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Heat Transfer Augmentation in Fin and Tube Heat Exchanger Embedded with Vortex Generators

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ABSTRACT

The aim of this study is to investigate experimentally turbulent flow around tube by using multi–shapes of winglet vortex generators mounted behind the tube at different angles of attack, in order to investigate the effect of winglet on heat transfer and pressure drop of fin and tube heat exchanger placed inside a rectangular duct with Reynolds numbers (Re) ranging from (19000) to (49000). Fin-and-tube heat exchangers are frequently employed in a variety of industries, including power systems and air conditioning. The heating element inside the tube receives a power source ranging from (10W) to (40W). In these studies, vortex generators of the triangular and rectangular types are installed at various angles of attack (45° and 60°) on the fin surface behind the tube. The findings demonstrate that depending on the form and angle of attack of the winglets, utilizing winglet pairs has an impact on the heat transfer coefficient and friction factor relative to other designs. With increasing angle of attack, the friction factor and heat transfer coefficient rise. The optimal design for improving heat transfer is the rectangular type vortex generator, whereas the triangular type produces the least amount of heat transfer. The rectangular style of duct has the greatest pressure drop. For rectangular and triangular heat exchangers, respectively, the average Nusselt number is increased by 19% and 12%. The average Nusselt number of a fin and tube heat exchanger is increased by (16.7-17.8%) % for the rectangular type and by (9.3-10.6%) % for the triangular type at an angle of attack (β=60°) compared to (without winglet) case for Reynolds number ranging between (19000-49000), where the triangle type gives the least amount of heat transfer compared to other shapes.

1. Introduction

Fin-and-tube heat exchangers are frequently utilised in a variety of industries, including aerospace, automotive, HVAC, refrigeration, power systems, chemical engineering, and electronic cooling. Typically, the liquid flows inside the tubes while the gas flows across them. Since the transfer coefficient on the gas side is typically significantly smaller than that on the liquid side, the performance of the heat exchanger is frequently limited by the gas side. The primary concern while designing a heat exchanger is how to optimise heat transfer in order to raise overall performance to satisfy the demands of high efficiency (energy savings) and low cost with the smallest possible volume and lightest possible weight [1].

In order to create mixed flow, disturb flow, and control the development of boundary layers, vortex generators are specific surfaces that are placed in the stream flow. The flow along the side edges splits and produces longitudinal vortices as a result of the pressure difference between the vortex generators' forward and backsides. Transverse, longitudinal, and horseshoe vortices can all be produced by the vortex generators. These vortices substantially alter the flow structure and have an impact on the temperature and velocity fields. These elements explain why vortex generators improve heat transfer [2]. A wealth of literature exists on heat transfer enhancement studies where these techniques applied separately or in combination. In this section a brief review of previous work related to some techniques of heat transfer enhancement.

2. Previous studies

In an experimental investigation of heat transfer in turbulent flow, **Khan, Hinton, and Baxter** [3] used inclined solid and perforated baffles along with rib turbulators to create a rectangular channel. All experiments were conducted at a constant Reynolds number of 37300 while adjusting the angle of the baffle with the bottom surface of the duct from 3° to 7°. The initial experiment used a smooth channel; the second experiment used rib put intermittently in the channel but without baffles; and the third experiment used solid and perforated baffles in conjunction with rib turbulators. Due to the flow obstruction generated by the baffle, which increases the velocity of the air flow over and through the baffle, they discovered that the local Nusselt number increases with the angle of inclination of the baffle.

A numerical analysis of the heat and fluid flow in a square duct with V-shaped ribs was carried out by **Jia, Saidi, and Sunden** [4]. With Reynolds numbers ranging from (15000) to (32000) and uniform heating at the ribbed surface, calculations were done for the (45°) V-shaped ribs in this study that were positioned both in line and staggered. This study showed that transverse straight ribs of the same geometry perform worse in terms of improving heat transfer enhancement than downstream and upstream pointed V-shaped ribs. The angled ribs' secondary flow, which improves fluid exchange between the core and the wall areas and thins the boundary layer close to the walls, was primarily responsible for this improved performance.

An experimental examination of heat transfer and friction factor was conducted by **Karwa, Maheshwari, and Karwa** [5] in an asymmetrically heated rectangular duct with solid or perforated baffles attached to one of the board walls. The study's Reynolds number ranges from 2850 to 11500. The duct's baffled wall is evenly heated, and the other three walls are insulated. The Nusselt number for perforated baffles was about (60 - 62.9) %, decreasing with an increase in the open area ratio (ratio of the perforations to the baffle frontal area) of these baffles, while the Nusselt number for solid baffles was about (73.7 - 82.7) % higher than that for the smooth duct over the study's range. The solid baffles' frictional coefficient was found to be (9.6 – 11.1) times of the smooth duct, which decreased significantly for the perforated baffles with the increase in the open area ratio to about (2.3 – 3).

Lin and Jang [6] conducted an experimental investigation on how wave-type vortex generators embedded within fin-tube heat exchangers could improve heat transfer efficiency. Additionally, 3-D turbulence study of the heat transfer and fluid flow related with wave-type vortex generators embedded fins and conjugate heat transfer investigations were conducted using numerical methods. In this investigation, Reynolds numbers between (680) and (4050) were used. The outcomes showed that the vortex generators' geometric characteristics had a significant impact on heat transmission and friction loss.
The average Nusselt number was found to improve by a maximum of 18.5% in this investigation, while the local Nu value increased by a maximum of (120%). However, the improvement in heat transmission was accompanied by a rise in the friction factor of (48%). They discovered that strengthening the longitudinal vortices with a higher vortex generator's height and a larger vortex generator's diameter significantly increased the heat transfer coefficient.

