Natural Heat Transfer from Finned Heat Exchanger Fixed in Variable Diverge Duct

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July 20, 2022

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Abstract. In the present work, an experimental study of natural heat transfer from heated cylinders fixed in variable shape (diverge air duct) made from Pyrex glass. Two finned cylinders made from mild iron, outer diameter12mm, inner diameter10 mm, rectangle fins dimensions 70 x 40 mm. The angle of the two walls of the duct was changed to six angles α: 0°, 3°, 5°,7°,10°, and 15° with the vertical to make diverge shape. The cylinders were fixed inside the duct at top, center, and bottom in three locations. The cylinders were heated by the supply of variable power of constant heat flux 88,177, 390, and 680 w/ m^2. The study was done with Rayleigh numbers ($15x10^4$ to $12\ x10^3$). the heat was carried by free convection by atmosphere air. The results of experimental found that, for top location, the value of Nusselt number decrease by 15 % at q•= 88 W/ m^2 when increase the value of angle from 0° to 3°, and decreased by 25 % at q•= 680 W/ m^2 when increase the value of angle from 0° to 10°. For center location, the value of Nusselt number decrease by 21.6 % at q•= 88 W/ m^2 when increase the value of angle from 0° to 3°, and decrease by 13.7 % at q•= 680 W/ m^2 when increase the value of angle from 0° to 10°. For bottom location, the value of Nusselt number decrease by 50 % at q•= 88 W/ m^2 when increase the value of angle from 0° to 5°, and decrease by 21 % at q•= 680 W/ m^2 when increasing the value of angle from 0° to 15°.The best location for the heat exchanger was at the bottom, and the best angle was 0°.

Keyword: *natural convection, heat convection, two finned cylinder, variable air duct, diverge,* rectangle fins.

Nomenclature:

A: surface area of heat exchanger for radiation (m^2).
A_t: total surface area of heat exchanger (m^2).
A_f: fin area (m^2).
A_b: bare cylinder area(m^2).
a: height of fin(m).
b: width of fin (m).
D: hydraulic diameter for the fin (m).
d_o: outer diameter for the cylinder (m).
d_i: inner diameter for the cylinder (m).
g: the gravitational of acceleration (m/ S^2).
h: heat transfer coefficient (W /m^2.k) .
I: electrical current (Ampère).
K_f: the thermal of conductivity for conduction (W/m. k).
L: the cylinder length(m)
Nu: the Nusselt number.
n: fins number.
α: shape wall angle

THE INTRODUCTION

The free convection for heat transfer has been a major concern due to its various practical engineering applications like nuclear reactor cooling systems and electronic equipment cooling, the cooling system of air conditioning, condenser tube, solar energy collectors. Heat transfer studied for various geometries, due to the low coefficients of heat transfer, The technology has progressed to improve the rate of heat transfer. One of the division's problems that has prompted concern in recent years is free convection heat transfer from one single cylinder or arrays of cylinders. Effects of confining walls on heat transfer rate from a single cylinder or arrays of cylinders have been achieved extensively in recent years. Many researchers explored the thermal transfer mechanisms from a cylinder through free convection. Hannani et al., [1], investigated were theoretical and numerical methods in order to perform heat transfer via natural convection from a array of horizontal cylinders fixed between two vertical walls with low Rayleigh numbers. The height of the walls remained constant, but cylinders numbers, separated distances, horizontal distance between two walls had been changed. According to the findings, the Nusselt number increased with increasing the Numbers of the cylinders, or increase the distance between two cylinders. As the Rayleigh number increased, so did the optimal separated distance of the walls. De and Dalal [2], studied the natural convection inside an enclosure centered on a horizontal cylinder with a square cross-section. Rayleigh number range (10^3-10^6) has been investigated. The aspect ratio (the ratio of the enclosure's height to its length) and location of the cylinder have been changed. The analysis revealed that the cylinder's location plays no significant role, as indicated by the local and the average Nusselt numbers. The lower the Rayleigh number, the greater the variation in overall heat transfer; however, as the Rayleigh number increased, the aspect ratio decreased. Constant wall temperature heating was more efficient than constant thermal flux for wall to heating in terms of overall the heat transfer. Xu et. al. [3], A numerical study of natural convection heat transfer from horizontal cylinder fixed in a triangular enclosure was carried out. The effects of the aspect ratio (cylinder diameter to enclosure height D/H), Rayleigh number, inner cylinder geometry, and inclination angle on heat transfer and flow were investigated. The results showed that at higher Rayleigh numbers, the construction of thermal plumes impinging on the enclosure's top corner, as well as the thermal layers in the bottom and center zones, was clearly observed. At the highest aspect ratio, the flow became multicellular, and as the aspect ratio increased, the influence became more significant. While it was discovered that the cross-section geometry had little effect on overall heat transfer.

