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Natural Convection Heat Transfer Optimization From A Horizontal Finned Tube

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***Abstract*—In this study natural convection heat transfer in annular fin arrays mounted on a horizontal tube was experimentally investigated. The aim of this study was to determine the effects of geometric parameters like fin diameter, fin spacing and base-to-ambient temperature difference on the heat transfer performance of fin arrays, and to find optimum parameters that maximize the heat transfer rate. Keeping the fin thickness fixed at 1.5 mm, fin diameter was varied as 40, 60, 90, & 120 mm and fin spacing was varied as 3.5, 4.5, 8.5, 13.5 & 28.5 mm so that 20 sets has been investigated. The base-to-ambient temperature difference was also varied independently and systematically with the power supply to heater ranging from 10 W to 125 W. The results have shown that the convection heat transfer rate from the fin arrays depends on the fin diameter, fin spacing and base-to-ambient temperature difference. In addition, for every fin diameter, for a given base-to-ambient temperature difference, there is an optimum value for the fin spacing for which the heat transfer rate from the fin array is maximized.**

Keywords*—*Heat Transfer; Natural Convection; Optimization; Annular Fins; Horizontal Tube.

Nomenclature

A Area, m^2
Do Outer diameter of the horizontal cylinder, m.
Df Fin diameter, m.
g Gravitational acceleration, m/s^2.
Gr Grashof number.
h Convection heat transfer coefficient, W/ (m^2. K).
h_{exp} Experimental convection heat transfer. coefficient of horizontal cylinder, W/(m^2. K).
K Thermal conductivity, W/ (m. K).
L Length of the horizontal cylinder, m.
N Number of fins.
Nu Nusselt number.
Pr Prandtl number.
$\dot{Q}$ Power input to the heater, W.
$\dot{Q}_c$ Convection heat transfer rate, W.
$\dot{Q}_r$ Radiation heat transfer rate, W.
Q_0 Total heat transfer rate from the horizontal cylinder, W.
$(\dot{Q}_0)_c$ Convection heat transfer rate from the horizontal cylinder, W.
$(\dot{Q}_0)_r$ Radiation heat transfer rate from the horizontal cylinder, W.
$\dot{Q}_{ri}$ Radiation heat transfer from surfaces, W.
Ra Rayleigh number.
S Fin spacing, m.
t Fin thickness, m.
T_a Ambient temperature, K.
T_f Film temperature, K.
T_w Wall temperature, K.
V Input voltage to the heater, V.

Greek Letters

α Thermal diffusivity, m^2/s.
β Thermal expansion coefficient, 1/K.
ε Emissivity.
ν Kinematic viscosity, m^2/s.
σ Stefan-Boltzmann constant, W/ (m^2.K^4).
ΔT Base to ambient temperature difference.

I. INTRODUCTION

Heat removal in an efficient way is necessary in order to maintain reliable operation of different applications. The operation of many engineering systems results in the generation of heat. This unwanted heat can cause serious overheating problems and sometimes leads to failure of the system. The heat generated within a system must be dissipated to its surrounding in order to maintain the system at its recommended working temperatures and functioning effectively and reliably. This is especially important in modern electronic systems, in which the packaging density of circuits can be high. In order to overcome this problem, thermal systems with effective emitters as fins are desirable. This has made the design of cooling systems more significant, and thermal management become an issue. Newton's law of cooling states that, by keeping the power input fixed and without exceeding a maximum temperature, the convection heat transfer rate from a surface can be increased either by increasing the convection heat transfer coefficient (h) or the surface area (A) or both of them. In the equation $\dot{Q}_c = h\ .A\ .\Delta T$, the convection heat transfer coefficient can be increased by either create forced flow over the surface, such forced convection is effective, extra space will be needed to accommodate a fan which causes additional initial and operational costs. Therefore, forced convection is not always preferable. On the other hand, use of a better fluid but this is not an economical solution. Use of a liquid as a coolant, for example, requires a heat exchanger, a pump, piping and other instruments. However, air is inexpensive and often readily available. Since it is not practical and economical to increase the heat transfer coefficient, increasing and optimizing the heat transfer area is widely preferred as the simplest and preferable method to enhance heat transfer [1]. Using extended surfaces in free convection is often used for economical, noiseless and maintenance free for many engineering applications such as cooling of electronic equipment, solar energy, heat exchangers, air conditioning, refrigeration, gas turbines, compressors and nuclear reactor fuel element energy etc. The only controllable variable affecting the convection heat transfer rate is the geometry of the fins. fins are increasing the surface area so increasing the heat transfer, but they may laminate the available area for the air flowing which reduce the heat transfer. Starner and McManus [2] one of the earliest studies about the heat transfer performance of rectangular fin arrays for horizontal and vertical orientation under natural convection. They had used four fin arrays sets positioned with base vertical, at 45° and horizontal to determine average heat transfer coefficients. Yuncu and Anbar [3] experimentally investigated natural convection heat transfer from rectangular fin arrays on a horizontal base. The results showed that natural convection heat transfer is governed by the fin spacing, fin height and temperature difference between fin base and surroundings. A correlation was presented. Lohar, G P [4] experimentally studied heat transfer rate from heated horizontal rectangular fin array under natural and forced convection. For natural convection the maximum heat transfer coefficient was obtained between the fin spacing 14 mm to 16 mm. and for forced convection the maximum was obtained between the fin spacing of 12 mm to 14 mm. Ranjan Kumar and Rama Krishna [5] experimentally investigated heat transfer characteristics of annular fins under forced convection. They studied the heat transfer characteristics of fin with 11mm diameter without circumferential fins, fin with 31mm diameter and annular fins with 31 mm diameter under forced convection at different power inputs and Reynolds number and found that The base temperature for annular fins is reduced by 30% when compared to fin with diameter 11 mm due to increase in surface area about 40%, and the base temperature for annular fins is reduced by 10% when compared to fin with diameter of 31mm due to decrease in surface area about 41%. M.A. Mokheimer [6] investigated the performance of annular fins with different profiles subjected to variable heat transfer coefficient. Nagarani, N Mayilsamy, K [7] experimentally investigated the analysis of heat transfer on annular circular and elliptical fins. Yildiz and Yuncu[8] experimentally investigated the performance of annular fins on a horizontal cylinder under natural convections. Fin diameter was varied from 35-125 mm and fin spacing varied from 3.6-31.7 mm. for that range they found that the optimum fin spacing varied between 7.7 mm and 8.5 mm for the 18 fin arrays were tested. A. Dogan, S. Baysal, and S. Baskaya, [9] Numerically investigated the natural convection heat transfer from an annular fin on a horizontal cylinder. Most of the previous studies [2]–[4], [10]–[22] were conducted for rectangular fin arrays Because of these studies, there are many data available in the literature for the flat fins. However, limited data are available for an array of annular fins, so in this experimental study, a wide range of annular fin configurations mounted on a horizontal cylinder was investigated and the effect of fin geometry like fin diameter fin spacing and base-to-ambient temperature difference on the heat transfer performance of fin arrays was inducted.

