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Full Length Research Paper

Effects of holes placed on perforated finned heat exchangers at different angles on the Nusselt and Reynolds numbers

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In this study, 6 mm diameter holes were opened on each circular fin on a heating tube in order to increase convective heat transfer. This is different from other studies in the literature. These holes were placed on the circular fins in such a way as to follow each other at the same chosen angle. The holes created turbulence in a region near the heating tube surface on the bottom of the fin. Experiments were performed to analyze the effect of this turbulence on heat transfer and pressure drop. These experiments were performed at six different angular positions in order to determine the best angular positioning. In addition, a perforated finned heater was compared with an imperforate finned heater to observe the differences. When Nu_d values were considered for the same air flow rate and the same $\overline{Re}_d$, the Nu_d value of the heater with holes was found to be 2.43% higher than the Nu_d value of the heater without holes. However, there was a 3.84% decrease in pressure.

Key words: Heat exchanger, heat transfer, convection.

INTRODUCTION

One method to make efficient use of energy is to improve heat transfer without imparting any energy on a fluid. Therefore, attempts have been made to apply methods such as shaping of the heat transfer area and orientation of the flow using different geometric profiles and designs. These methods, which aim at improving heat transfer, have been analyzed analytically and experimentally by many researchers. Souidi and Bontemps (2001) studied the countercurrent gas-liquid flow in plate-fin heat exchangers with plain and perforated fins and compared their results with the experimental results in the literature for rectangular canals. Wierzbowski and Stasiek (2002) analyzed heat transfer by using a liquid crystal technique in a plate-fin heat exchanger. Wang et al. (2001) studied the experimental correlations resulting from heat transfer in a plate-fin heat exchanger under crosscurrent and frictional current conditions, and determined an experimental correlation describing the performance of the air side of the model fin. Leu et al. (2001) also analyzed the performances of the air sides of plate-fin heat exchangers with round and oval shapes. Their results showed that the pressure drop increases with the fin angle while heat transfer and friction increase with fin length. Karabacak (1992) showed the effects of fin parameters on the heat transfer that occurs in the form of radiation and free convection in a horizontal cylindrical heater with rectangular sections and circular fins. Karabacak (1996) also investigated the experimental relationships between heat flow, Nusselt number and temperature difference in a finned heater. Xiao and Tao (1990) studied a naphthalene sublimation technique to examine the heat transfer coefficients and friction factors for wavy plate-fin heat exchangers with different fin lengths. They applied thermal performance assessments for the same pumping power and the same pressure drop and found that both the Sherwood number and the friction factor increase while the mean heat transfer coefficient and pressure

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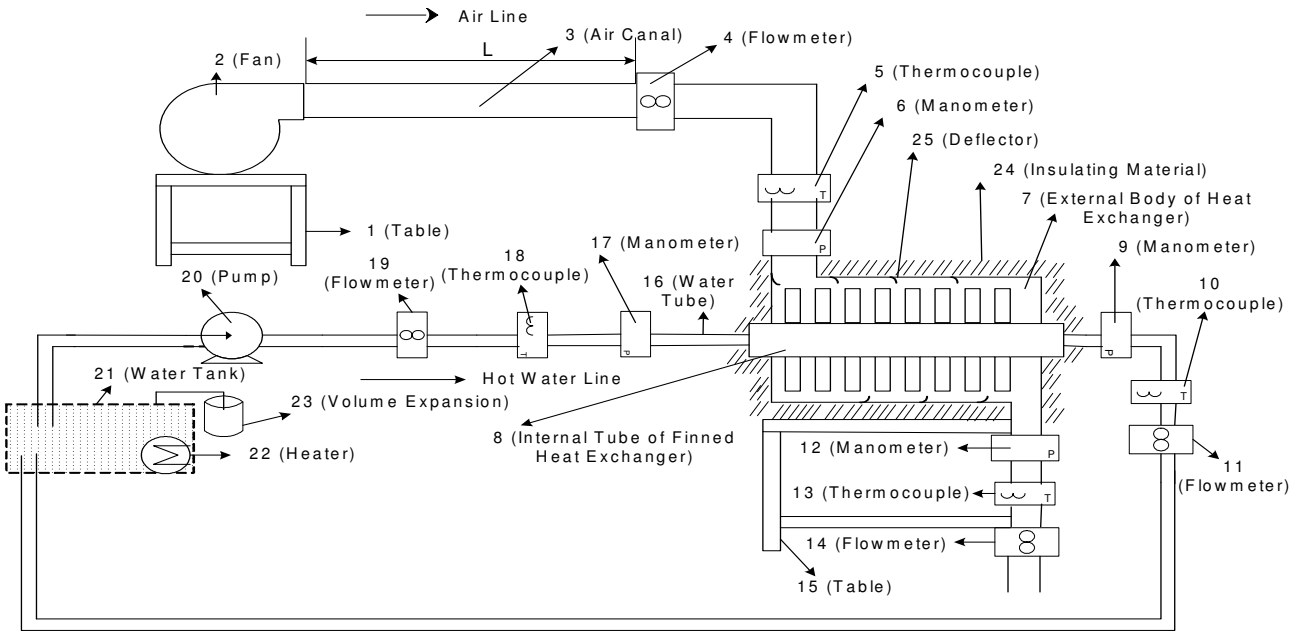


Figure 1. Schematic diagram of the experimental set – up.

drop decrease with an increase in fin spacing. Kayansayan's (1993) analysis found that boundary layer interaction between the fin and the plate leads to a reduction in the Nusselt number and in the convection and conduction value of the finned surface over an insulated cylinder. Yan and Sheen (2000) performed an experiment to determine the heat transfer and pressure drop characteristics of a plate (wavy and partitioned) fin heat exchanger. Yakut and Sahin (2004) experimentally investigated the flows that cause the vibration characteristics of the turbulators used to increase the heat transfer in heat exchangers. They found that the Nusselt number increases with the Reynolds number.

The goal of the present study is to reduce thick of the boundary layer that is formed on the circular fins placed on the heating tube, thus increasing heat transfer through convection in this area.