Omar [4], investigated numerically the prandtl number effects on the free convection heat transfer of circular cylinder in an enclosed enclosure. The work achieved prandtl numbers on the heat transfer and flow characteristics. The study used various prandtl numbers [0.03,0.7,7, 50] , various Rayleigh numbers [10^4, 10^5 , 10^6] and various ratios of (W/D) which is enclosure width to cylinder diameter[1.667,2.5, 5]. The numerical results showed that the Nusselt number increased with the Rayleigh number increasing, the W/D ratios had an influence on the Nusselt number results for Pr=0.03 A. Dogan et. al. [5], has demonstrated free convection heat transfer between annular fins on a horizontal cylinder. The study's goal was to look into the effects of fin diameter (D) and fin spacing (S). Air served as the working fluid. The findings revealed that the convection heat transfer rate from the fin arrays is affected by fin diameter and fin spacing. The rate of convection heat transfer increased with fin diameter in all arrangements. As the fin spacing was increased, the rate of convection heat transfer decreased. A correlation was obtained for the optimum fin spacing (Sopt) depending on fin diameter and Rayleigh number as : (SOPT /D) = 26.43Ra-0.366

Ghufran and Hassan, [6],investigated theoretical and experimental the natural heat transfer by convection from (3) finned cylinders by spiral fins and fixed between two plates and study the effect of finned cylinder inclination angle.. Three fin cylinders with outside diameters of (12) mm, inner diameters of (10) mm, and fin diameters of (30) mm are made of (steel iron). The inclination angle of the heated cylinders was changed to four different inclination angles (0°, 30°, 60°, and 90°), as well as four different heat flux are (121.16 , 278.8 , 497.9 , and 792.5) W/m^2. Theoretical research was done by using (ANSYS Fluent) to mimic the practical aspect. The Nusselt number for finned cylinders was greatest at a 30° angle with low heat, according to experimental data.

Abdullah and Hassan [7],investigated Experimental of natural heat transfer from two horizontal finned cylinders fixed in converge duct, made from Pyrex glass. The angle of the two walls of the duct was changed to five angles α : 0°, 3°, 5°,10°, and 15° with the vertical, The cylinders were fixed inside the duct at top, center, and bottom in three locations, used constant heat flux 88,177, 390, and 680 w/ m^2 .The results of experimental found that, for top location, the value of Nusselt number increased by 7.8 % at q•= 88 W/ m^2 .For bottom location, the value of Nusselt number increased by 35 % at q•= 88 W/ m^2 when increase the value of angle from 0° to 3°.

EXPERIMENTAL SETUP

The schematic diagram is shown in Figure 1. It consists of externally two finned cylinders which fixed in variable air duct in order to change the angle to make diverge of different inclined angle α :(0°,3°,5°,7°,10°,15°). The two cylinders fixed in horizontal array. The air duct made from Pyrex glass the dimensions 550×140×70×5 mm height, length, width and thickness respectively, open from top and bottom to the surrounding air using air as a heat transfer medium. The cylinders with (d_o)of (12) mm, (d_i) of (10)mm and (L) of (160)mm, made from mild iron. 22 Rectangle fins made from aluminum with dimension of 70 x40 mm with thickness (t) 1 mm fixed with radially on the outer surface of cylinders. Using voltage regulator for electrical power supply system, variac, digital multimeter, clamp meter, data logger and thermocouples, two thermocouples for each cylinder to measure pipe and fins temperature and two thermocouples for air inlet and outlet temperature of the duct. Hot wire used to measure air velocity at inlet and outlet of duct. The cylinders heated by electric resistance is type (Ni-Cr), used (MnO2) with material has higher thermal conductivity as insulator for electrical to separate between the tube and electric coil. Four ceramic covers of 8mm thickness are utilized to minimize heat lost from the ends of the cylinder.

The variac. type (TDGC2), used to control the electric power supply for cylinders by a variable voltage. (Variac) (4 Amperes) and the volt from (0-300) Volt that supplies with a constant thermal flux to cylinders

The Multimeter. Used (VC890D), used to voltage supply measurement to heater to obtain the thermal flux required on a cylinder by joint to output lines from Variac in the parallel

The Clamp Meter. The clamp meter (DT200) , used to current measurement which supply to the heater, to obtain **a** power supply which calculated from the current and the voltage .

Temperature Data Logger. Data-logger type (TC-08) Pico Technology Limited Company, England, it contains 8-input points. It is utilized to obtain the signal coming from thermocouples and sending it again to computer to record the temperature for the selected point by Pico log 6 beta software program the range for reading from (-270 to +1820) o C

Hot wire. Hot wire used type (GM8903), its used to measure air velocity at the top of air duct outlet of the duct.

