Al-Khwarizmi Engineering Journal, Vol. 8, No. 2, PP 12- 29 (2012)

Al-Khwarizmi Engineering Journal

An Experimental Study on the Effect of Shape and Location of Vortex Generators Ahead of a Heat Exchanger

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(Received 15 September 2011; accepted 30 January 2012)

Abstract

An experimental study is carried out on the effect of vortex generators (Circular and square) on the flow and heat transfer at variable locations at (X = 0.5, 1.5, 2.5 cm) ahead of a heat exchanger with Reynolds number ranging from $62000 < Re < 125000$ and heat flux from $3000 \le q \le 8000$ W/m^2.

In the experimental investigation, an apparatus is set up to measure the velocity and temperatures around the heat exchanger.

The results show that there is an effect for using vortex generators on heat transfer. Also, heat transfer depends on the shape and location. The circular is found to be the best shape for enhancing heat transfer at location [X_m=0.5 cm] distance before heat exchanger is the best location for enhancing heat transfer. The square is the best shape for enhancing heat transfer at location [X_m=2.5 cm] distance before heat exchanger is the best location for enhancing heat transfer.

The results of flow over heat exchanger with vortex generators are compared with the flow over heat exchanger without vortex generators. Heat transfer around heat exchanger is enhanced (56%, 50%, 36%) at location (X=0.5, 1.5, 2.5cm) respectively by using circular vortex generators without turbulator and heat transfer around heat exchanger is enhanced (39%, 42%, 51%) at location (X=0.5, 1.5, 2.5cm) respectively by using square shape vortex generators without turbulator.

Keywords: *Vortex generator: VGs, NUX: Local nusselt number heat exchanger, vortex flow, heat transfer.*

1. Introduction

Convectional types of heat exchanger are externally plain tube in cross flow. They are widely used in chemical, petrochemical, automotive industry, cooling towers, heated pipes refrigerators of power plants as well as in applications for heating, refrigeration and air conditioning. In finned tube heat exchanger liquid or steam flows through the tube and gas through fin ducts, [1], [2]. Heat transfer is closely related to fluid dynamics. That is why heat transfer is considered simultaneously with fluid dynamics [3].

Heat transfer and fluid dynamic around curvilinear body as cylinder is complex process and need a big efforts to find out temperatures, pressure and velocity distribution. Therefore one must know what happens when the fluid flows over bluff bodies as sphere, wire and tube.

The ability to manipulate a flow field to improve efficiency or performance is of immense technological importance. Flow control is one of the leading areas of research of many scientists and engineers in fluid mechanics. The potential benefits of flow control include performance and maneuverability, affordability, increased range and payload, and environmental compliance. The intent of flow control may be to delay/advance transition, to suppress / enhance turbulence, or to prevent/promote separation. The resulting benefits include drag reduction, lift enhancement, mixing augmentation, heat transfer enhancement, and flow–induced noise suppression. The objectives of flow control may be interrelated, leading to potential conflicts as the achievement of one particular goal may adversely affect another goal. For example, consider an aircraft wing for which the performance is measured by the improvement in lift-to-drag ratio. Promoting transition will lead to a turbulent boundary layer that is more resistant to separation and increased lift can be obtained at higher angle of incidence. The viscous or skin – friction drag for a laminar boundary layer can be an order of magnitude smaller than that for a turbulent boundary layer. However, a laminar boundary layer is more prone to separation resulting in a loss in lift and an increase in form drag.

The performance of liquid-to-air and two-phase-to-air heat exchangers is important in many applications, including thermal management and processing systems found in the air conditioning, automotive, refrigeration, chemical, and petroleum industries. Improving the performance of these heat exchangers can lead to a smaller surface-area requirement, reduced material cost, and a lower heat exchanger mass. Furthermore, improving heat exchanger performance can have a significant impact on the environment through improvements in energy efficiency. The total thermal resistance in these heat exchangers can be considered as the sum of three contributions: the liquid or two-phase convective resistance, the wall conductive resistance, and the air-side convective resistance. The air-side convective resistance is typically the dominant resistance to heat transfer,[4], and efforts to improve these heat exchangers should focus on the air - side heat-transfer behavior. The type of turbulator using heat exchanger is shown in Figure (1). [5]

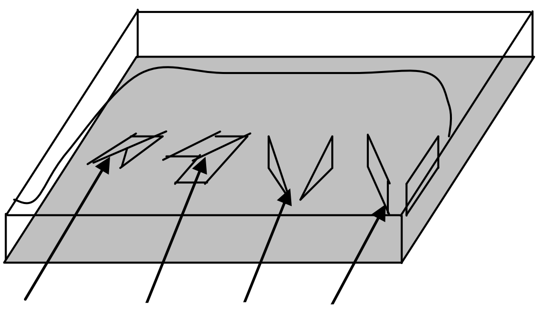


Fig. 1.Type of Vortex Generator.