MATERIALS AND METHODS

Information on the experimental apparatus, devices used and procedures followed were given in detail by Yakar (2007). A summary of that information is presented below.

Experimental set - up

Heating systems with finned tubes used in practice are usually liquid/gas heat exchangers. Therefore, we manufactured an experimental set - up inside of which water is carried, and outside of which air passes. All control parameters of the experimental set - up were controlled entirely by a computer. Similarly, all of the measurements were also made by a computer. The experimental set - up is given in Figure 1.

As seen in Figure 1, because air was used as the heated fluid, to fin heating power for different mass flow rate of air, fan (no. 2) that has a variable round and enables the air to enter the air channel was installed on a table (no. 1) 1 m away from the floor. Considering the fully - developed hydraulic circumstances, the canal (no. 3) through which the air that was supplied by the fan and is to be heated in the heater is 3 m of long. The mass flow rate of the heated air at the entrance and at the exit of the heat exchanger was measured by a propeller - type flow meter (no. 4 and no. 14). The temperatures were measured by a T - type Copper - Constantan thermocouple. The same type of thermocouples were used to measure the temperatures on the tube's outer surface at locations between the fins and on the fin ends at 15, 45 and 75 cm distances from the 900 mm long finned heating tube. This case is shown in Figure 2.

Because measurements were made according to suction and compression in the study, a manometer was used that could be run in either case and that could measure the input and output pressure difference of the air (no. 6 and no. 11). The tube through which heating water passes was galvanized steel. The fin material was also galvanized steel. The external diameter of the tube was 29 mm and the external diameter of the fin was 87 mm. In addition, the pressures (no. 9 and no. 17), temperatures (no. 10 and no. 18) and volumetric flow rate (no. 11 and no. 19) of the heating water were determined while it entered and exited the heat. The heat exchanger, along with its equipment, was installed on the no. 15 experimental set table. The water used to heat the air was carried to the heat exchanger by a black tube (no. 16). The hot water used to heat the air was carried from the water tank (no. 21) to the heat exchanger by a pump (no. 20). The water used to heat the air was heated again while passing through the heat exchanger by heaters (no. 22) placed inside the 250 L water tank (no. 21). The capacity of the heaters was determined according to the heating capacity of the system. The volume expansion (no. 23) when the water in the water tank was heated was stored in the reservoir. Because the length of the heating tube was short and the c_p of water is high, the body into

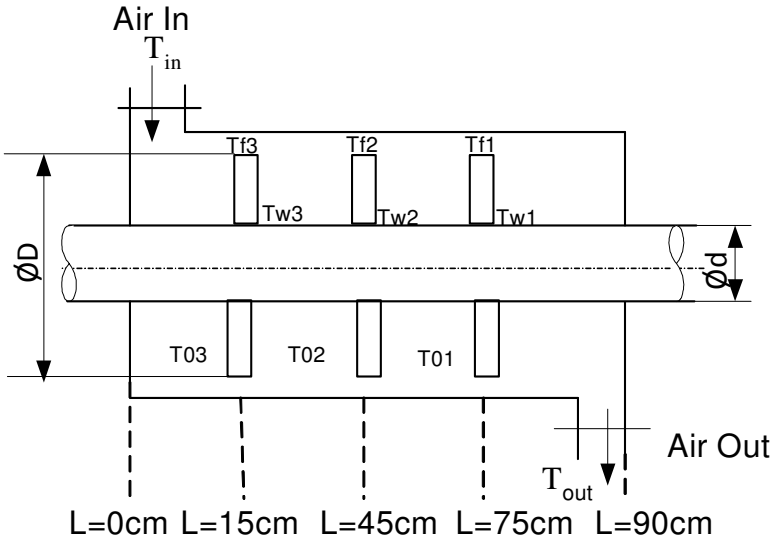


Figure 2. Temperature measurement points on the air side of heat exchanger.

Table 1. Uncertainties of the values measured in experiments.

Measured value	Uncertainty in measurement tool
Pressure difference on the air side	±0.16 mbar
Temperature on the air side	±0.5ºC
Velocity on the air side	±0.2 m/s
Tube diameter on the air inlet side	±2 mm
Distance between two fins	±1 mm
Fin diameter	±0.5 mm
Flow on the water side	±0.4 l/h
Pressure on the water side	±0.2 mbar
Temperature on the water side	±0.1ºC

Table 2. Uncertainty in the values calculated with the units measured in the experiment.

Calculated value	% error
$\dot{m}_h$ (kg/s)	0.34
u_m (m/s)	0.35
Q_T (W)	0.16
Q_h (W)	0.34
T_0 (ºC)	0.58
T_w (ºC)	0.58
T_f (ºC)	0.58

which the finned tube was placed was insulated with an insulating material (no. 24) of 10 mm thickness so as to obtain more sensitive results by preventing heat loss. Moreover, the route of the air in the heating tube was lengthened by means of deflectors (no. 25) placed in the inner surface of the external body. The heat transfer was thus increased by increasing the period of contact with hot surfaces.