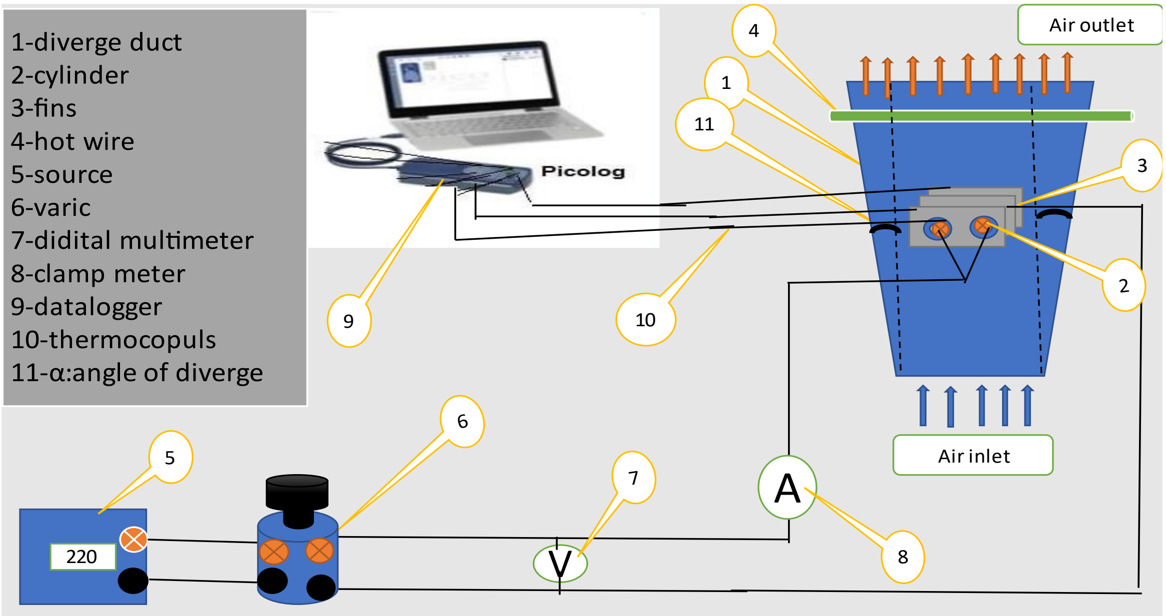

FIGURE 1. schematic diagram of the experimental set

Thermocouples Installation and Distribution

In this study, thermocouples types K [Ni-10% (+) Cr versus Ni-5% (-) aluminum silicon] they are used to measure the temperatures of the cylinders and fins. The probe of thermocouple wire installed on the cylinder surface and fins by drilling holes (0.5mm) diameter and it depth about(0.5mm) then the thermocouples are inserted inside the holes then fixed by good type of epoxy.

In the work used eight thermocouples: two thermocouples along cylinders, two thermocouples along fins, two thermocouples for air inlet and outlet of the duct, two thermocouples to measure the wall temperature of the duct to calculate the heat losses by the conduction , the thermocouple connect to datalogger by wires then the datalogger send the signal of computer by using (USB) wire and the computer read the temperature for each thermocouple by using program (picolog) .Fig(2) shows the photo picture of experimental set .

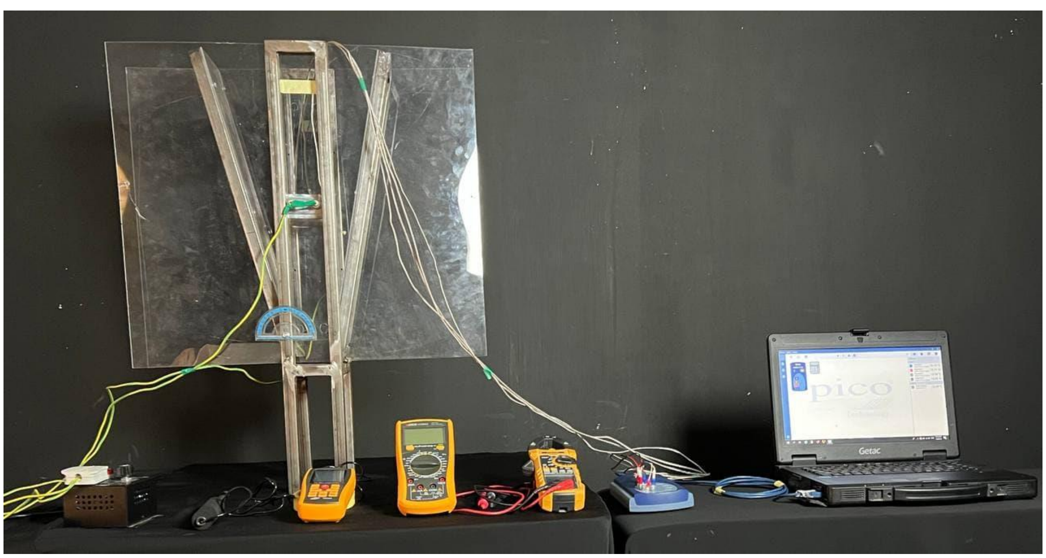

FIGURE 2. Photo of experiment work

MATHEMATIC RELATION

The variables used in the work were the angles of the walls duct from rectangle to diverge, for each angle of diverge has four levels of constant thermal flux (88,177, 390 and 680) W/ m^2, and three fixed locations (top, center and bottom), the steady state of temperature reached after one hour, temperature measured at all points of cylinder, fins and the air temperature, the voltage and the current measured also.

Computation

The thermal flux calculates from

$Qg = V * I$ (1)

$Qg = Qconv + Qcond + Qrad$ (2)

$Qcond = Uwall * Awall * (Tw - T_{amb})$ (3)

$Qcond$ (neglected) because it was found very small due to low thermal conductivity of Pyrex glass.

$Qconv = h * At * (Ts - T_\infty)$ (4)

$Qrad = \sigma * A * \varepsilon * (Ts4 - T_\infty 4)$ (5)

$At = Ab + Af$ (6)

$A_b = \pi * d_o * (L - N * t) * 2$ (7)

$A_f = \left[(a * b) - 2\left(\frac{\pi D^2}{4}\right) \right] * 2 + [2(a*b) * t* n)]$

the Nusselt number and the Rayleigh number calculate from:

$Nu = h * D/k$ (8)

$D = \frac{4A}{P} = \frac{4ab}{2(a+b)} = \frac{2ab}{(a+b)}$ (9)

$h_{ave} = \frac{Q_{conv}}{A_t * \Delta T}$ (10)

$Ra = Gr.Pr = \frac{g.\beta.\Delta T.D^3}{\nu_f^2} * \frac{\mu_f\, Cp_f}{K_f} = \frac{g.\beta.\Delta T.D^3}{\nu.\alpha}$ (11)

$\nu = \frac{\mu}{\rho}$ (12)

$\alpha = \frac{K}{\rho C_p}$ (13)

Air properties.