II. EXPERIMENTAL EQUIPMENT AND INSTRUMENTATION

The flowchart of the experimental setup are showing in Fig. 1. The effective length of the horizontal cylinder is 300 mm. The outer diameter of the cylinder is 26 mm. The cylinder material was chosen as Aluminum because of its high thermal conductivity. A chrome wire heater, which was rated for 500 W and 220 V, AC, passes through the inside of the cylinder. The cylinder was filled with aluminum powder material in order to provide uniform temperature distribution to the cylinder. The cylinder is mounted on a steel supporting-frame, which is 1.2 m high above the ground to prevent ground effects. The ends of the cylinder are insulated with fiber glass wool insulation in order to prevent the side effects and the heat loss from the ends. In addition, the experiments were performed in a closed large room with constant temperature, free of air currents to avoid any forced convection. Table 1 lists the dimensions of the 20-fin arrays were tested in this study. The cylinder dimensions and the fin thickness were kept fixed during the experiments. We have selected aluminum fins because of its higher thermal conductivity, low radiative emissivity, and easy machinability. During the experiments, both electrical power measurements and temperature measurements at various points on the surface of the finned horizontal cylinder were indicated.

A. Electrical Measurements

The electrical power is supplied through a regulated a-c supply. The output of supply was fed to two parallel variac to adjust the power suppling to the heater and to control the heating level. The power supply is measured by calibrated Wattmeter; also, Voltmeter and Ammeter were used to double-check the Wattmeter readings. The thermocouple and power circuits are shown in Fig. 1.

B. Temperature Measurements

The ambient temperature was measured with a digital calibrated thermometer. The cylinder surface temperature was measured at seven points by using copper-constantan thermocouples and the average of those seven readings may be taken as the surface temperature. The distribution of the thermocouples are shown in Fig. 2.

Set No.	Fin Diameter D (mm)	Fin Spacing S (mm)	No. Of Fins N
1	120	3.5	60
2	120	4.5	50
3	120	8.5	30
4	120	13.5	20
5	120	28.5	10
6	90	3.5	60
7	90	4.5	50
8	90	8.5	30
9	90	13.5	20
10	90	28.5	10
11	60	3.5	60
12	60	4.5	50
13	60	8.5	30
14	60	13.5	20
15	60	28.5	10
16	40	3.5	60
17	40	4.5	50
18	40	8.5	30
19	40	13.5	20
20	40	28.5	10

Table 1. *Dimensions of the fin configurations.*

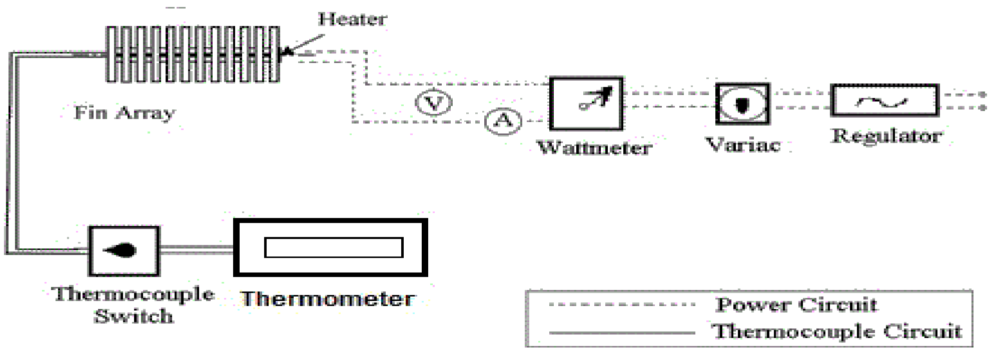

Fig. 1. *Flow chart for the experimental setup and power and thermocouple Circuits*