In a rectangular channel with common flow down and common flow up scenarios, **Kim and Yang** [7] conducted an experimental research to determine the flow and heat transfer properties of embedded counter-rotating vortices in turbulent boundary layers. The distance between the vortex generators is (40mm) apart, and the angle of attack of the vortex generators is adjusted to control the strength of longitudinal vortices at (20°, 30°, and 45°).In this experiment, the Reynolds number is around (31400). They discovered that there are two maximum values in the local heat transfer distributions that exit for the three angles of attack in common flow down scenarios. With the common flow up, only one maximum value exists, therefore the common flow down cases show better heat transfer characteristics than common flow up cases.

The performance of a plate heat exchanger with a rectangular winglet pair was numerically analysed by **Chung, Park, and Lee** [8] to look at the combined effects of vortex generators and louvred fins. Reynolds number (300-1100), angles of attack (15°, 30°, 45°, and 90°), and louvre angles (0°, 15°, 30°, and 45°) are the variables taken into account in this study. Figure 2-7 shows how louvred fins cause transverse vortices whose rotation axes are parallel to the main flow direction. When the angle of attack is set to (90°), the winglet in figure (2-7) can be viewed as a standard louvre fin, and when it is set to (0°), it can be viewed as a vortex generator. The heat transfer rate is increased at the expense of pressure loss as the angle of attack or the louvre angle lowers, according to the results. It has been discovered that an attack angle of (30°) and a louvre angle of (15°) yield the optimal performance. Louvre angle alone is found to have very little of an impact on heat exchanger performance, while louvre angle and angle of attack combined have a significant impact.

Pesteei, Subbarao, and Agarwal [9] conducted a thorough experimental study to identify the optimal location for the delta winglet pair by measuring local heat transfer coefficients on fin-tube heat exchangers with winglets using a single heater and five different positions of winglet type vortex generators, as well as flow loss in a charnel. At a Reynolds number of around 2250 and a 45° angle of attack, measurements were taken. The use of winglet-type vortex generators increased heat transfer significantly, according to the results. In comparison to a plain fin-tube heat exchanger, they found that the average Nusselt number increased by roughly (46%) and the local heat transfer coefficient increased by several times. The recirculation zone showed the greatest improvement, and the ideal position for the winglets was at X=0.5D and Y=0.5D.They discovered that placing the winglet pair on the downstream side improves the heat transfer coefficients most effectively.

Tiggelbeck et al. [10] investigated experimentally the effect of four basic forms of vortex generators (delta wing, rectangular wing, pair of delta winglets, pair of rectangular winglets) on the heat transfer enhancement and flow losses and compared the results between these forms in the Reynolds Number range of 2000 to 9000 and for angles of attack between (30°) and (90°). They found that the vortex generators increase heat transfer and the flow losses in channel, and for all vortex generators geometries there exists an optimum angle of attack between (50°) and (70°) for maximum heat transfer. However, the flow losses increase monotonically with the angle of attack. Results show that the winglet gives better performance than wings and a pair of delta winglets perform slightly better than the pair of rectangular winglets.

Wu and Tao [11] performed a three-dimensional numerical simulation for laminar flow heat transfer of the fin and tube heat transfer surface of a compact heat exchanger in an aligned arrangement with delta - winglet vortex generators, under the conditions of accounting for all physical factors such as the heat conduction in the fin and winglet, the thickness of the winglet and punched hole, and including all rows of tubes in the computational element. The effects of the attack angle (30° and 45°) and Reynolds number (800–2000) of the delta winglet vortex generator are investigated. They found that the delta winglet pair punched out from the fins can increase the average Nusselt number by (16 – 20) % for angle of attack (30°) and by (20 – 25) % for angle of attack (45°), whereas it increases the pressure drop by (10 – 12) % for (β = 45°) and decreases it by (18 – 10) % for (β = 30°) compared with plain – plate fin and tube heat exchanger.

In order to examine the impact of winglets on heat transfer and pressure drop of a fin and tube heat exchanger installed inside a rectangular duct, this study will experimentally investigate turbulent flow around tubes using winglet vortex generators mounted behind the tubes at various angles of attack. The temperature of the area around the tube and the surface of the fin was measured using a rig. Additionally, measurements of flow characteristics such air temperature, pressure differential, and velocity were made. The rig is constructed such that the winglet pairs can be utilized with the tube at various angles of attack (β), as well as to change the flow velocity and heat flux.

3. Experimental Investigation

3.1 Test section

The upstream section is connected to a rectangular duct with a cross section of (300 mm × 60 mm) and a length of 1500 mm using screws, and the remaining area of the upstream section outlet is blocked by a plate that has been bent from the test section's bottom surface. To stop air leakage, silicon sealant is applied to the outside of every contact point between two adjacent parts. The fin and tube heat exchanger is positioned in the middle of the test section width, 1000 mm from the test section intake. A (130mm×130mm) square window at the test section top surface is formed above the tube which is covered by (140mm×140mm) movable plate in order to assemble and disassemble the fin-tube and winglets easily and in a relatively short period of time. As indicated in figure (1-1), cork is used to cover the whole external surface area of the test section. Cork is 50 mm thick, has a thermal conductivity of (k=0.047 W/m.C), and functions as an insulator by reducing heat losses (i.e., by maintaining the balance of heat transmission).