1.1. Vortex Generation

At the front of region of tube there is high heat transfer due to small boundary layer (B.L) thickness but there is a small heat transfer at the rear region especially at separation point. This region is therefore prime focus area for transport enhancement, [4]. The strategy proposed here involves placing the vortex generators (V.Gs) in front of or behind the tube to prevent growth of boundary layer (B.L) and to transfer the separation point to the rear region of tube in order to reject the minimum heat transfer region to the rear. The (V.Gs) are small plates placed in the stream flow of mixed flow, disturbing flow and controlling the growth of boundary layer, [6], has any shape; the common shapes of (V.Gs) (winglets) are circular and square. The (V.Gs) winglets affect both tube and fin, all previous studies were studied the effect of wing and winglet on heat transfer from the fin by generating a vortex due to pressure difference between front surface and back surface, this vortex will mix the hot fluid near the surfaces with cold main flow, this process enhances heat transition from the surfaces, [7]. The effect of winglet on tube guides the flow at high momentum in to low heat transfer region, [6].The size, shape and angle of attack of winglet determine the specific characteristics of the vortices generated in the flow, [8].

1. 2. Application of VGs

The advantages of these vortex generators or tabulators (turbulence promoters) are to:

1. Improve heat exchange in compact heat exchangers and electronic equipment packages or microelectronic devices in industrial application ,[9] like the wide use of plate-fin and fin-tube heat exchangers, for example, in dry cooling towers, in chemical industry and in automotive applications ,[10].
2. Enhance heat transfer in channels. e.g, parallel plate channel, rectangular, triangular, square ducts, U-Bend of strong curvature applications, and grooved channel, [11].
3. Increase heat transfer rate inside or out side tubes, for example, gas flow out side the tube and liquid flow inside the tube and the finned tubes situated in vertical channel.
4. Enhance the cooling capability of gas-cooled nuclear rector, for example, finned nuclear fuel.

5. Increase internal cooling in the passages of modern gas turbine blades and vanes that must be protected from hot gas streams.
6. Improve the aerodynamic performance by using various types of vortex generators, for example, to improve the performance of conical diffusers, or by using thin slender wings to make modern combat airplanes fly at high angles of attack.
7. Decrease vortex losses in channels, for example, in channels of power plants, ventilation systems, and in various pipes, owing to the influence of the positive pressure gradient associated with variation in cross section or bending of the channel intense formation of vortices which takes place as a rule due to flow separation. These formations of vortices cause an increase in hydraulic losses and in degree of no uniformity. A new method of decreasing losses is based on division of vortices by transverse baffles or fins positioned on one side [12].
8. In aerodynamics, in spite of longitudinal (stream wise) vortices, which lead to an improvement of 80%, also a reduction of 5% behind the investigated grids depends on the wavelength and the intensity of the disturbance [13].

1.3. Applications of Heat Exchangers

Flow of this nature can be found in engineering systems of significant technological interest such as heat exchangers, nuclear reactor cores, air-cooled solar collectors, some microelectronic circuit boards, waste water aeration tanks as well as chemical mixers and other chemical engineering applications [14].

1.4. Objective of Present Work Problem

The experimental goals of this study are to: -

1-Investigate the flow and heat transfer phenomena of multi-types vortex generators located in rectangular ducts and to enhance heat transfer in turbulent flows.

2-Set up an experiment model for a duct rig especially designed and manufactured for this study to cover wide range of applications with more accurate tools of measurements for flow pattern and temperature isothermal contours in air and solid domains.

3-Investigate the effect of changing Reynolds number, velocity, temperature, and thermal performance, and average Nusselt number on ducts and VGs surfaces.

2. Literature Survey

2.1. The Effect of Vortex on Heat Exchanger

Chien-Nan Lin and Jiin-Yuh Jang [15] studied the use of fins with embedded wave-type vortex generators to enhance heat transfer in fin-tube heat exchangers .An infrared thermo vision is used to visualize the temperature distribution on the surface of a scaled-up plain fin and upon fins with embedded vortex generators. Numerical methods are used to investigate the conjugate heat transfer and to perform a 3-D turbulence.

Analysis of the heat transfer and fluid flow associated with wave type vortex generators embedded fins. The current results indicate that heat transfer and friction losses are strongly dependent on the geometric parameters of the vortex generators. This study identifies maximum improvement of (120) % in the local heat transfer coefficient and an improvement of (18.5) % in the average heat transfer coefficient. Furthermore; it is found that a reduction in fin area of approximately (18–20) % may be obtained if vortex generators embedded fins are used in place of plain fins. Finally, it is noted that the magnitude of the attainable fin area reduction increases for higher Reynolds numbers.

Pesteei, et al [16] measured Local heat transfer coefficients on fin-tube heat exchanger with winglets using a single heater of 2 inch diameter and five different positions of winglet type vortex generators. The measurements were made at Reynolds number about 2250. Flow losses were determined by measuring the static pressure drop in the system. Results showed a substantial increase in the heat transfer with winglet type vortex generators. It was observed that average Nusselt number increased by about (46) % while the local heat transfer coefficient was improved by several times as compared to plain fin-tube heat exchanger. The maximum improvement is observed in the re-circulation zone. The best location of the winglets was with DX = 0.5D and DY = 0.5D. The increase in pressure drop for the existing situation was of the order of (18) %.