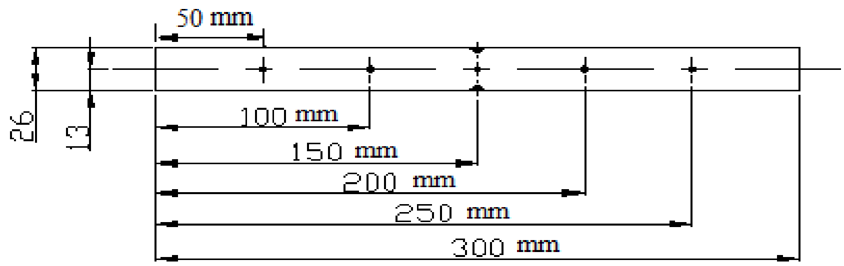

Fig. 2. Thermocouple locations.

Seven holes were drilled through the cylinder to install the thermocouples. Then, the hot junctions of the thermocouples were inserted from the inside of the cylinder, in order not to disturb the flow over the outer surface. The thermocouple junctions were fixed into those holes and covered with fused aluminum. To avoid disturbing the flow past the front surface, temperature measurements was not measured at the fin tips. Since fin material (aluminum) has high thermal conductivity and fin heights are short (maximum fin height is 50 mm), it was assumed that the temperatures along the fin and at the fin tip did not vary significantly from the cylinder surface.

III. SETUP CALIBRATION

Before starting the experiments, the set-up was calibrated and verified by determining the convection and radiation heat transfer rates from a horizontal bare cylinder without fins. The convection heat transfer rate was determined by subtracting the radiation heat transfer rate from the total heat transfer rate to the surroundings, since the convection heat transfer rate cannot be measured directly. The cylinder was heated with 10 power inputs. For each of the power inputs, the temperature of the horizontal cylinder $T_w$the ambient temperature, T_a and the power input were measured at steady state conditions. Cylinder temperature was measured by using seven thermocouples that were placed on the cylinder surface as illustrated in Fig. 2. Steady state condition was assumed to be reached when readings taken at thirty-minute intervals did not vary by more than half degree. The power input to the heater is equal to the total heat transfer from the horizontal cylinder to the surroundings at steady state conditions. Heat is transferred to the surroundings by natural convection and radiation. Radiation heat transfer rate from the horizontal cylinder was calculated by considering the environment as a blackbody at ambient temperature, as

$$\left(\dot{Q}_0\right)_r = A.\varepsilon.\sigma.\left(T^4{}_w - T^4{}_a\right) \tag{1}$$

Emissivity of the cylinder was chosen from the tables as 0.04. Then, the convection heat transfer rate from the cylinder was evaluated as,

$$\left(\dot{Q}_0\right)_c = \dot{Q}_0 - \left(\dot{Q}_0\right)_r \tag{2}$$

The heat transfer coefficient based on the outer surface area of the cylinder was evaluated as,

$$h_{exp} = \frac{(\dot{Q}_0)_c}{A.(T_w - T_a)} \tag{3}$$

Rayleigh and Nusselt numbers were evaluated according to the experimental data for the bare cylinder as,

$$Ra = \frac{g.\beta.d^3.(T_w - T_a)}{\nu.\alpha} \tag{4}$$

$$Nu_{exp} = \frac{h_{exp}.d}{k} \tag{5}$$

All of the physical properties necessary to evaluate Rayleigh and Nusselt numbers have taken at film temperature $T_f = (Tw + Ta)/2$ and d is the outer diameter of the cylinder.

Experimental Nusselt numbers have compared with the Nusselt numbers that have evaluated by using correlations for horizontal cylinders from the literature. The correlations that used for the comparison are, Churchill and Chu's relation [21],

$$Nu_D = \left[0.6 + \frac{0.387.(Ra_D)^{1/6}}{\left[1+\left(\frac{0.559}{Pr}\right)^{9/16}\right]^{8/27}}\right]^2$$

(6)

And Morgan's relation [22]

$$Nu_D = 0.48.{Ra_D}^{0.25} \tag{7}$$

The above correlations and experimental Nusselt numbers versus Rayleigh numbers were plotted in Fig. 3.

It is obvious from Fig. 3 that experimental data are in an excellent agreement with the correlations from the literatures, so the setup is ok and then the fin arrays now attached.

IV. EXPERIMENTAL PROCEDURE FOR FIN ARRAYS

1. Fin arrays has been assembled and tightly fitted to the aluminum horizontal cylinder then 20 sets were experimentally investigated, thermocouples and heaters were attached and placed in position, and connections were made as per requirements.
2. The heater input was adjusted with the help of voltage regulator and the fin arrays was heated about 4-5 hrs. To reach the steady state operation.
3. The temperatures of surface of the tube and ambient temperature were recorded at the time intervals of 30 min. and steady state condition was assumed to be reached when the difference between two successive readings of each thermocouple is less than 0.5℃.
4. The temperature readings are taken from the seven attached thermocouples for the different heat inputs (25, 50, 75, 100 and 125 Watts) for all the fin arrays.