A hole is made in the centre of a square copper plate (70mm70mm and 1mm thick) by fitting a copper tube with dimensions of (35mm diameter), (40mm height), and (1mm thickness). The copper plate is modelled after a fin and is fixed at half tube height before being attached with an adhesive to maintain the tightness. As a heating element, a heating wire with a (0.5mm) diameter, (500W), and total electrical resistance of (200) is employed.

Six (5mm) diameter holes are produced along a circular (15mm) porcelain bar that is made of a dielectric substance and has excellent conductivity, and the heating wire is wound spirally through these holes. In order to fix the porcelain round bar and heating wire in place and achieve good thermal conductivity, soft magnesium oxide (MGO) powder is used. Two round discs of plaster (gypsum) material, each measuring 25 mm in diameter and 15 mm in length, are placed at each end of the tube to act as a heat insulator. These discs are fastened to the ends of the tubes using nuts and washers and an adhesive, with the screw going through the bar and the discs. This screw is used to connect the fin–tube heat exchanger to the top and bottom plates of the duct. The two terminal of the heater is insulated by sleeve and connected to the power supply through a transformer.

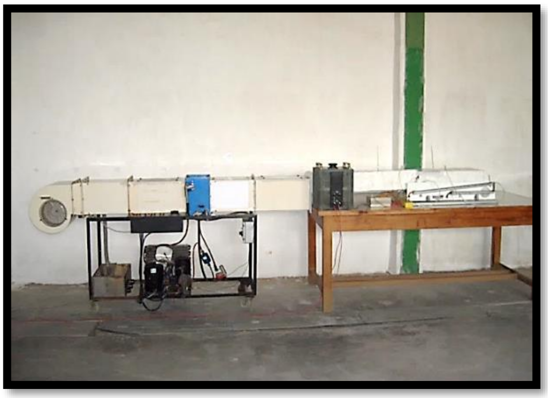


Figure (3-1) Photograph of Experimental Apparatus.

3.2 Experimental Procedure

This examination entails experimental analysis of the improvement of heat transfer from fin and tube heat exchanger placed in rectangular duct by using vortex generators. The purpose of this experiment is to investigate the impact of the shape and angle of attack of the vortex generator on the convection heat transfer coefficient and friction factor with variable Reynolds number and power supplied to the heater (Q).In this experiment, four different power levels are used for the heater within the tube as well as the centrifugal fan, which produces four different air flow velocities. The experimental procedure can be explained in the following steps:

1-Fixing the winglet on the fin surface behind the tube where the tips of the winglet pair is coincident at points ($X = R, Y = \pm R$) and at any angle of attack required.

2-Providing the apparatus with electrical power.

3-Operating the centrifugal fan and regulating the power supply to obtain the desired air flow velocity.

4-Operating the refrigeration system with or without (one / two) air flow heaters according to the environment condition to adjust the inlet air temperature range (20-22ºC).

5-Supplying the electrical power to the heater in the tube by voltage transformer to obtain the required power supply for the test.

6-The system is remained at this state until it reaches to the steady state condition, which takes about one hour or more from the start of operation, where the temperature distribution around the tube and on the fin surface is remained unchanged and the final data of temperature is confined in account.

7-Reading and recording voltage difference, electric current of the power supply to the heater element, temperatures on the tube and fin surfaces, inlet and outlet temperatures of the test section and static pressure drop across the test section.

8-The air flow velocity through the test section is varied to four values, and then step (6) is repeated.

9-Changing the power supply for the heater element in the tube to four values, and then steps (6), (7) and (8) are repeated.

10-Changing the angle of attack (45º and 60º) and winglet type (triangular, and rectangular type vortex generator) and the above steps are repeated.

3.3 Data Reduction

The following steps are followed in order to perform the calculations operations:

Computation of the air properties such as density, kinematics viscosity, thermal conductivity and Prandtl number. These correlations of air are [12]:

$$\rho = 4.9303 - 0.0258T_f + 6.397\times10^{-5}T_f^2 - 7.5\times10^{-8}T_f^3 + 3.36\times10^{-11}T_f^4 \quad \text{......... (1)}$$

$$\nu = -1.9937838\times10^{-6} + 2.2549115\times10^{-8}T_f + 1.2926274\times10^{-10}T_f^2 - 3.8486514\times10^{-14}T_f^3 + 7.2081991\times10^{-18}T_f^4 \quad \text{....(2)}$$

$$k = 1.5945531\times10^{-3} + 8.726387\times10^{-5}T_f - 2.1443534\times10^{-8}T_f^2 - 1.5194104\times10^{-12}T_f^3 + 2.853825\times10^{-15}T_f^4 \quad \text{...... (3)}$$

$$\Pr = 0.81658609 - 5.4793735\times10^{-4}T_f + 7.5008944\times10^{-7}T_f^2 - 3.9308824\times10^{-10}T_f^3 + 7.0216547\times10^{-14}T_f^{-4} \quad \text{......... (4)}$$

The properties of air can be estimated both at air and film temperature, where the temperature is substituted in the above correlations in Kelvin. The film temperature is calculated as follows [12]:

$$T_f = \frac{T_{b_{in}} + T_{b_{out}}}{2} \quad \text{......... (5)}$$

The Reynolds number (depending on channel height) is computed at the duct exit by the following equation:

$$\mathrm{Re}_H = \frac{U.H}{\nu_f} \qquad \text{......... (6)}$$

where (H) is duct height in (m) and (U) is mean velocity of air at duct exit in (m/sec).