$T_f = \frac{T_S + T_\infty}{2}$ (k) (14)

$T_S = \frac{T_P + T_f}{2}$ (k) (15)

$T_\infty = \frac{T_{a1} + T_{a2}}{2}$ (k) (16)

volumetric expansion coefficient.

$\beta = \frac{1}{T_f}$ $(\frac{1}{K})$ (17)

Procedure

The experimental work was in the large room with no air currents, at average ambient temperature from (18 – 22) C and pressure at (1atm). A current experiment work interest with studying behavior of a heat transfer from two finned cylinders, the cylinders fixed in horizontal array fixed in variable air duct, three fixed positions (top, center and bottom) used after each value of the heat flux (88,177,390 and 680) W/ m2 and angle (0°,3°,5°,7°,10°,15°). The temperature measured after reach to study state.

THE RESULT AND DISCUSSION

The experimental work was done with variables, thermal flux (88,177,390 and 680) W/ m^2, Rayleigh number values (15x103to 14 x104), and walls angles from the y axis (0°-15°). Three locations (top, center, and bottom). Nusselt number of free convection of cylinder was calculated for each of the above cases and plotted for each model Test. Figure 3, shows the change in the average temperature of the cylinders, fins, and outlet air temperature with the inclination angles at (q•=88 W/ m^2) at the top location and the air velocity at the duct exit. It shows that the average temperature for cylinders reached a maximum value at an angle (0◦), then increasing with the increase of the inclination angle of walls until reached to the maximum value at the angle (3◦) this mean reduce heat transfer between the air and the cylinders. Moreover, the velocity and average temperature of the outlet of the air duct will decrease with the increase of the inclination angle of the walls until it reaches the minimum value at an angle (3◦) because the cross-section area of the outlet of the duct increase, hence the Nusselt number to minimum value at 3◦ and keep constant of other angles as shown in figure 4, at top. For the same location top, when increasing heat flux to 680 W/ m^2, the average temperature for the cylinder reaches the maximum value at an angle (10◦), because the outlet velocity decrease due to increase in cross-section area, so Nusselt number will be minimum at angle (10◦) as shown in figures (5, 6), so the Nusselt number will decrease about 15 % at q•= 88 W/ m^2 for angle (3◦) and decrease about 25 % at q•= 680 W/ m^2 for angle (10◦). In figure 7, also shows the change in the average temperature of the cylinders, fins, and temperature of the outlet air with the inclination angles at (q•=88 W/ m^2) at the center location as well as the air velocity at the duct exit. It shows that the average temperature for cylinders reached a maximum value at an angle (0◦), then increase with the increase of the inclination angle of walls until reached maximum value at the angle (3◦) this mean reduce the heat transfer between the air and the cylinder. Moreover, the velocity and average temperature of the outlet of the air duct will decrease with the increasing the inclination angle of the walls until it reaches the minimum value at an angle (3◦) because the cross-section area of the outlet of the duct increase, hence the Nusselt number to minimum value at 3◦ and keep constant of other angles as shown in figure 8, at the center. For the same location center, when increasing heat flux to 680 W/ m^2, the average temperature for the cylinders reaches the maximum value at an angle (10◦), because the outlet velocity decrease due to increase in cross section area, so Nusselt number will be minimum at angle (10◦) as shown in figures (9, 10), wherefore the Nusselt number will decrease about 21.6 % at q•= 88 W/ m^2 for angle (3◦) and decrease about 13.7 % at q•= 680 W/ m2 for angle (10◦). For the bottom location for the same reason, shows the Nusselt number will decrease about 50 % at q•= 88 W/ m2 for angle (5◦) and decrease about 21 % at q•= 680 W/ m^2 for angle (15◦), as shown in figures (11-15). The result for q•= 177 W/ m^2 , q•= 390 W/ m^2 are in between the value of q•= 88 W/ m^2 and q•= 680 W/ m^2.

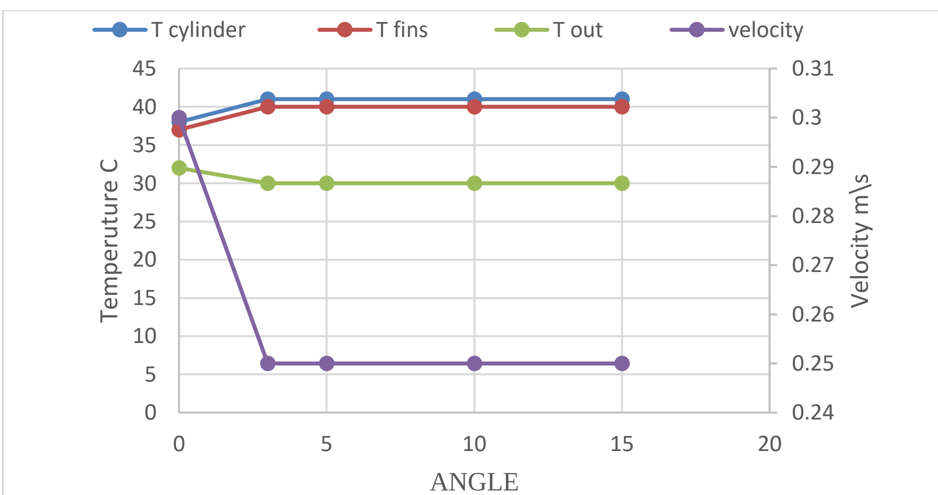

FIGURE 3. temperature and velocity change, against deferent angle for diverge, top, q=88 W/m^2

