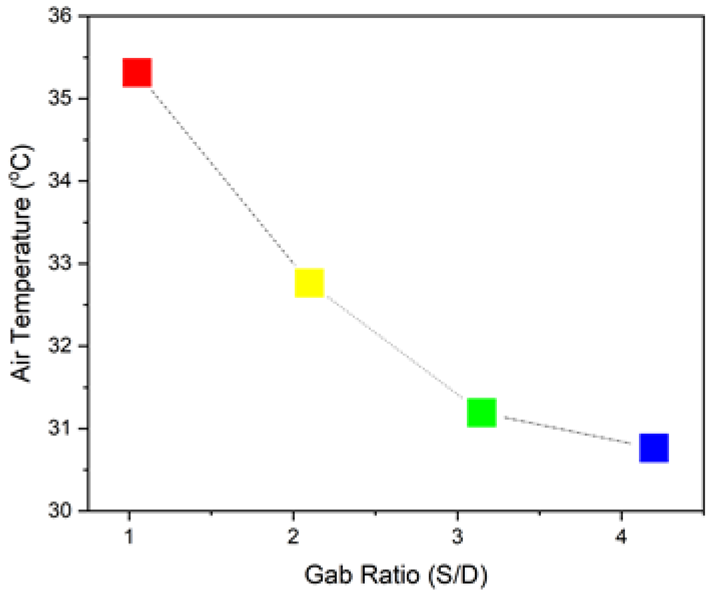


Figure 9 Distribution of air temperature across gap ratios (S/D) of 1.05, 2.10, 3.15, and 4.20 in pipe 10

3.3. Flow Patterns

Figure 10 shows vector flow velocity contours of air crossing the pipe (which is attached vertically to the convection plate with the coated aluminum foil) at an S/D value of 2.10. The air flow starts to accelerate in the upstream pipe with a pressure gradient smaller than zero, which is not sufficientto make the air particles stop because the pressure decreases in the direction of flow. When the air flow crosses the circular profile, a portion of air flow istrapped into the wake area in front of the pipe attached to the convection plate. And when the flow crosses the circular profile, the flow is accelerated and the shear force is unable to stop the flow distribution.

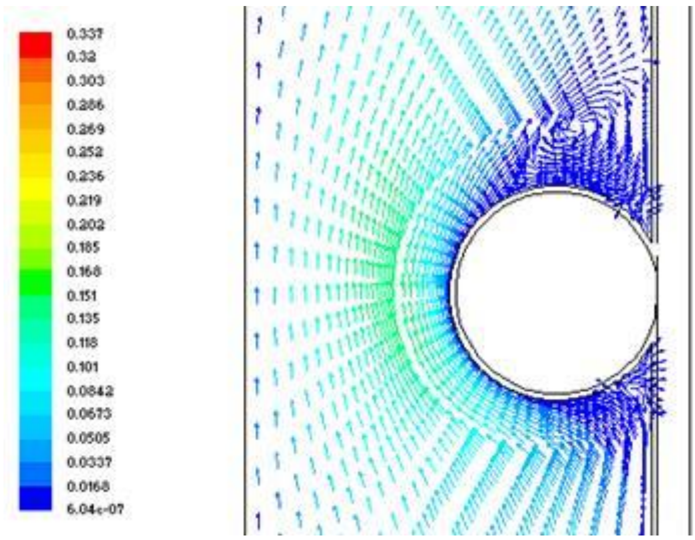


Figure 10 Vector flow velocity (m/s) contours at a gap ratio (S/D) of 2.10 in pipe 10

After passing through the circular profile, the airflow distribution begins to slow down gradually so that the momentum of the air is unable to cope with the arising shear forces and backpressures. This is due to increased pressure in the flow direction with an adverse pressure gradient greater than zero resulting in separation. Furthermore, the boundary layer will separate from the circle profile which is then called to as a "wake" that occurs in a low-pressure area. Vortex formation is visible in the wake area on the back side of the pipe due to the accelerated airflow that disrupts the flow profile on the downstream pipe. The formation of this separation vortex in the downstream pipe has a negative impact on the flow pattern because it slows the acceleration of the airflow on the pipe behind it if it has more than one pipe, whileit has a positive impact in terms of increased heat transfer characteristics of tangential velocity and Nusselt number on the downstream pipe.

Figure 11 shows the outer side of the convection plate at an S/D value of 1.05. Degradation of color indicates that there was an increase in speed caused by the plate and air temperature gradient so that the buoyant force increased.