The average temperature of tube or fin is computed by dividing the tube or fin surface into small areas around each thermocouple.

The average tube temperature ($\overline{T}_{tube}$) is:

$$\overline{T}_{tube} = \frac{\sum_{i=1}^{i=5}(T_i A_i)}{\sum_{i=1}^{i=5}(A_i)} \qquad \text{......... (7)}$$

where subscript (i) refers to number of individual thermocouple.

Also, the average fin temperature ($\overline{T}_{fin}$) is:

$$\overline{T}_{fin} = \frac{\sum_{i=6}^{i=12}(T_i A_i)}{\sum_{i=6}^{i=12}(A_i)} \qquad \text{......... (8)}$$

The electric power input to the heater is computed as follows:

$$Power_{input} = I \times V \quad \text{(Watt)} \qquad \text{.......... (9)}$$

The total convective heat transfer from the fin and tube heat exchanger to the air can be computed as follows:

$$Q_{air} = Q_{convection} \qquad \text{.......... (10)}$$

assume

($Q_{loss} = 0$)

$$Q_{convection} = Power_{input} = I \times V \qquad \text{.......... (11)}$$

Computation the average and local heat transfer coefficient of fin and tube is as follows [13]:

$$Q_{convection} = h_{tube} A_{tube}\left(\overline{T}_{tube} - T_f\right) + h_{fin} A_{fin}\left(\overline{T}_{fin} - T_f\right) \qquad \text{.......... (12)}$$

assume

$$h_{tube} = h_{fin} = \overline{h} \qquad \text{............ (13)}$$

$$Q_{covection} = \overline{h}\left[A_{tube} + A_{fin} \times \frac{\left(\overline{T}_{fin} - T_f\right)}{\left(\overline{T}_{tube} - T_f\right)}\right]\left(\overline{T}_{tube} - T_f\right) \qquad \text{............ (14)}$$

The thermal effectiveness of fin (fin efficiency) ;(ϕ) is defined as:

$$\phi = \frac{\overline{T}_{fin} - T_f}{\overline{T}_{finbase} - T_f} \qquad \text{......... (15)}$$

But

$$\overline{T}_{tube} = \overline{T}_{finbase} \qquad \text{......... (16)}$$

Therefore;

$$Q_{convection} = \overline{h}\left(A_{tube} + \phi A_{fin}\right)\left(\overline{T}_{tube} - T_f\right) \qquad \text{... (17)}$$

$$q_{convection} = \frac{Q_{convection}}{\left(A_{tube} + \phi A_{fin}\right)} \qquad \text{......... (18)}$$

where ($q_{convection}$) is the total heat transfer rate to the fluid per unit area (heat flux).

The average heat transfer coefficient ($\overline{h}$) for fin and tube heat exchanger is estimated as follows:

$$\overline{h} = \frac{q_{convection}}{\left(\overline{T}_{tube} - T_f\right)} \qquad \text{......... (19)}$$

and local heat transfer coefficient for fin or tube surface is as in the following:

$$h_i = \frac{q_{convection}}{\left(T_i - T_f\right)} \qquad \text{......... (20)}$$

The average and local Nusselt number (depending on channel height) of the fin and tube is computed as in following:

$$\overline{Nu}_H = \frac{\overline{h}.H}{k_f} \qquad \text{......... (21)}$$

where k_f is the thermal conductivity of the air calculated at film temperature.

The local Nusselt number of fin or tube is:

$$Nu_i = \frac{h_i . H}{k_f} \quad \text{......... (22)}$$

Computation of the friction factor in the test section is as in the following [14]:

$$f = \frac{\Delta p}{\frac{1}{2}\rho U^2} \times \frac{D_h}{L} \quad \text{......... (23)}$$

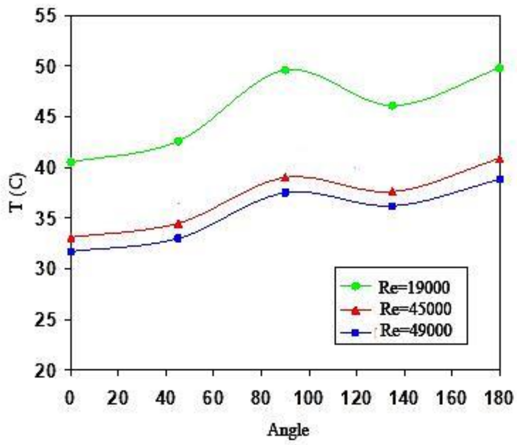
4. Results and Discussion

In the experiment, a rectangular duct is used to house a fin and tube heat exchanger that is fed with cold air that is kept at a controlled temperature between 22 and 25 °C, either with or without vortex generators. The 500-watt heating element is put within the tube. Four thermocouples around the tube and six thermocouples in the surface of the fins make up the ten thermocouples used to measure wall temperature, and two thermocouples are used to measure intake and output air temperatures along with thermometers for increased accuracy. In order to quantify the pressure drop across the duct, two pressure taps are further installed at equal distances before and after the fin and tube heat exchanger.