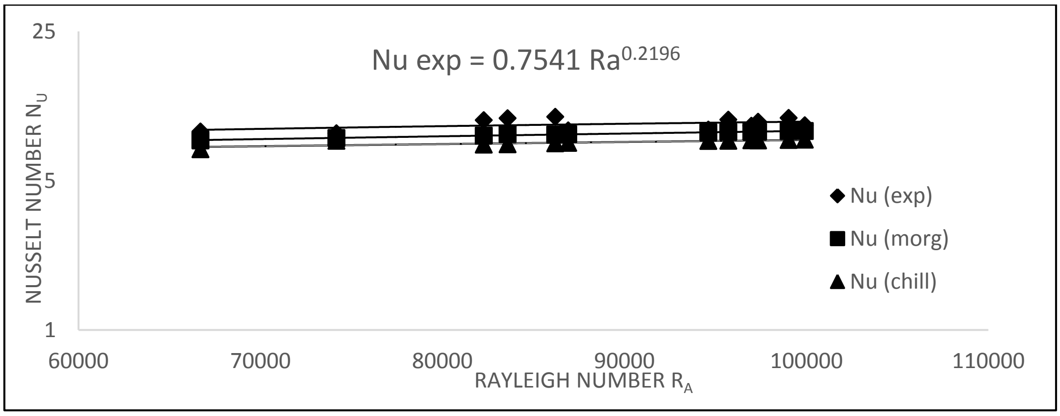

Fig. 3. *Nusselt against Rayleigh for Experimental, Equations 6, and Equation 7.*

The heat loss by radiation from the fin arrays was calculated by the following equation,

$$\dot{Q}_r = 2.N.\dot{Q}_{r1} + N.\dot{Q}_{r2} + N.\sigma.\varepsilon.\pi.d.t.(T^4{}_w - T^4{}_a) \quad (8)$$

Where $\dot{Q}_{r1}$, is the net radiation heat transfer rate from a surface of a fin and $\dot{Q}_{r2}$ is that from the base cylinder between two adjacent fins. D is the fin diameter, N is the number of fins and t is the fin thickness. $\sigma = 5.67 * 10^{-8}$ Is the Stefan-Boltzmann constant and $\varepsilon = 0.04$ is the emissivity of the tube and fin surfaces. The convection heat transfer rate from the fin arrays was then evaluated as,

$$\dot{Q}_c = \dot{Q} - \dot{Q}_r \quad (9)$$

V. RESULTS AND DISCUSSION

Natural convection heat transfer from annular fins around as horizontal cylinder was experimentally investigated and 20 different of fin configurations were tested. The results were utilized to reveal the effects of geometric parameters like, fin spacing, fin diameter and the effects of base to ambient temperature difference on the steady state heat transfer rates from annular fins around a horizontal tube. The results shows that the convection heat transfer is affected by fin spacing fin diameter and base to ambient temperature difference. Fig. 4 to Fig. 7 showing the effecting of base to ambient temperature difference on the convection heat transfer rate for a given fin diameter and as a function of fin spacing, the figures are plotted for fin diameters 40, 60, 90, and 120 mm. Normally, for small values of fin spacing, total number of fin along the horizontal cylinder is increased. Therefore, it is expected that the heat transfer rate increase from the fin arrays. However, as shown in Figures the convection heat transfer rate from fin arrays increases as fin spacing decreases up to a certain value and then it starts to decrease as the fin spacing decreases. This means that there is an optimum value for the fin spacing at a given fin diameter and base-to-ambient temperature difference for which the convective heat transfer is maximum. The reason for this is at a very low fin spacing the boundary layer at both sides of the fin become small hence the convection heat transfer rate decreases. As fin spacing is much increased, the convection heat transfer rate approach to that of horizontal cylinder.

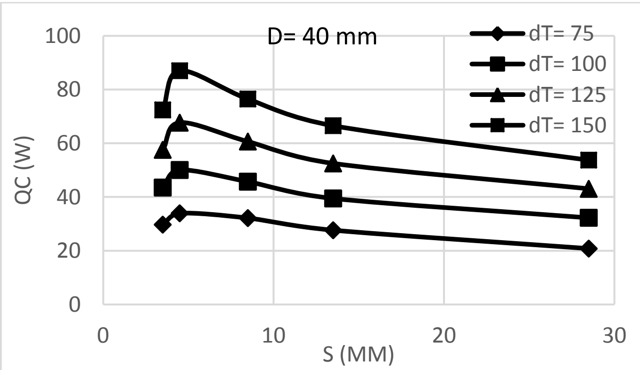

Fig. 4. *Variation of convection heat transfer rate with base-ambient temperature difference at fin diameter 40 mm.*

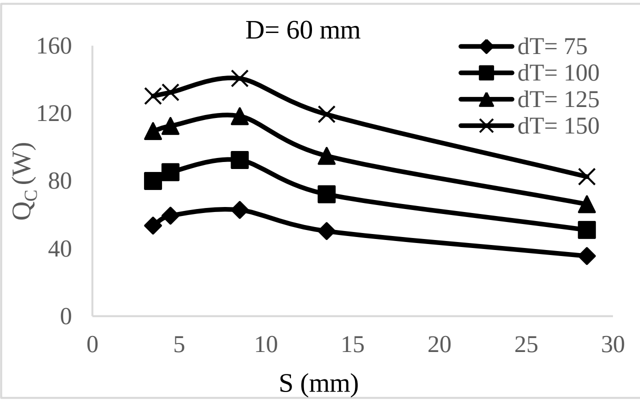

Fig. 5. *Variation of convection heat transfer rate with base-ambient temperature difference at fin diameter 60 mm.*