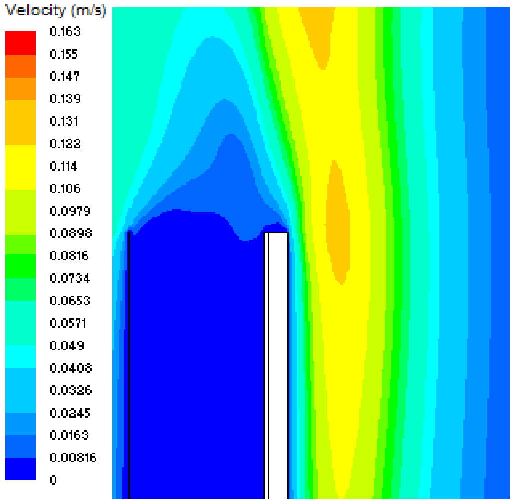


Figure 11 Contour of outside flow rate at agap ratio (S/D) of1.05

3.4. Tangential Velocity

Figure 12 shows the distribution of tangential air velocity across the pipe at gap ratios (S/D) of1.05, 2.10, 3.15, and 4.20. The greater the gap ratio, the greater the tangential air velocity across the pipe surface.

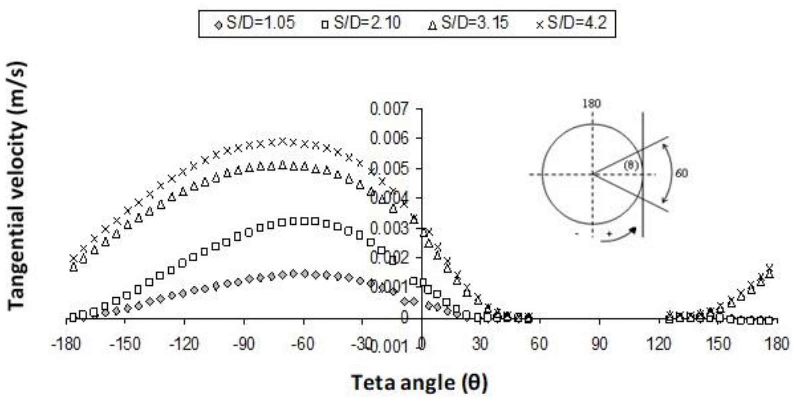


Figure 12 Distribution of tangential velocity at gap ratios (S/D) of1.05, 2.10, 3.5, and 4.20

It also appears that tangential velocity distribution in the region where $-75^{o}<\theta<60^{o}$undergoes acceleration and then decelerates to the $\theta<-75^{o}$ region. At θ values of 60º–120º no tangential velocity distribution occurs; this is because the area of the pipe is attached to the convection plate so that no air flows. However, in the area of $120^{o}<\theta<180^{o}$ the flow has accelerated again due to vortex in the downstream of the pipe.

3.5. Nusselt Numbers

Figure 13 shows the Nusselt numberdistributionat gap ratiosof 1.05, 2.10, 3.5, and 4.20. The greater the gap ratio the greater the Nusselt number. This is due to the velocity of air entering through the channel causing the temperature difference between the pipe and the steeper air, so the distribution of Nusselt numbers also increases.

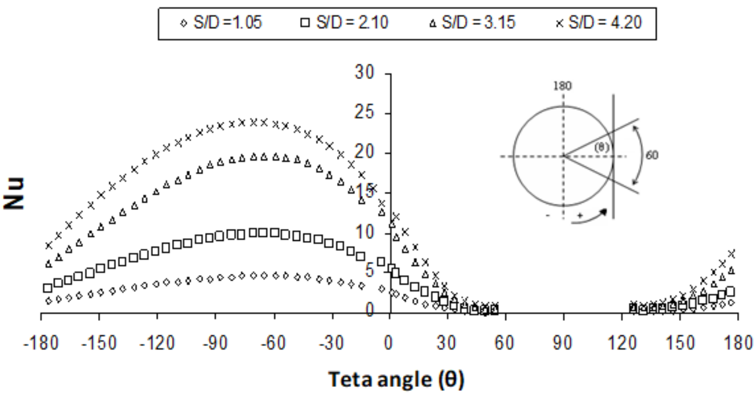


Figure 13 Nusselt number distribution atgap ratios (S/D) of1.05, 2.10, 3.15, and 4.20

Nusselt number increases at $-75^{o}<\theta<60^{o}$because the flow is accelerated;it decreases at $\theta<-75^{o}$. At θ values of 60–120º, there is no Nusselt numbers distribution; this is because the area of the pipe attaches to the convection plate so that no air flows. However, in the area of $120^{o}<\theta<180^{o}$ the flow has accelerated again due to vortex in the downstream of the pipe.

4. Conclusions

The numerical results obtained by varying the gap ratio from 1.05 to 4.20 led to the following conclusions: (1) The increase in the gap ratio from 1.05 to 2.10 caused a significant increase inthe total condenser heat transfer (an increase of 2.2% or 6.33 Watt); (2) Because the pipe was attached to the convection plate, the air velocity, tangential velocity, and Nusselt numberare increased. This causes the upstream and downstream areas to be vortex-dominated, which has a positive impact on heat transfer characteristics; (3) The air flow characteristics inside the canal tend to be the same, where free air moves from the bottom upward in the canal and convection wall due to buoyancy; (4) On the side of the pipe that adheres to the convection plate, tangential velocity and Nusselt number are zero because there is no air flowing on the side of the pipe.

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