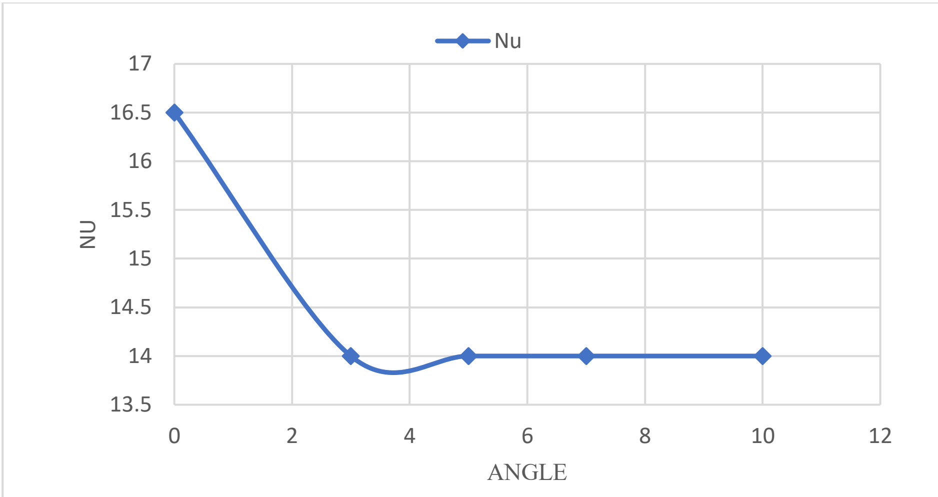

FIGURE 4. Nusselt number change, against deferent angle for diverge, top, q=88 W/m^2

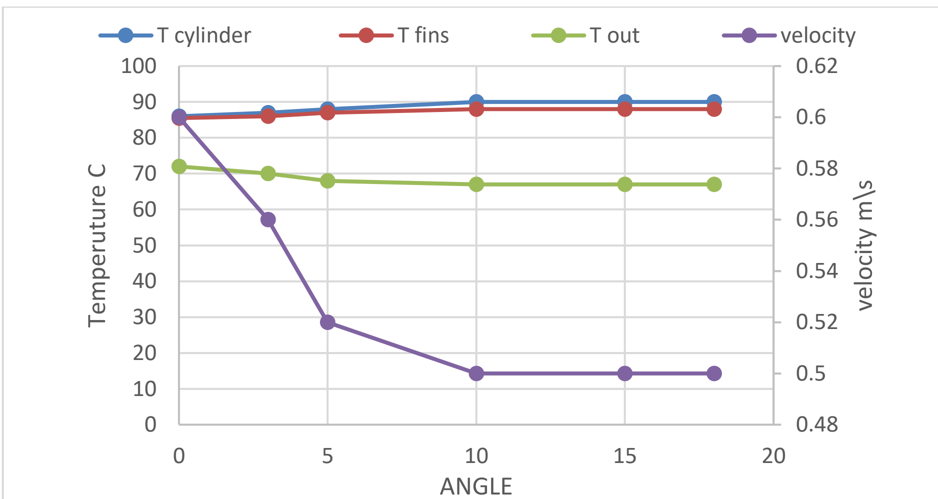

FIGURE 5. Temperature and velocity change, against deferent angle for diverge, top, q=680 W/m^2

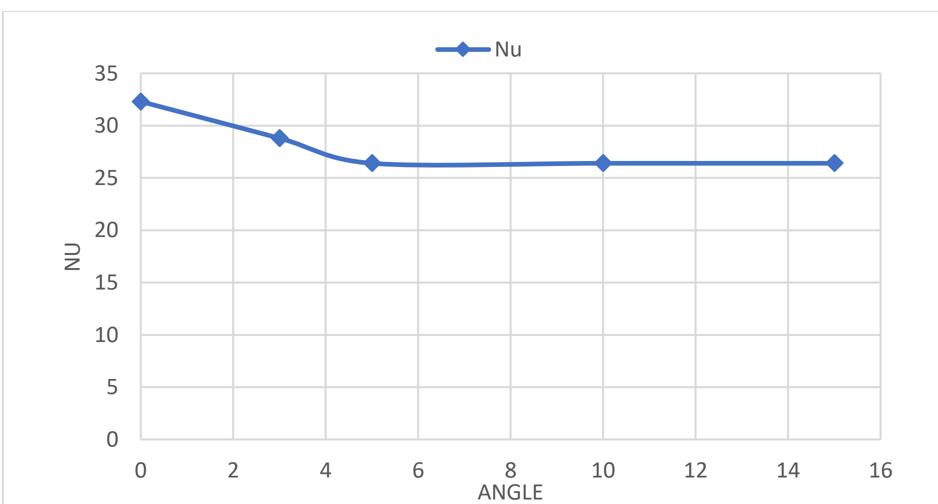

FIGURE 6. Nusselt number change, against deferent angle for diverge, top, q=680 W/m^2