Error analysis of the experiment

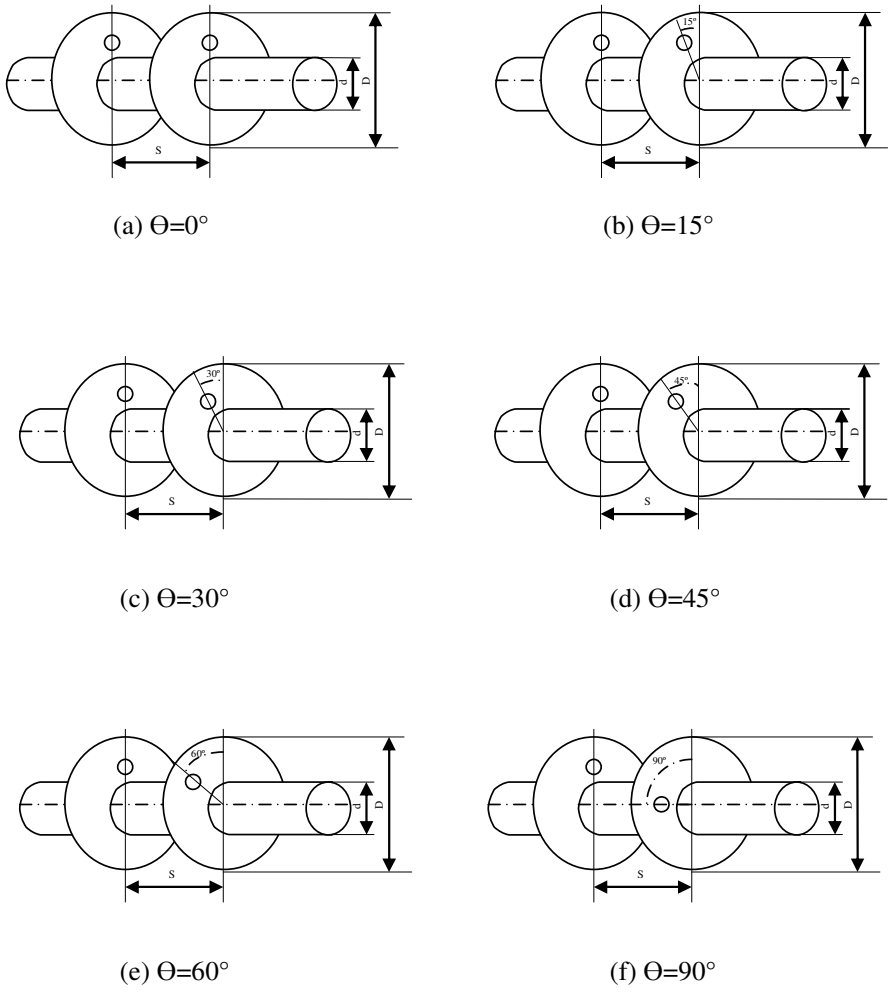
Each of the measurement tools used in experiments has a measurement uncertainty. While measuring, the errors (uncertainty) that arose from the measuring sensitivities of the tools must be determined. For example, if R depends on "n" independent variables (x_1, x_2,….x_n), then the error in R, w_R, depends on these variables according to

$$w_R = \left[\left(\frac{\partial R}{\partial x_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} w_2 \right)^2 + \left(\frac{\partial R}{\partial x_3} w_3 \right)^2 + ... + \left(\frac{\partial R}{\partial x_n} w_n \right) \right]^{1/2} \quad (1)$$

The uncertainty values of the measurement tools used in the experiments are given in Table 1, and the error analysis of the values measured and calculated in experiments is given in Table 2.

Processing the experimental data

In this study, the external diameter of the body into which the finned tube was placed was 154 mm, the heating tube external diameter was kept fixed at 29 mm (d), the fin diameter was 87 mm (D) and the fin thickness was 0.5 mm (t). However, the distance between the fins was varied to be one of five different values (s); these were 4, 8, 10, 12 and 15 mm. The heating tube length was kept fixed at 900 mm. In accordance with the goal of the experimental study, convective heat transfer was intended to be increased by reducing thick of the boundary layer on the fins by using 6 mm diameter holes that were opened at different angles on the circular fin that was placed on the heating tube. These same-diameter holes on each fin could follow each other at the same selected angle.



(a) Θ=0° (b) Θ=15°

(c) Θ=30° (d) Θ=45°

(e) Θ=60° (f) Θ=90°

Figure 3. Finned heating tubes where it was opened 6 mm-diameter holes at the angular locations Θ=0°, Θ=15°, Θ=30°, Θ=45°, Θ=60° and Θ=90°.

Experimental studies were carried out at six different angular positions. These angles were 0, 15, 30, 45, 60 and 90°. Figure 3 shows the circular finned heating tubes that had 6 mm-diameter holes at above angles.

Flow of the heating fluid (water) was kept fixed in the experiments while water temperatures were varied, thus allowing the experiment to be conducted at seven different temperatures. These temperatures were 40, 45, 50, 55, 60, 65 and 70℃. Air flow was varied to be one of three different values: 0.04, 0.06 and 0.08 kg/s.

MATHEMATICAL FORMULATION

Heating power (Q_t), which is created by the hot fluid (water), is transferred to the air that passes over the external part of the tube through radiation (Q_R) and convection (Q_c).

When the temperature distribution along the fin is known, it is theoretically possible to calculate the amounts of both convection and radiative heat transfer from the cell made by the two neighbouring fins. If the energy balance is written for the elementary area dA, as shown in Figure 4, then the following equation is obtained:

$$\frac{d^2T}{dr^2}+\frac{1}{r}\frac{dT}{dr}=\frac{2h_f}{\lambda_f t}(T-T_o)+\frac{2\sigma\varepsilon}{\lambda_f t}\left[F_{dr-w}\left(T^4-T_w{}^4\right)+F_{dr-o}\left(T^4-T_o{}^4\right)+\int_{r_i}^{r_o} dF_{dr-dr'}\left(T^4-T'^4\right)\right] \quad (2)$$

The expression given in equation (2) is usually a nonlinear integro-differential equation. In addition, shape factors contain complex integral expression depending on fin geometry. Therefore, the equation is hard to solve. In this study, fin temperature was taken to be the arithmetical average of the experimentally determined fin-end temperature and fin-bottom temperature.

Heat transferred by radiation

When calculating the amount of radiative heat transfer from the circular fin surfaces placed on the heating tube, it is necessary to first determine the shape factors of the surfaces. The shape factors depend on the fin parameters of the surface in the closed cell made by two neighbouring fins, as shown in Figure 5, and given in reference (Yakar, 2007).