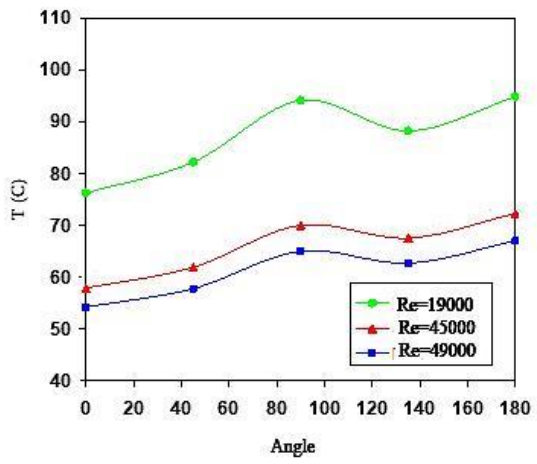
4.1 Temperature Distribution Around the Tube

The stagnation point (θ =0°) typically has the lowest temperature at a single tube with an air flow stream provided to it because of the incoming cold fluid flow. The temperature rises towards the rear as the boundary layer thickness increases (thermal resistance), peaking at the separation point (90°). Recirculation flow (reverse flow), which is present over the rear side of the tube where the wake's eddies sweep the surface, causes a drop after the separation point, but this decrease will not last since the eddies recirculate some of the heated fluid. But when using winglet pair, the maximum temperature will be controlled due to delay or prevent the separation of flow as well as guiding the flow to move toward the rear of tube.

The influence of Reynolds number on the temperature distribution around the tube for the fin and tube case (without winglet) at various heat transfer rate values is depicted in Figure (4-1). For all values of the Reynolds number (Re) and heat transfer rate (Q), the temperature distribution throughout the tube is essentially the same, with the minimum value occurring at the front stagnation point (θ=0°) and rising due to the development of the boundary layer up to the separation point. Beyond the separation point, the temperature drops as a result of recirculating flow, but eddies circulate some heat fluid towards the back of the tube, causing a little rise in temperature in the wake zone behind the tube (θ= 180°). The temperature decreases with increasing Reynolds number due to increase the flow velocity around the tube and decreasing the boundary layer thickness. It is known that the temperature distribution around the tube and also on the fin surface increases with increasing the power supplied to the heating element inside the tube (Q) due to increase the rate of heat transfer to the tube and fin wall from the heater at all (Re)



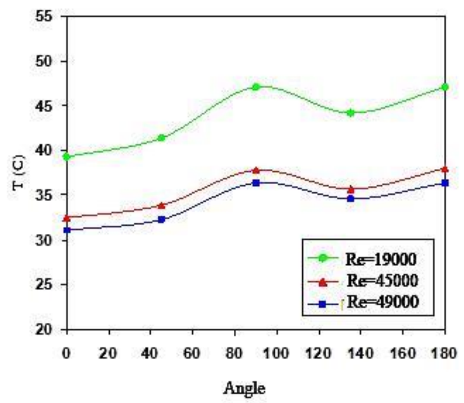
(a)



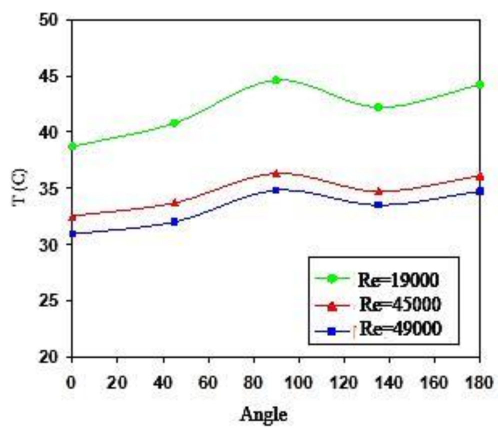
(b)

Figure (4-1) Variation of the Temperature Distribution Around the Tube with Reynolds number for Fin &Tube (without winglet) Case at: (a) Q=10W (b) Q=40W .

The fluctuation in temperature distribution throughout the tube is depicted in Figures (4-2 and 4-3) with a Reynolds number of (Q=10W) and two angles of attack (45° and 60°) for all shapes. With the exception of the temperature values, the temperature distribution around the tube is generally the same for all forms and as for the fin and tube (without vortex generator) instance in figure (4-1). The Reynolds number (Re), heat transfer rate (Q), and angle of attack (ß) all affect the temperature value. While the temperature value increases with a rise in the power supplied to the heater, the temperature value decreases as Reynolds number increases because the heat of the surface will transfer at a higher rate with accelerating the flow velocity. This is because the heat transfer rate to the tube and fin surface will increase as (Q) increases. Due to the effect of the vortices created by the winglet, the extended circumference of the tube that is most affected by angle of attack is extended approximately (90°–180°), whereas the extended circumference of the tube that is farthest from the winglet is less or not affected by angle of attack. The temperature values of interest decrease with increasing angle of attack for all shapes due to increasing the momentum of flow pass the tube. These temperature values are decreased by about (6.1)% at an angle of attack of (ß=45°) and (12.3)% at (ß=60°) for triangular type vortex generator, while reduced by (11.4)% at (ß=30°) and (16.2)% at (ß=60°) for rectangular type.