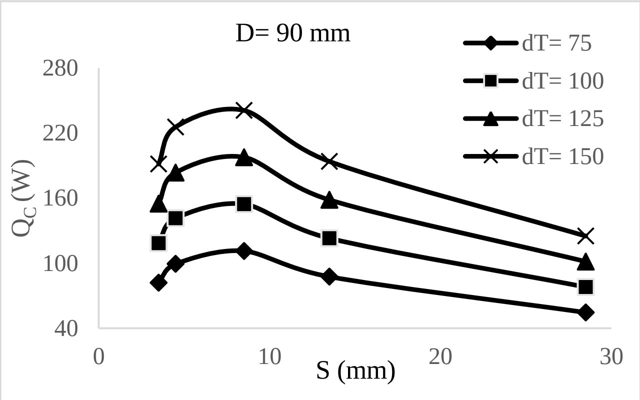

Fig. 6. *Variation of convection heat transfer rate with base-ambient temperature difference at fin diameter 90 mm.*

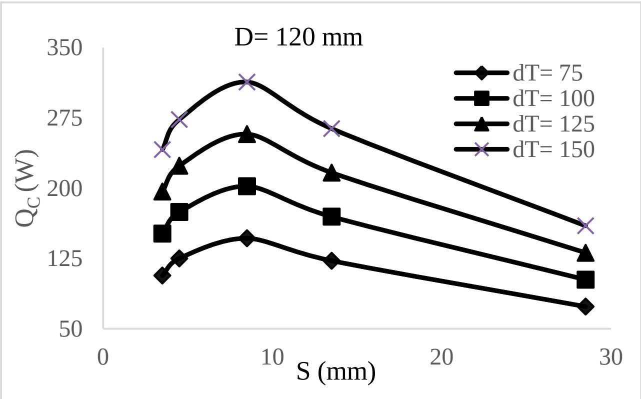

Fig. 7. *Variation of convection heat transfer rate with base-ambient temperature difference at fin diameter 120 mm.*

The convection heat transfer rates from the fin arrays and horizontal cylinder are presented from Fig. 8 to Fig. 12 for different fin spacings, S = 3.5, 4.5, 8.5, 13.5 and 28.5 mm. The results for four fin diameters 40, 60, 90 and 120 mm, and for a horizontal cylinder are plotted against temperature difference between the base cylinder and the ambient. The figures shows that the convection heat transfer rate increases with increasing the fin diameter and with increasing the base to ambient temperature difference.

The convection heat transfer rates from the fin arrays and horizontal cylinder are presented from Fig. 13 to Fig. 16 for fin diameters D = 40, 60, 90, and 120 mm. In the figures, the results for five fin spacings S = 3.5, 4.5, 8.5, 13.5 and 28.5 mm, and for a horizontal cylinder are plotted against base-to-ambient temperature difference. The figures confirming that, the convection heat transfer rate from fin arrays depends on fin diameter, fin spacing and base-to-ambient temperature difference. Showing the same results as figures 4-7, that the convection heat transfer rate from fin arrays increases as fin spacing decreases up to a certain value and then it starts to decrease as the fin spacing decreases. At low temperature differences, the heat transfer rates are closer to each other and tend to diverge at high temperature differences. For all fin arrays, the convection heat transfer rates are higher than that for the horizontal cylinder. As fin spacing is increased, convection heat transfer rates approach to that of horizontal cylinder.

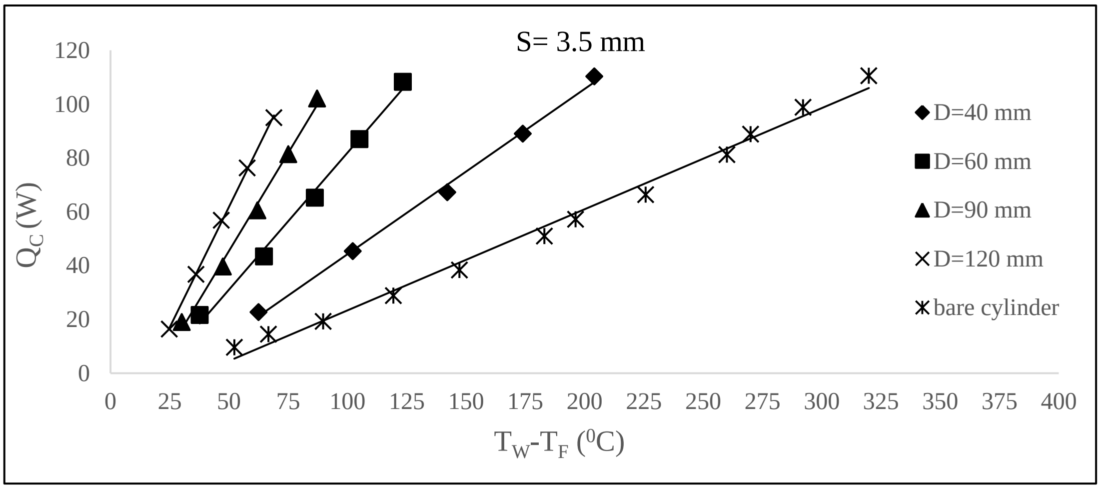

Fig. 8. *Variation of Convection Heat Transfer Rate with Base-to-Ambient Temperature Difference for Different Fin Diameters for Fin Spacing 3.5 mm.*