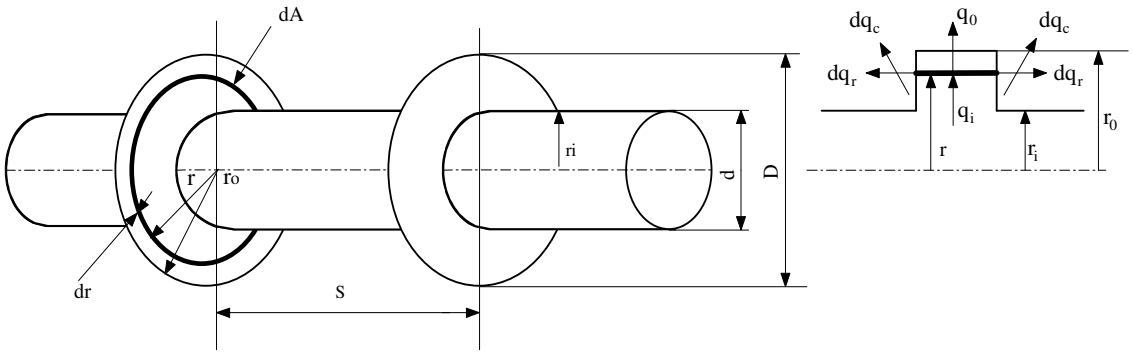


Figure 4. Circular finned and horizontal tube heater.

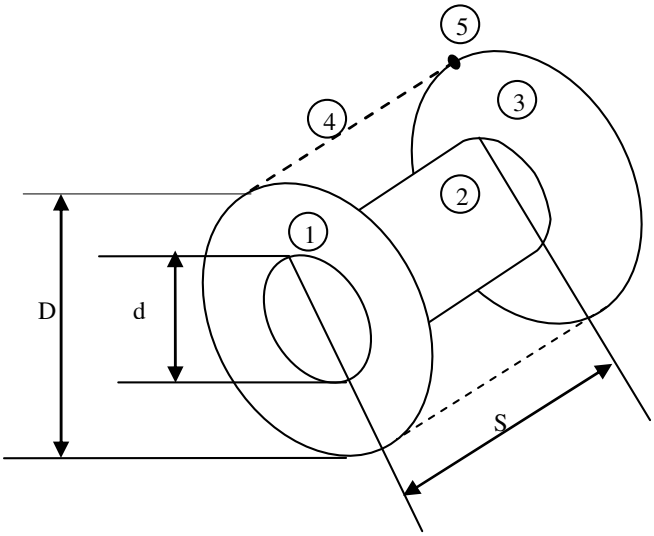


Figure 5. Closed cell component formed between two fins.

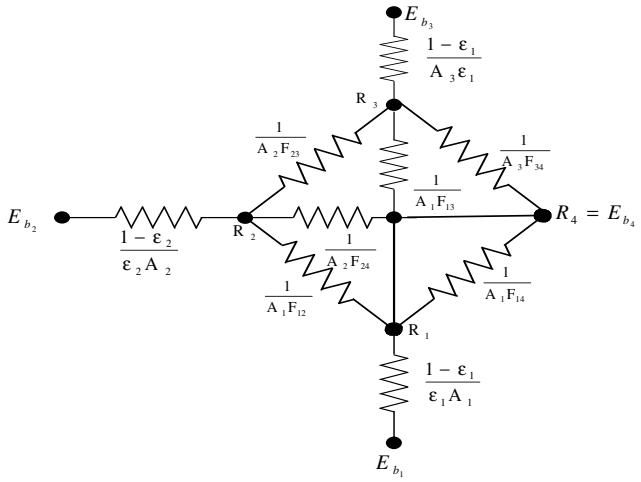


Figure 6. Electrical analogy circuit pertaining to the closed cell formed between the two fins.

To calculate the radiative heat transfer from the fin surfaces to the surrounding air, an analogous electrical circuit, as shown in Figure 6, was used for the thermal interaction between the surfaces.

Because the sum of currents in each node of the electrical circuitis “0”, one can write for any point R_i, i = 1, 2 and j = 1, 2, 3, 4

$$\frac{\frac{E_{bi} - R_i}{\frac{1-\varepsilon_i}{\varepsilon_i}}}{} + \sum_j \frac{R_j - R_i}{\frac{1}{F_{ij}}} = 0 \quad (3)$$

Where,
$E_{bi} = \sigma T_i^4$ – Black body radiation energy (W/m^2)
R_i – Surface radiation (W/m^2)
ε_i – Surface emissivity coefficient

According to Figure 6, radiation from (1) and (2) surfaces are the same, ($R_1 = R_3$) and ($R_4 = E_{b4}$). In addition to the condition $q_{14} = q_{34}$, radiative heat transfer from the cell formed by two fins to outer air medium can be stated as;

$$q_4 = 2q_{14} + q_{24} \quad (4)$$

Where, q – Surface heat transfer per unit

$$q_{14} = F_{14}(R_1 - R_4) \quad (5)$$

$$q_{24} = F_{24}(R_2 - R_4) \quad (6)$$

In the horizontal heater formed by n circular fins, radiative heat transfer from all of the heater cells is:

$$Q_{R1} = (n - 1)A_4 q_4 \quad (7)$$

As a result,

$$Q_{R1} = K(T_w/100)^4 + M(T_f/100)^4 - N(T_o/100)^4 \quad (8)$$

Here, T_f is the fin temperature, T_w is the heating tube surface temperature and T_0 is the temperature of the heated air from the upper part of the fin. The values of the K, M and N coefficients given in equation (8) are given in reference (Yakar, 2007). For the galvanized steel fin and the heating tube material, emissivity coefficients $\varepsilon_{gal} = 0.06$ and $\varepsilon_d = 0.15$ were used for the solution of equation (8), respectively.