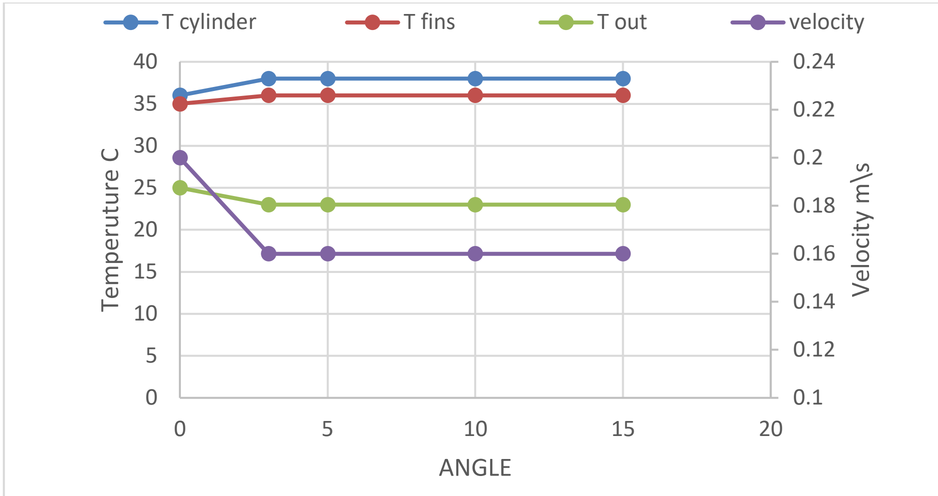

FIGURE 7. temperature and velocity change, against deferent angle for diverge, center, q=88 W/m^2

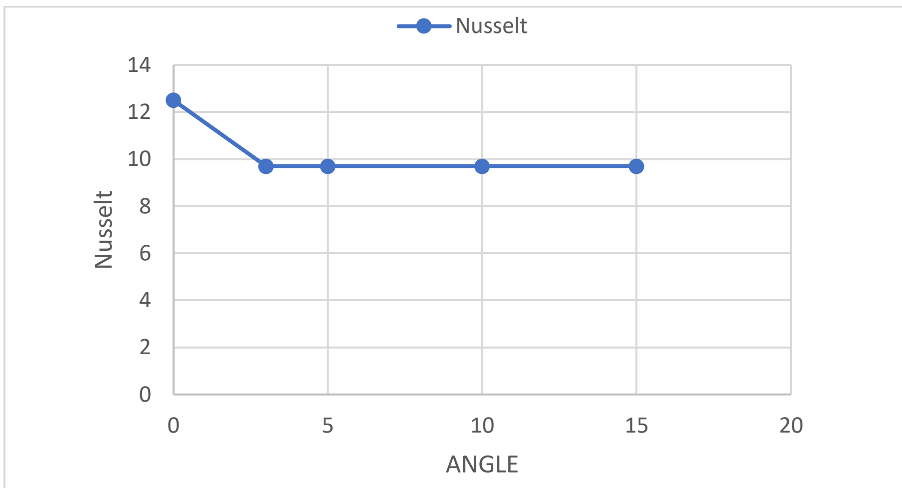

FIGURE 8. Nusselt number change, against deferent angle for diverge, center, q=88 W/m^2

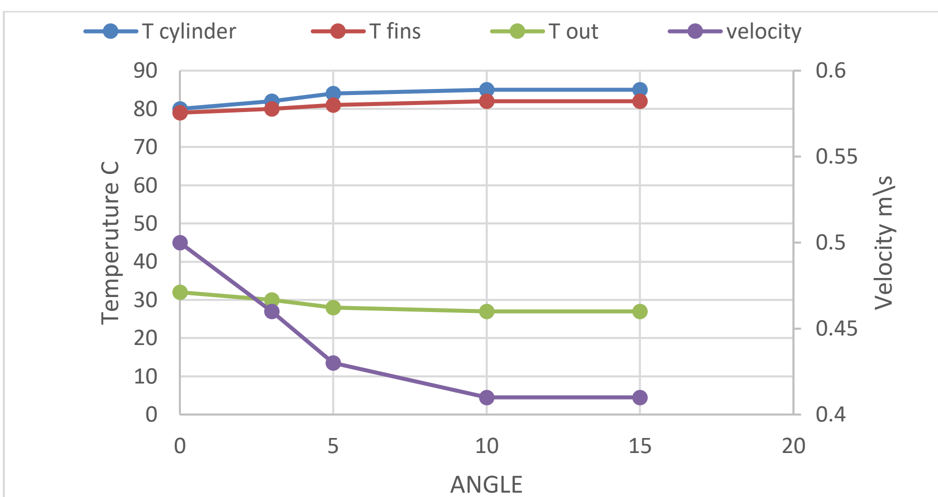

FIGURE 9. temperature and velocity change, against deferent angle for diverge, center, q=680 W/m^2