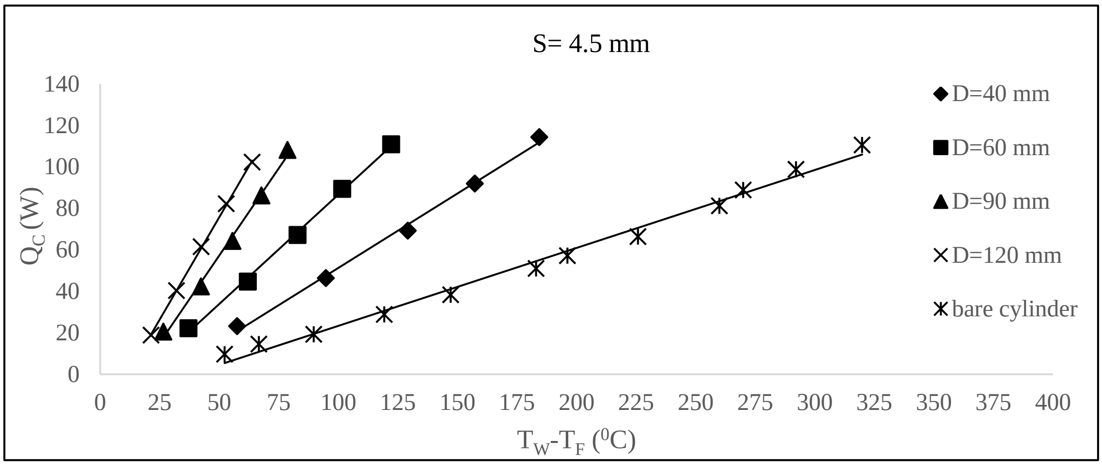

Fig. 9. *Variation of Convection Heat Transfer Rate with Base-to-Ambient Temperature Difference for Different Fin Diameters for Fin Spacing 4.5 mm.*

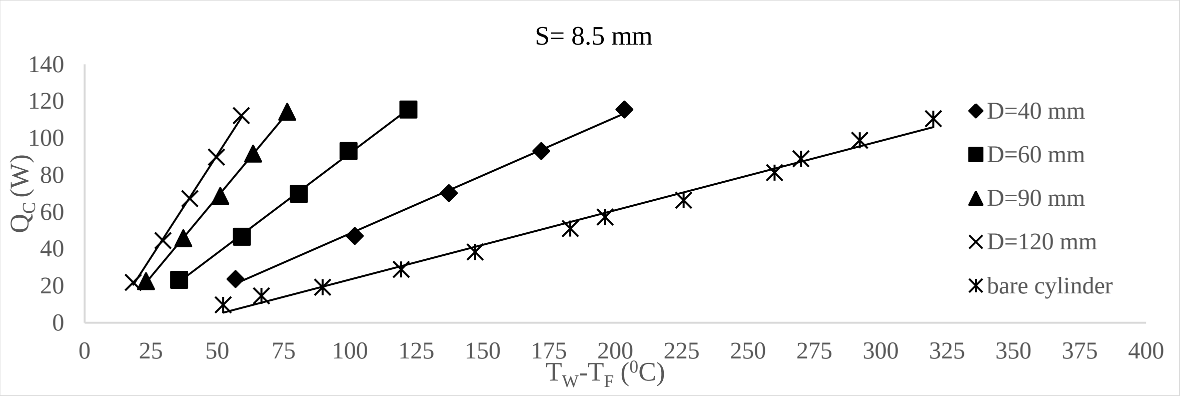

Fig. 10. *Variation of Convection Heat Transfer Rate with Base-to-Ambient Temperature Difference for Different Fin Diameters for Fin Spacing 8.5 mm.*

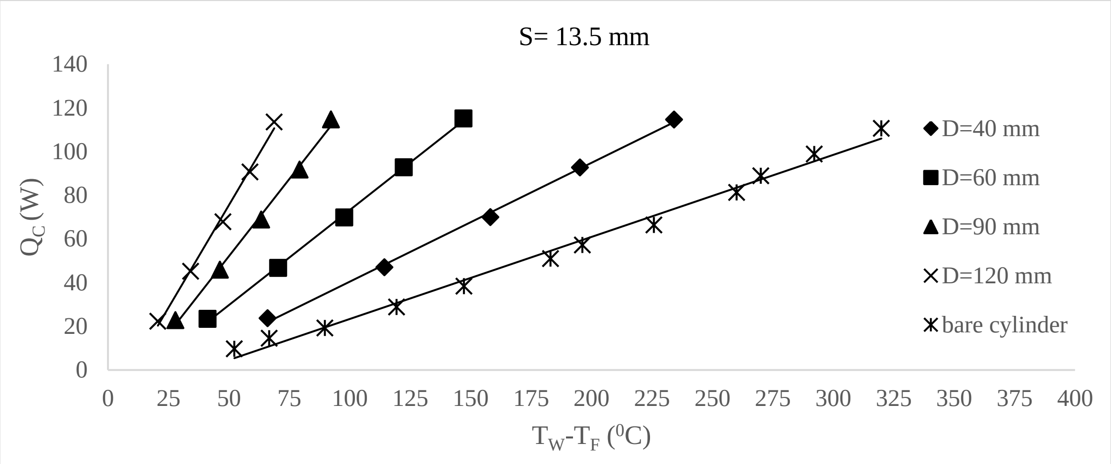

Fig. 11. *Variation of Convection Heat Transfer Rate with Base-to-Ambient Temperature Difference for Different Fin Diameters for Fin Spacing 13.5 mm.*