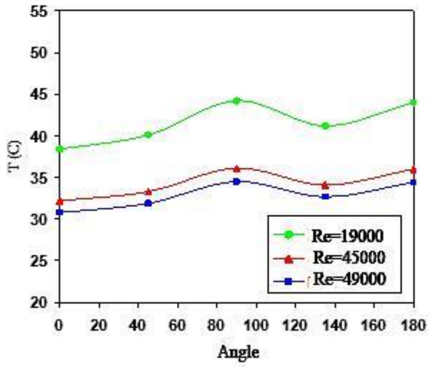


(a)

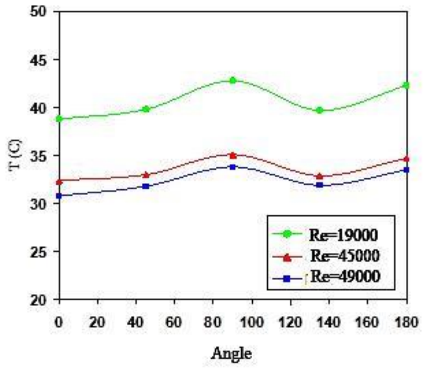


(b)

Figure (4-2) Variation of the Temperature Distribution Around the Tube with Reynolds number for Triangular Type Vortex Generator Case for Q=10W at : (a) β=45° (b) β=60°.



(a)



(b)

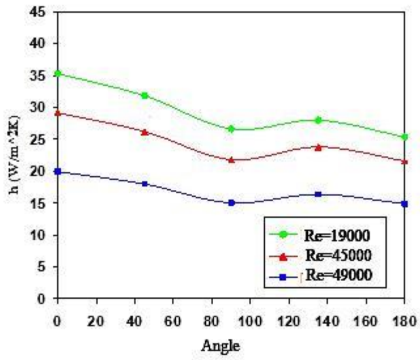
Figure (4-3) Variation of the Temperature Distribution Around the Tube with Reynolds number for Rectangular Type Vortex Generator Case for Q=10W at : (a) β=45° (b) β=60°.

4.2 Local Heat Transfer Distribution Around the Tube

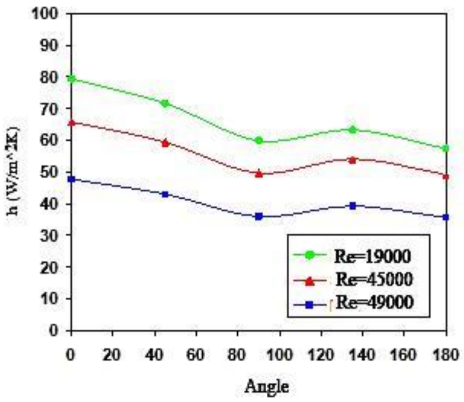
Due to rising boundary layer thickness (thermal resistance), the local heat transfer coefficient from a single tube has its highest value at the front stagnation point (θ=0°) and decreases with distance around the tube. The tube's sides (where the separation point occurs) are where the heat transfer coefficient is at its lowest; however, following this point, the local heat transfer will increase due to the presence of recirculation flow (also known as reverse flow) over the tube's back side, where the wake's eddies sweep the surface. However, because the eddies circulate some of the heated fluid, the heat transfer coefficient over the back is not greater than in the front. Due to the thinning of the boundary layer brought on by the increase in (Re), the flow's capacity for convective heat from the tube also increases. The average temperature difference (denominator) between the wall (including fin and tube) and air increases as (Q) increases because the average wall temperature rises relative to the average air bulk temperature, increasing the heat flux (numerator of the energy balance equation). However, because the rate of heat transfer is increasing faster than the typical temperature difference between a wall and the surrounding air, the heat transfer coefficient rises as (Q) grows.

Figure (4-4) shows the effect of Reynolds number on local heat transfer coefficient around the tube for fin and tube (without winglet) case at different values of (Q). In general, local heat transfer coefficient distribution around the tube is the same of all (Re) and (Q) values, where the results display that (h) is higher in stagnation point and decrease downstream of this point around the tube through growth of boundary layer thickness and reaches minimum value at the separation point (θ=90°), then increases due to reverse flow that dominant in the rear of tube. The heat transfer coefficient increases with increasing (Re) and (Q) values for all cases, as mentioned before.

Figures (4-5) and (4-6) depict the fluctuation in the heat transfer coefficient around the tube for all forms at (Q=8W) and two angles of attack (30° and 60°). The distribution of the local heat transfer coefficients is identical to that shown in figure (4-4) other than the value of (h). The average temperature difference between the wall and air bulk is decreased because the air bulk temperature is slightly influenced, and under the condition of the same heat transfer rate, the heat transfer coefficient will be increased by using vortex generators. This is because the temperature distribution around the tube (especially 90°180°) is decreased by using different shapes of vortex generators and different angles of attack (as mentioned in the previous section). With rising Reynolds number (Re) and heat transfer rate (Q), the heat transfer coefficient also rises. For triangular type tubes, the heat transfer coefficient is increased by roughly 7.4% in comparison to the case of flow over tube without winglet, followed by 18.2% for rectangular type tubes at Re of (19000), Q of (10W), and angle of attack (ß=45°). Utilizing vortex generators to accelerate the flow and sweep the surface of the tube results in an increase in the flow capacity for convective heat transfer, which is the primary reason for improving heat transfer in the rear of the tube.