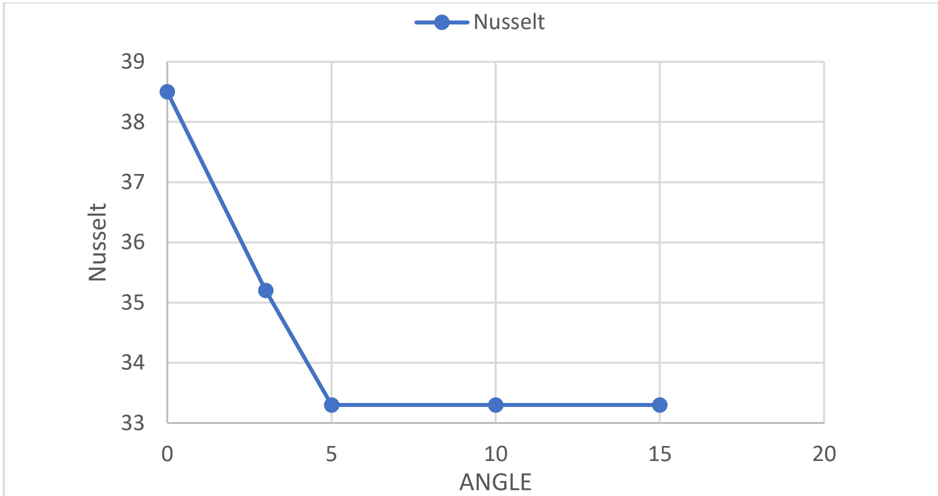

FIGURE 10. Nusselt number change, against deferent angle for diverge, center, q=680 W/m^2

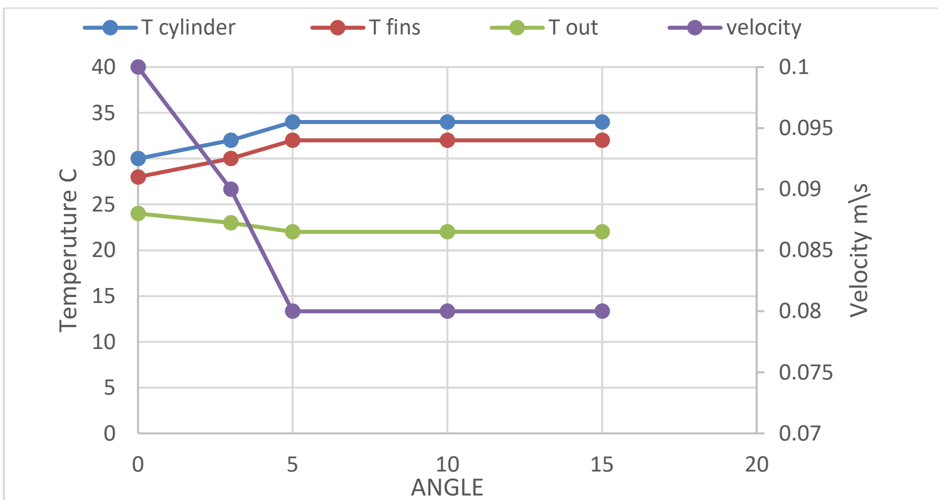

FIGURE 11. temperature and velocity change, against deferent angle for diverge, bottom, q=88 W/m^2

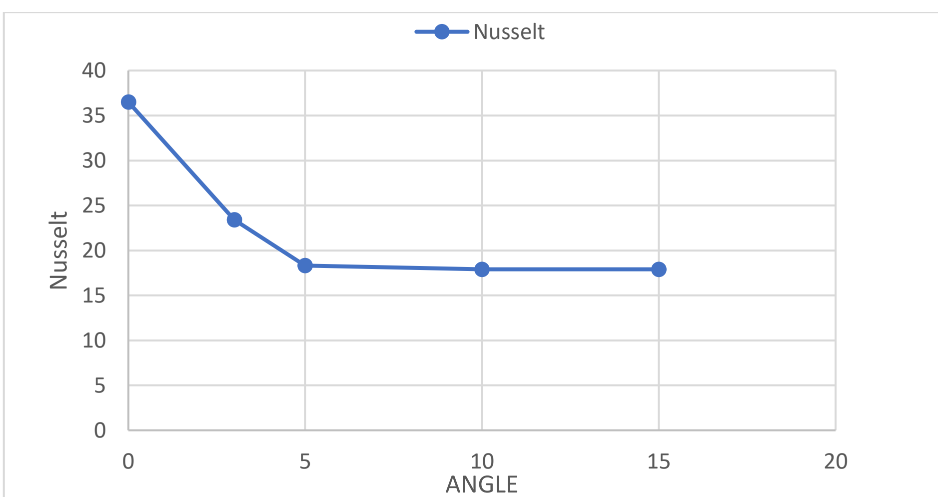

FIGURE 12. Nusselt number change, against deferent angle for diverge, bottom, q=88 W/m^2