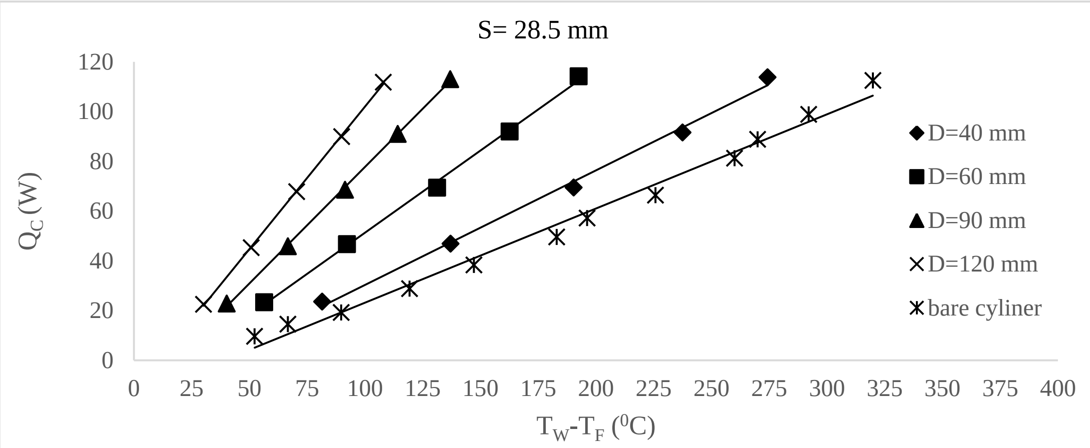

Fig. 12. *Variation of Convection Heat Transfer Rate with Base-to-Ambient Temperature Difference for Different Fin Diameters for Fin Spacing 28.5 mm.*

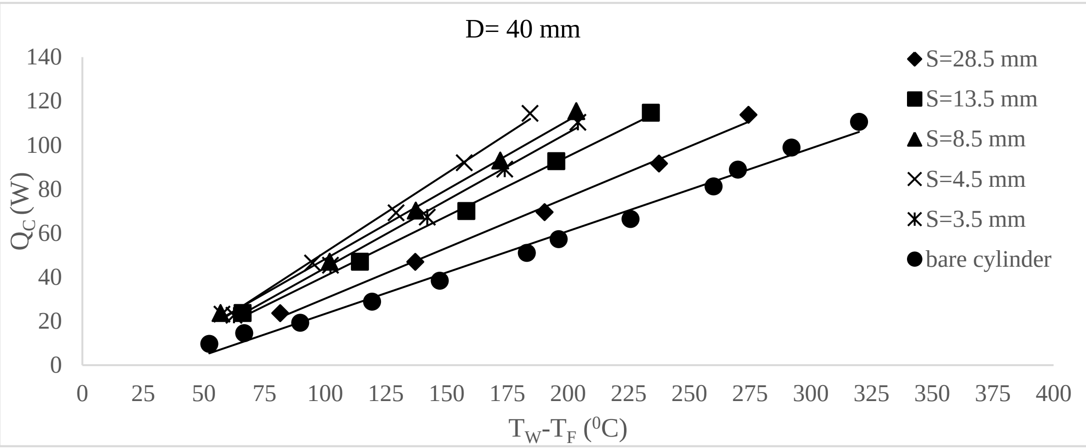

Fig. 13. *Variation of Convection Heat Transfer Rate with Base-to-Ambient Temperature Difference for Different Fin Spacings for Fin Diameter 40 mm.*

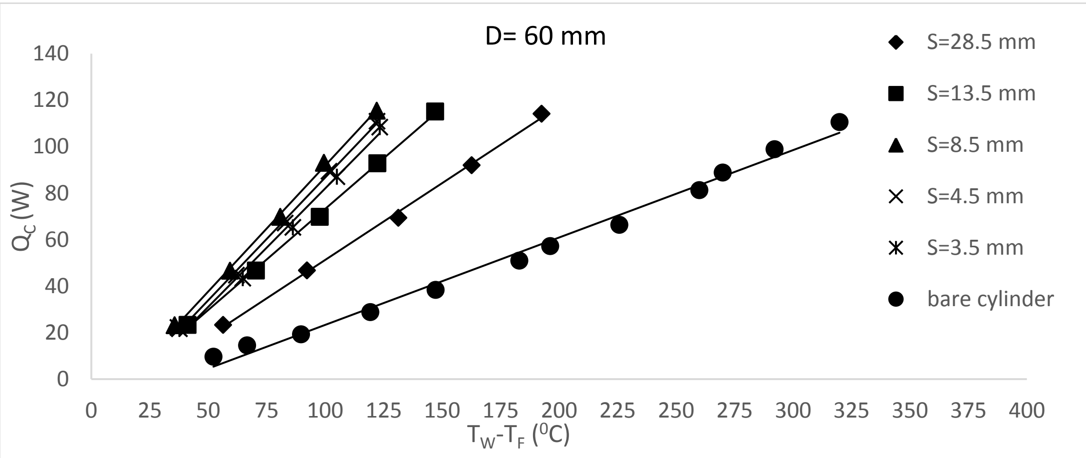

Fig. 14. *Variation of Convection Heat Transfer Rate with Base-to-Ambient Temperature Difference for Different Fin Spacings for Fin Diameter 60 mm.*