Radiative heat transfer from fin ends is as follows:

$$Q_{R_2} = \sum (A_F)_{end} \sigma \varepsilon \left[\left(\frac{T_{f_{end}}}{100} \right)^4 - \left(\frac{To}{100} \right)^4 \right] \quad (9)$$

The total radiative heat transfer from all of the heating surfaces is as follows:

$$\Sigma Q_R = Q_{R1} + Q_{R2} \quad (10)$$

Heat transferred by convection

The convective heat transfer is given by,

$$Q_c = h_o(\eta_o A_o)(T_w - T_o) \quad (11)$$

Here, $A_0 = A_w + \Sigma A_f$: total heat transfer surface and η_o: represents the efficiency of the finned surface. It can be calculated with the equation (Yakar, 2007):

$$\eta_o = 1 - \frac{\sum A_f}{A_o}(1 - \eta_f) \quad (12)$$

Fin efficiency η_f was given in reference (Yakar, 2007) according to the fin parameters. Moreover, A_w is the area of the outer surface of heating tube and ΣA_f is the total surface area of the fin. The convective heat transfer coefficient h_o is determined by the following equation when total energy input Q_t is known:

$$h_0 = \frac{Q_t - \Sigma Q_R}{\eta_0 A_0 (T_w - T_0)} \quad (13)$$

η_o in equation (12) contains a heat convection coefficient that must be solved for by iteration. Iteration is continued until the η_o, η_f and h_o values converge to within 1%. Depending on the h_o value obtained by iteration, the Nusselt number is determined as follows (depending on the outer diameter of the heating tube):

$$Nu_d = \frac{h_0 d}{\lambda} \quad (14)$$

RESULTS AND DISCUSSION

The heated air flows through the spaces between the fins and the body. As the section area changes in the direction of the flow, the air decelerates and accelerates. The air also flows to the heating tube in the direction of the flow. First it flows parallel, then vertical and then parallel again because of the deflectors used in the system. Therefore, the movement of the air on the heating tube changes during the flow in such a way that both the flow section and the flow direction are continually repeated. However, mass flow remains the same in all the flow and section changes. Considering this, the Reynolds number that characterizes the air should be defined according to the mass velocity $\dot{G} = \frac{\dot{m}_{air}}{A_s}$. Here $\dot{m}_{air}$ is the mass flow of the air passing through the heater at a unit time and A_s is the area of reference section perpendicular to the flow direction between two fins (A_s = D.s). However, under experimental conditions, not all of the air mass passes through the section between two fins. Some of it passes through the intersection between the fin end and the outer body, and some passes through the holes that are at certain angular positions. Therefore, the Reynolds number cannot be defined exactly if one considers the flow perpendicular to the tube. In addition, the heater should contain the effects of the other flows. This case is rather complicated. Now that heat production of the finned heating tube, the fin, the heating surface and the air temperatures are known for a certain geometric position and study condition, the values of radiative heat transfer and convective heat transfer can be determined. As convective heat transfer is dependent on movement of the air, the Re_d number (which characterizes the flow) should take into account the convective heat transfer calculated for unit surface ($q_c = Q_c/A_o$). As a result, the Reynolds number should be described according to ($q_c = Q_c/A_o$). After the required adjustments are made, the modified Reynolds number (which is obtained in such a way as to take this into account), can be stated as follows, depending on the measured values:

$$\overline{Re}_d = Re_d \, Nu_d = \frac{q_c \dot{G} d^2}{\mu \lambda (T_w - T_0)} \quad (15)$$

Here, physical properties of the air, $T_{film} = \frac{T_w + T_0}{2}$, are obtained with the use of the film temperature.

$\overline{Re}_d$ values were determined for the cross flows and for air flows at 0.04, 0.06 and 0.08 kg/s according to the imperforate state and according to the 0, 15, 30, 45, 60 and 90° angular positions of the holes opened on the fins of finned heating tubes. The heating tube's diameter was 29 mm, the fin diameter was 87 mm, and the distance between the fins was 10 mm. The tube length was 900 mm. According to different water input temperatures, the change of the modified Reynolds number $\left(\overline{Re}_d\right)$ is shown in the diagrams between Figures 7 and 9.

According to the experimental results for each of the three different air flows (0.04, 0.06 and 0.08 kg/s), the $\overline{Re}_d$ values for the imperforate position at water input temperatures from 40 - 60°C in the cross flow are higher than the $\overline{Re}_d$ values at the other positions. Above 60°C, however, the $\overline{Re}_d$ values at Θ = 90° are higher than the $\overline{Re}_d$ values of the other positions. This case is shown in Figures 7, 8 and 9.

These results show that the flow characteristics of the air in the heater change. This change is also influenced by Θ, the angular positioning. The temperature gradient that causes the heat transfer between the fluids in cross flow is different from each other. It can be considered that

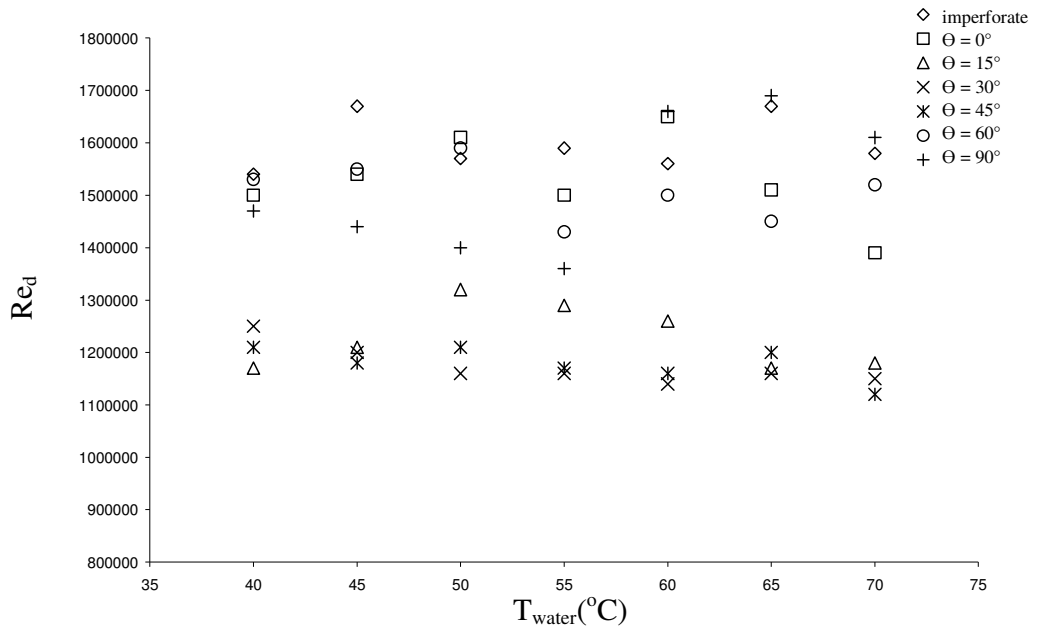


Figure 7. Change of $\overline{\mathbf{Re_d}}$ value according to water input temperature $\mathbf{T_{water_i}}$ in cross flow at 0°, 15°, 30°, 45°, 60°, 90° angular positions without holes for s/d=0.345, $\dot{\mathbf{m}}_{\mathbf{air}} = \mathbf{0.04}$ kg/s and $\dot{\mathbf{m}}_{\mathbf{water}} = \mathbf{1.1}$ kg/s.