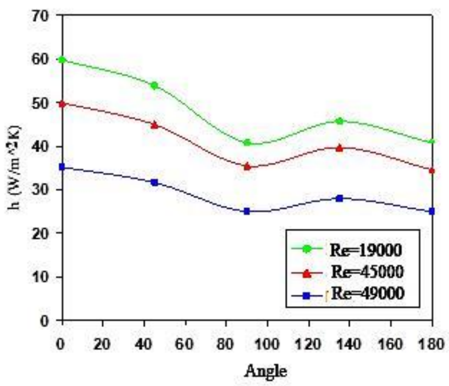


(a)

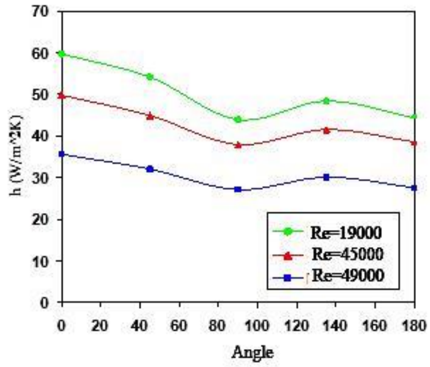


(b)

Figure (4-4) Variation of Heat Transfer Coefficient Around the Tube for Fin &Tube (without winglet) Case with Reynolds number at: (a) Q=10W (b) Q=40W.

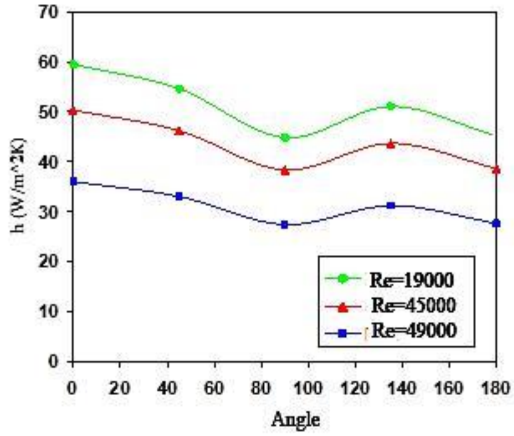


(a)

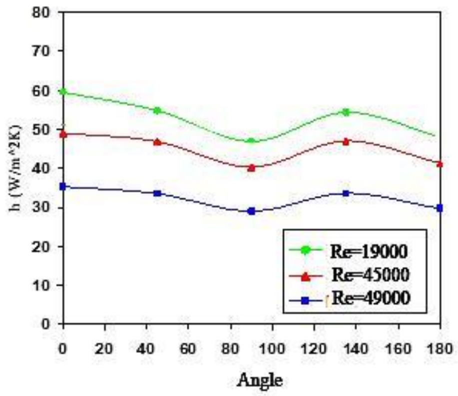


(b)

Figure (4-5) Variation of Heat Transfer Coefficient Around the Tube for Triangular Type Vortex Generator Case with Reynolds number for Q=8W at: (a) β=45° (b) β=60°.



(a)



(b)

Figure (4-6) Variation of Heat Transfer Coefficient Around the Tube for Rectangular Type Vortex Generator Case with Reynolds number for Q=10W at: (a) β=45° (b) β=60°.

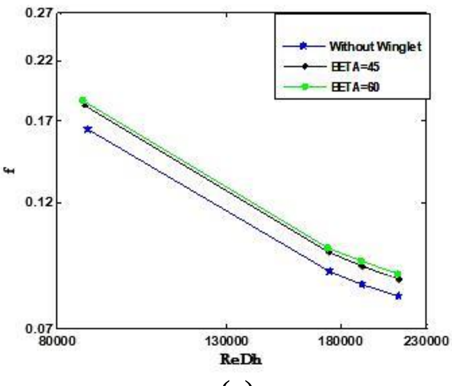
4.3 Friction Factor

The average and local Nusselt figures show that each of the previously discussed examples significantly increases heat transport. This benefit is not without trade-offs, though. Due to the form drag of the vortex generators, using them increases pressure drop across the channel at the expense of increased heat transfer; however, for a plain fin and tube heat exchanger, where the tube is a major source of drag, this incremental increase in pressure drop is relatively small. The local resistance of the tubes (flow separation from the tube results in a large pressure loss), as well as the friction resistance of the fin surface, contribute to the heat exchanger's air flow resistance.

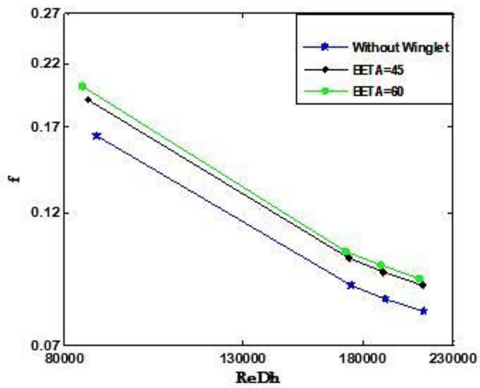
The primary contributor to the overall pressure drop is the local resistance of the tubes. When the winglets are fitted, the local resistance of the winglets as well as the aforementioned two factors contribute to the air flow resistance of the heat exchanger. Winglets have two effects: on the one hand, they increase form drag; on the other, they delay the separation of the boundary layer from the circular tube and reduce the wake zone behind the tube, reducing form drag from the tubes. Therefore, it is uncertain if using winglets will result in an increase or decrease in the overall pressure drop. In this experiment, the pressure drop is increased by all forms and attack angles. The pressure drop results are presented inform of friction factor and plotted as a function of Reynolds number dependent on the hydraulic diameter and compared with fin and tube (without winglet) case. No effect was observed on the pressure difference as changing the heat transfer rate (Q) values throughout the experiment, this means that the pressure drop measurement is independent of the temperature measurement.