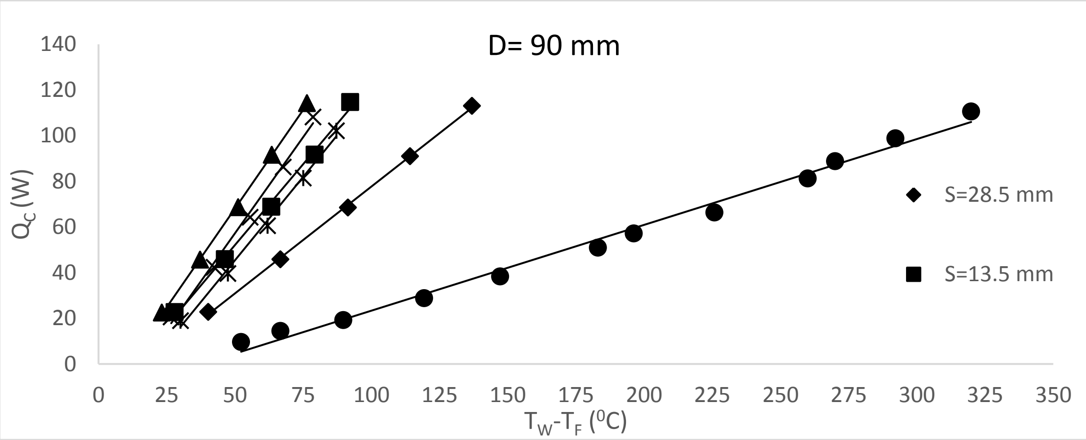

Fig. 15. *Variation of Convection Heat Transfer Rate with Base-to-Ambient Temperature Difference for Different Fin Spacings for Fin Diameter 90 mm.*

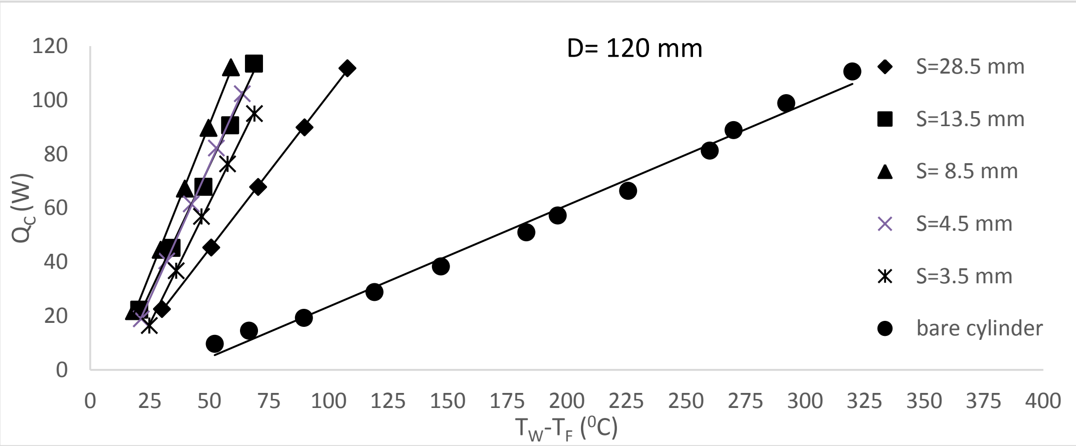

Fig. 16. *Variation of Convection Heat Transfer Rate with Base-to-Ambient Temperature Difference for Different Fin Spacings for Fin Diameter 120 mm.*

Table 2. *The optimum fin spacing S (mm).*

ΔT (^{0}C)	D=40mm	D=60mm	D=90mm	D=120mm
75	6	7.9	8.2	8.4
100	5.9	7.6	8	8.3
125	5.8	7.4	7.7	8.2
150	5.7	7.4	7.5	8.2

VI. CONCLUSION

In this study, the convection heat transfer rate around annular fin arrays on a horizontal tube was experimentally investigated and it was concluded that:

1- To verify that the setup is ok, the convection heat transfer rate from the horizontal bare tube was experimentally investigated, and the experimental results between Nusselt numbers versus Rayleigh numbers showed good agreement with the previous work [27], [28].

2- For all fin configurations, the convection heat transfer rate increases with increasing fin diameter.

3- At low temperature differences, the heat transfer rates are closer to each other and tend to diverge at high temperature differences.

4- For all fin arrays, the convection heat transfer rates are up to 4 times higher than that for the horizontal bare tube. Obviously because of the increase in the heat transfer area by adding fins. As fin spacing is increased, convection heat transfer rates approach to that of horizontal bare tube.

5- For a given fin diameter, decreasing the fin spacing is increasing the number of fins and the surface area, that means the heat transfer rate should be increases, but decreasing the fin spacing to certain value causes decreasing the air flow around the fin arrays therefore decreases on the heat transfer rate, that means that there is an optimum value of fin spacing.

6- For practical engineering applications the optimum fin spacing might be taken as approximately 7.5 mm for fin diameter ranged between 40 to 120 mm. the optimum fin spacing parameters are showing in Table. 2.

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