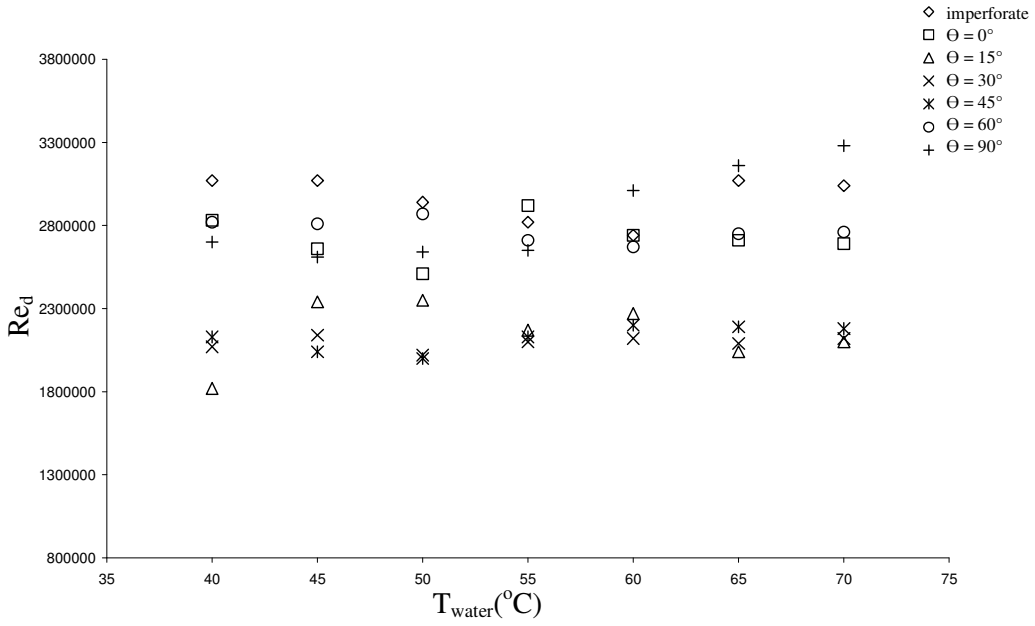


Figure 8. Change of $\overline{\mathbf{Re_d}}$ value according to water input temperature $\mathbf{T_{water_i}}$ in cross flow at 0°, 15°, 30°, 45°, 60°, 90° angular positions without holes for s/d=0.345, $\dot{\mathbf{m}}_{\mathbf{air}} = \mathbf{0.06}$ kg/s and $\dot{\mathbf{m}}_{\mathbf{water}} = \mathbf{1.1}$ kg/s.

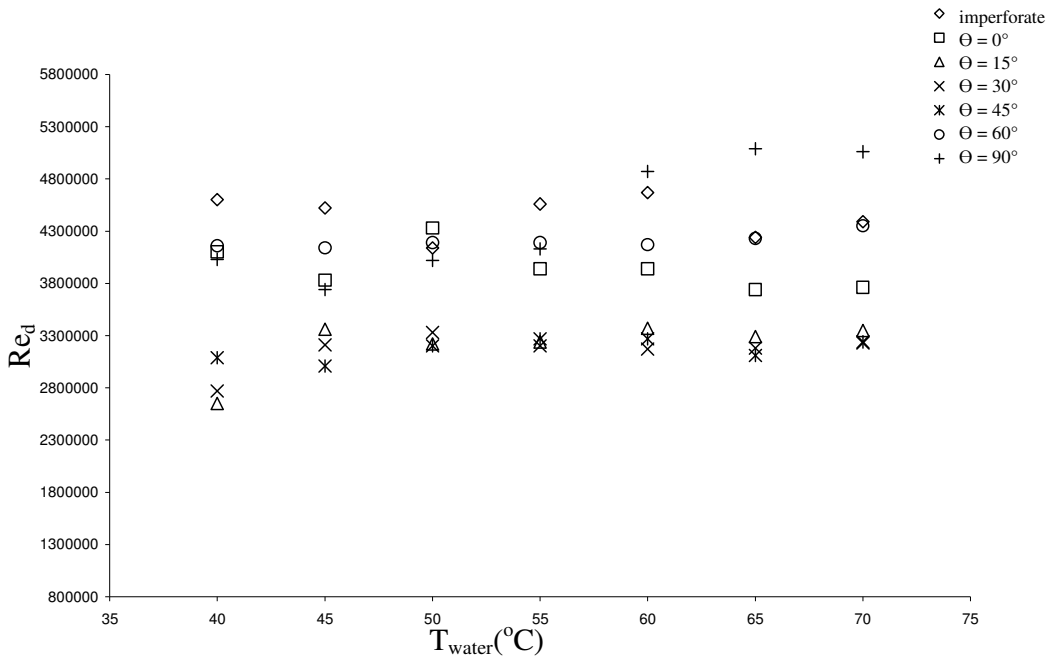


Figure 9. Change of $\overline{\mathbf{Re_d}}$ value according to water input temperature $\mathbf{T_{water_i}}$ in cross flow at 0°, 15°, 30°, 45°, 60°, 90° angular positions without holes for s/d=0.345, $\dot{\mathbf{m}}_{\mathbf{air}} = \mathbf{0.08}$ kg/s and $\dot{\mathbf{m}}_{\mathbf{water}} = \mathbf{1.1}$ kg/s.

this case will change the $\overline{Re_d}$ number. Because the physical properties of the fluid (the air) changes.