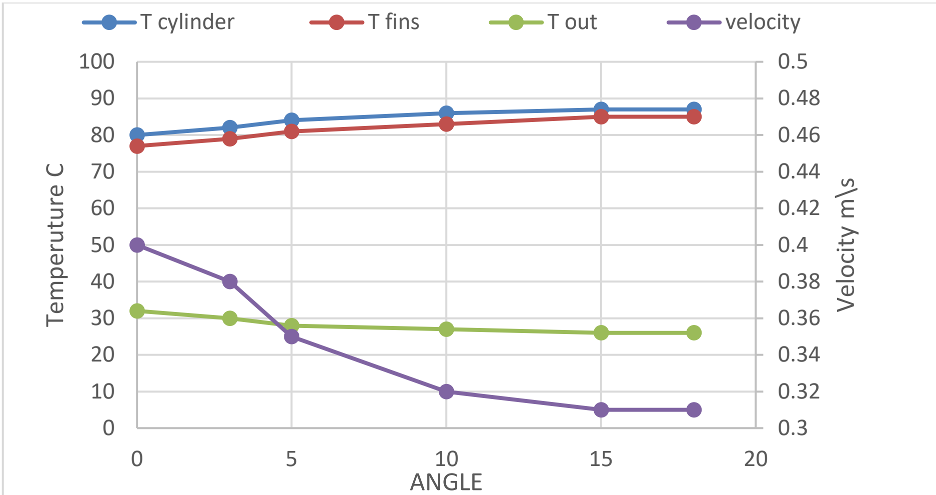

FIGURE 13. temperature and velocity change, against deferent angle for diverge, bottom, q=680 W/m^2

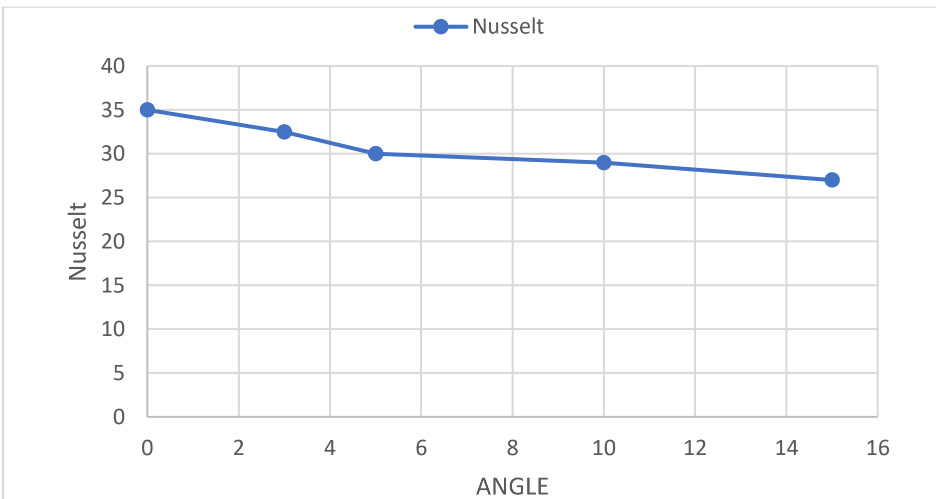

FIGURE 14. Nusselt number change, against deferent angle for diverge, bottom, q=680 W/m^2

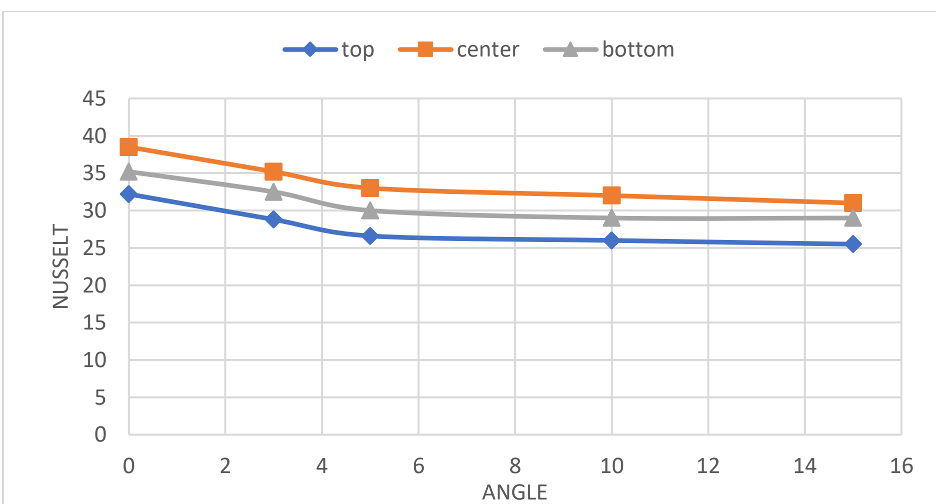

FIGURE 15. Nusselt change, against deferent angle for diverge top, center and bottom, q=680 W/m^2

CONCLUSIONS

- For the top location, the value of the Nusselt number decreased by 15 % at q•= 88 W/ m^2 when increasing the value of angle from 0° to 3° and increased by 25 % at q•= 680 W/ m^2 when decreased the value of angle from 0° to 55°
- For the center location, the value of the Nusselt number decreased by 21.6 % at q•= 88 W/ m^2 when increasing the value of angle from 0° to 3° and increased by 13.7 % at q•= 680 W/ m^2 when decreased the value of angle from 0° to 10°.
- For the bottom location, the value of the Nusselt number decreased by 50 % at q•= 88 W/ m^2 when increasing the value of angle from 0° to 5° and increased by 21 % at q•= 680 W/ m^2 when decreased the value of angle from 0° to 15

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