The influence of angle of attack on the fluctuation of friction factor with Reynolds number for all forms is shown in Figure (4–7). The relationship between friction factor and Reynolds number usually always follows the same trend, demonstrating that the two variables are inversely proportional. It is true that as (Re) grows, the fluid's velocity will also increase, which raises the static pressure differential across the duct. The static pressure across the duct is anticipated to decrease when the winglets are attached to the fin-tube heat exchanger, increasing the pressure drop across the duct in comparison to the baseline situation. Due to the acceleration of the flow towards the tube and in the wake zone as (ß) increases, the pressure decreases for all forms when the angle of attack is increased. It is well known that increasing the angle of attack causes the pressure difference between the winglet's suction surface and front surface that faces the flow to grow, which in turn strengthens the vortex and increases resistance to the flow. At angles of attack (45° and 60°, respectively) and in the measured Reynolds number range, the friction factor increases relative to the (without winglet) case by about (9.2 and 12.6%) for the triangular type and (14.1 and 17.5%) % for the rectangular type vortex generator.

Figure (4-8) illustrates the relationship between the geometry of the vortex generator and the variation in friction factor with Reynolds number for various angles of attack. The friction factor rises over the range of Reynolds numbers and at ß=45° for the triangular and rectangular vortex generators, respectively, by almost (10.4 and 12.2) % and (13.3 and 17.2) %. The results show that the rectangular type, which have a maximum pressure drop because of an increase in controlling the flow to pass at high velocity towards the tube and wake region, which means at this shape (rectangular) there is no enough space to pass the flow, leads to accelerate the flow, while the triangular type has a minimum pressure drop across the duct than the other.

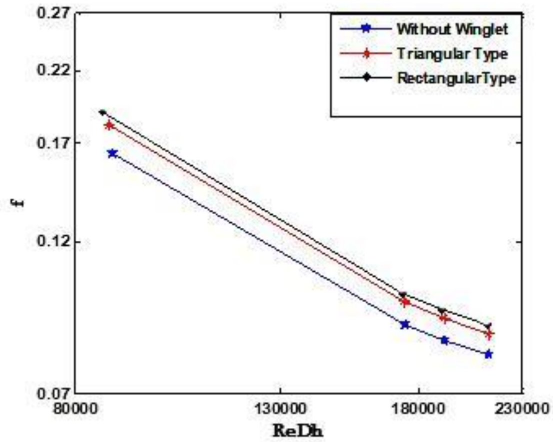


(a)

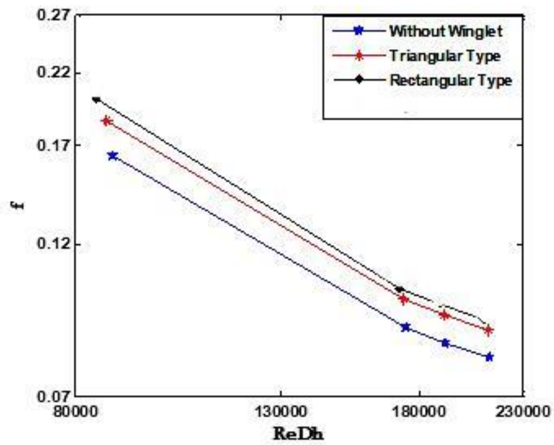


(b)

Figure (4-7) The relationship between Friction factor & Reynolds number with different Angles of Attack for : (a)Triangular Type, (b) Rectangular Vortex Generator.



(a)



(b)

Figure (4-8) The relationship between Friction factor & Reynolds number for all shapes with different Angles of Attack: (a) β=45°, (b) β=60°.

5. Conclusions

The following conclusions can be extracted depending on the results obtained from the current work:

1- There is no effect of varying heat flux on the both heat transfer enhancement parentage and friction factor for different angles of attack and flow velocity.

2- With increasing angle of attack, the pressure drop across the duct likewise increases, as does the heat transfer coefficient.

3- The triangle type gives minimum heat transfer with respect to other shapes, where the average Nusselt number of fin and tube heat exchanger is enhanced by (16.7-17.8)%for the rectangular type and by (9.3-10.6)%for the triangular type at an angle of attack (β=60°) with respect to (without winglet) case for Reynolds number ranging between (19000–49000).

4- Whereas the friction factor in the duct is increased by (16.5%) % for the rectangular type and by (11.1%%) % for the triangular type at an angle of attack (θ=60°) with respect to the (without winglet) case for Reynolds number ranging between (19000-49000), the maximum pressure drop across the duct occurs in the rectangular type, while the minimum pressure drop occurs in the triangular type.

5- Heat transfer coefficient increases with increasing angle of attack, also the pressure drop across the duct shows an increase.

The following ideas are presented as a continuation of the current investigation into integral low finned tubes for future research projects:

1- Improving heat transfer coefficient of fin and tube heat exchanger by using other shapes of vortex generators (such as spherical and corrugated shapes) can be studied.

2- Studying the effect of winglet pairs on heat transfer and pressure drop from tube banks.

3- Replacing the circular tube with an oval tube and testing its effect on heat transfer and pressure drop.

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