Because boundary layer thickness is related to flow speed and flow speed is related to the Reynolds number, the heat convection coefficient (h) in the convective heat transfer is dependent on the distribution of temperature within the thermal boundary layer. Because Pr ≈ 1 in gases, the Nusselt number that expresses the convective heat transfer must be a function of the Reynolds number. The change of Nu_d values according to the $\overline{Re_d}$ for the imperforate and perforated position in the interval $0° \le \theta \le 90°$ of the finned heating tube is shown in the diagrams between Figure 10 and 12.

At air flow values of 0.04, 0.06 and 0.08 kg/s, the $\overline{Re_d}$ and Nu_d values at the Θ = 90° imperforate angular position in cross flow appear to be greater than the $\overline{Re_d}$ and Nu_d values at the other positions. This can be seen in Figures 10, 11 and 12. Because the environmental temperature of the angular position at which Nu_d values are greater than the other angular positions is observed to be greater than the environmental temperatures of the other positions, it is thought that this case is influenced by the temperature of the environment where the experimental set – up is located. If the surrounding air of a certain mass flow enters the heater at a high temperature, the mean temperature value appears to be greater than the mean temperature value at the lower surrounding air temperature, whereas the volume of the same mass flow increases because density decreases as temperature increases. As the increasing volume passes through the same flow section, its average speed increases. The currents created by the increase in average speed and acceleration increase the Reynolds number. As a result, both the Reynolds number and the Nusselt number increase.

Conclusion

The heater whose fins did not have any holes and the heater whose fins had holes with an angle Θ = 0° and an s/d = 0.345 were studied. When Nu_d values were considered for the same air flow rate and the same $\overline{Re_d}$, the Nu_d value of the heater with holes was found to be 2.43% higher than the Nu_d value of the heater without holes. However, there was a 3.84% decrease in pressure.

For the imperforate state and with Θ = 90° in the s/d = 0.345 state, when the Nu_d numbers were examined for the cross flow with the same $\overline{Re_d}$ numbers, it was found that the Nu_d number in the perforated state was 2.94%

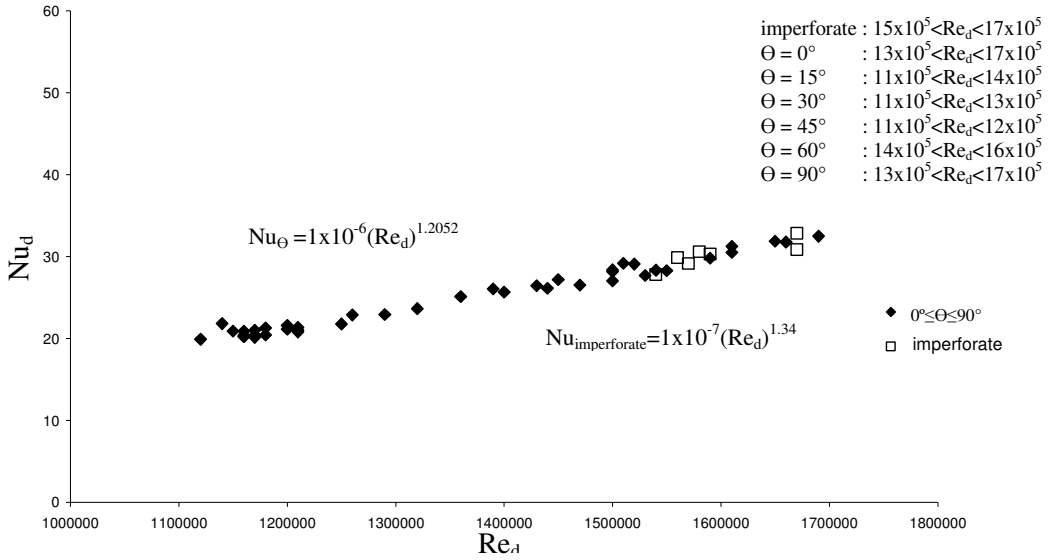


Figure 10. Change of Nu_d number according to $\overline{Re_d}$ for s/d=0.345, $\dot{m}_{air} = 0.04$ kg/s and $\dot{m}_{water} = 1.1$ kg/s in the interval $0^0 \le \theta \le 90^0$ and in imperforate position for cross flow.

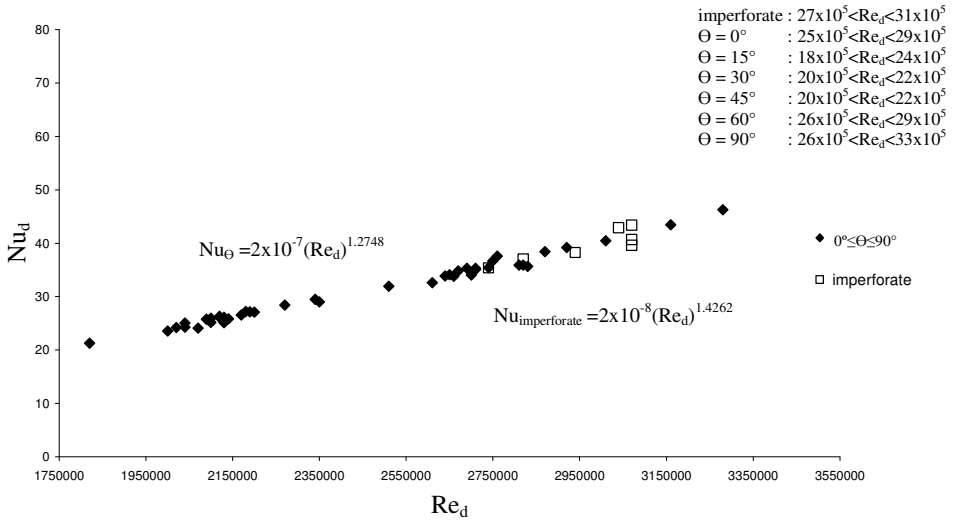


Figure 11. Change of Nu_d number according to $\overline{Re_d}$ for s/d=0.345, $\dot{m}_{air} = 0.06$ kg/s and $\dot{m}_{water} = 1.1$ kg/s in the interval $0^0 \le \theta \le 90^0$ and in imperforate position for cross flow.

higher than the number in the imperforate state, and that there was no change in the pressure drop.

ACKNOWLEDGEMENT

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Nomenclature: s, Distance between fins (m); **L,** Length of heating tube (m); **D,** Fin diameter (m); **d,** Outer diameter of heating tube (m); **t,** Fin thickness (m); w_R,

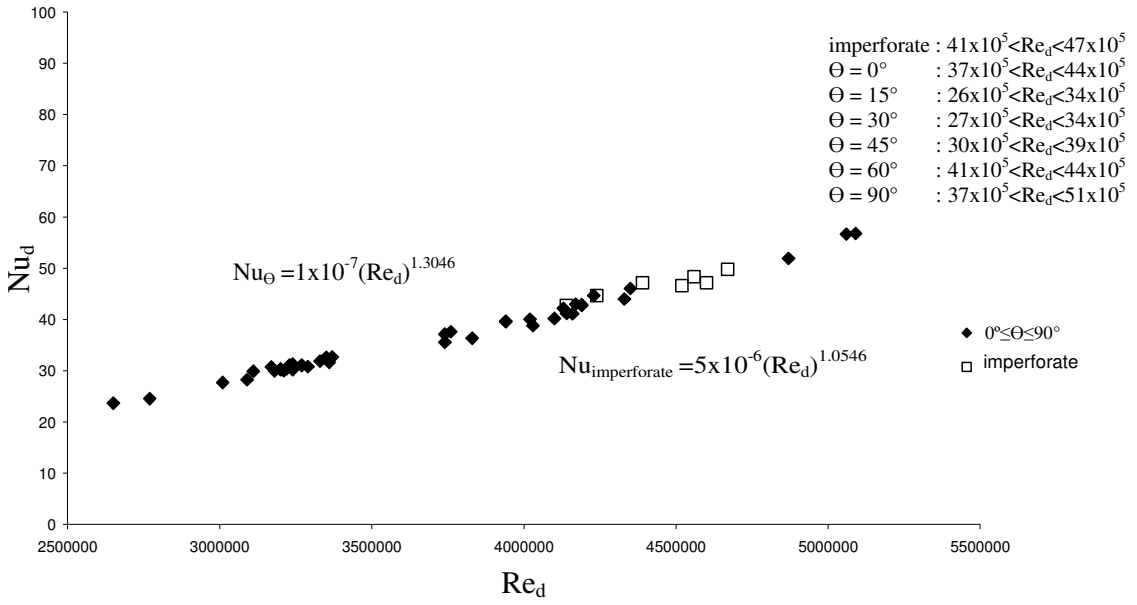

Figure 12. Change of Nu_d number according to $\overline{Re_d}$ for s/d=0.345, $\dot{m}_{air} = 0.08$ kg/s and $\dot{m}_{water} = 1.1$ kg/s in the interval $0^0 \le \theta \le 90^0$ and in imperforate position for cross flow.

Uncertainty; **e_{bi}**, Black body radiation energy (W/m^2); **q,** Surface heat transfer per unit (W/m^2); **Q_t**, Heat energy supplied from hot fluid (W); **Q_c**, Convective heat transfer (W); **ΣQ_R**, Total radiative heat transfer (W); **Q_w**, Amount of the heat transferred by air over heating tube (W); **Q_f**, Amount of the heat transferred by air over fins (W); **F_{ij}**, Shape factor between i surface and j surface; **h_f**, Heat convection coefficient around fin (W/m^2℃); **h_w**, Heat convection coefficient around body (W/m^2℃); **h_o**, Visible heat convection coefficient (W/m^2℃); **A_o**, Total heat transfer area (m^2); **c_{water}**, Specific heat capacity of water (kJ/kg℃); **c_{air}**, Specific heat capacity of air (kJ/kg℃); **A_w**, Heating tube surface area except total fin surface area (m^2); **ΣA_f**, Total area of the fins on the tube (m^2); **u_m**, Average speed of air (m/s); **T_w**, Temperature of heating tube surface (℃); **T_o**, Temperature of the heated air (℃); **T_f**, Fin temperature (℃); $T_{f_{end}}$ **,** Fin end temperature (℃); T_{water_i} **,** Input temperature of water to tube (℃); T_{water_o} **,** Output temperature of water from tube (℃); **ΔT_{water}**, Difference between the input and output temperature of water in heating tube (℃); ΔT_{air} **,** Difference between the input and output temperature of air from heat exchanger (℃); T_{air_i} **,** Input temperature of air into heat exchanger (℃); T_{air_o} **,** Output temperature of air from heat exchanger (℃); $\dot{m}_{water}$ **,** Mass flow rate of water (kg/s); $\dot{m}_{air}$ **,** Mass flow rate of air (kg/s); $\dot{V}$ **,** Volumetric flow rate of air (m^3/h); $\dot{G}$ **,** Mass flow speed (kg/m^2s); **Nu_d**, Nusselt number; **Re_d**, Reynolds number; $\overline{Re}_d$ **,** Modified Reynolds number.

Greek symbols: **ε_i**, Emissivity coefficient for surface; **λ,** Heat conduction coefficient (W/m℃); **λ_f**, Fin material heat conduction coefficient (W/m℃); **σ ,** Stefan-Boltzmann constant (W/m^2K^4); **η_o**, Finned-surface efficiency; **η_f**, Fin efficiency; **v,** Kinematic viscosity (m^2/s); **μ,** Dynamic viscosity (kg/ms).

Subscripts: **t,** Total; **c,** Convection; **R,** Radiation; **w,** Wall; **f,** Fin.

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