Torii, et al [17], in their paper propose that can augment heat transfer but nevertheless can reduce pressure-loss in a fin-tube heat exchanger with circular tubes in a relatively low Reynolds number flow, by deploying delta winglet-type vortex generators. The winglets are placed with a heretofore-unused orientation for the purpose of augmentation of heat transfer. This orientation is called ‘‘common flow up’’ configuration. The proposed configuration causes significant separation delay, reduces form drag, and removes the zone of poor heat transfer from the near-wake of the tubes. This enhancement strategy has been successfully verified by experiments in the proposed configuration. In case of staggered tube banks, the heat transfer was augmented by (30) % to (10) %, and yet the pressure loss was reduced by (55) % to (34) % for the Reynolds number (based on two times channel height) ranging from 350 to 2100, when the present winglets were added. In case of in-line tube banks, these were found to be (20%) to (10%) augmentation, and (15%) to (8%) reduction, respectively.

In a Joardar and Jacobi [18] the effectiveness of delta-wing type vortex generators was experimentally evaluated by full-scale wind-tunnel testing of a compact heat exchanger typical to those used in automotive systems. The mechanisms important to vortex enhancement methods are discussed, and a basis for selecting a delta-wing design as a vortex generator is established. The heat transfer and pressure drop performance are assessed at full scale under both dry- and wet-surface conditions for a louvered- fin baseline and for a vortex-enhanced louvered-fin heat exchanger. An average heat transfer increase over the baseline case of (21) % for dry conditions and (23.4) % for wet conditions was achieved with a pressure drop penalty smaller than (7) %. Vortex generation is proven to provide an improved thermal-hydraulic performance in compact heat exchangers for automotive systems.

2.2. Wing and Winglet–Type Vortex Generators

Heat transfer and fluid mechanics data were obtained by Pauley and Eaton [19], for a turbulent boundary layer with arrays of embedded stream wise vortices containing both counter-rotating and co-rotating vortex pairs. The data show that these arrays can cause both large local variations in the heat transfer rate and significant net heat transfer augmentation over large areas. Close proximity of other vortices strongly affects the development of the vortex arrays by modifying the trajectory that they follow. The vortices in turn produce strong distortion of the normal two-dimensional boundary layer structure, which is due to their secondary flow. When one vortex convects another toward the wall, a strong boundary layer distortion occurs. The heat transfer is elevated where the secondary flow is directed toward the wall and reduced where the secondary flow is directed away from the wall. When adjacent vortices lift their neighbor away from the wall, minimal modification of the heat transfer results . The primary influence of grouping multiple vortex pairs into arrays is the development of stable patterns of vortices. These stable vortex patterns produce vortices that interact with the boundary layer and strongly modify the heat transfer far downstream, even where the vortices have decayed in strength.

Vortex generators are small plats placed in the flow path (Sohal et al)[7] **,** using to generate a secondary flow or vortices by swirl and disability the flow, along the side edges of vortex generator the flow was separated and it generated a longitudinal vortices due to pressure difference between the forward and back side of vortex generators.

Vortex generators such as pins, ribs, wings, winglets have been successfully used as a powerful way for enhanced heat transfer in the development of modern heat exchangers. Vortex generators can be generating a transfer longitudinal vortex with strongly disturbed the flow structure. Longitudinal vortex generated by using wing and winglet and spiral the flow around there axis. Many researchers have studied the effect of vortex generator on heat transfer and pressure drop from the duct wall numerically and experimentally with common type of vortex generated (rectangular and triangular (delta) shapes).

The analogy between heat and mass transfer has been used by Wang et al, [20], to obtain local and average heat transfer characteristics over a complete flat tube-fin element with four VGs per tube. Several types of surfaces involved in heat transfer process such as fin surface mounted with VGs, its back surface (mounted without VGs) and flat tube surface are considered. The mass transfer experiments are performed using naphthalene sublimation method. The effects of the fin spacing and VGs parameters such as height and attack angle on heat transfer and pressure drop are investigated. The comparisons of heat transfer enhancement with flat tube-fin element without VGs enhancement under three constraints are carried out. The local Nusselt number distribution reveals that VGs can efficiently enhance the heat transfer in the region near flat tube on fin surface mounted with VGs. On its back surface the enhancement is almost the same as on the fin surface mounted with VGs but enhanced region is away from flat tube wall with some distance. Average results reveal that increasing of VGs height and attack angle increases the enhancement of heat transfer and pressure drop, whereas small fin spacing causes greater increase in pressure drop. The heat transfer performance, correlations of Nusselt number and friction factor are also given.

Tiggelbeck et al, [21], Investigated the effect of four types of vortex generators (delta wing, delta winglet rectangular wing and rectangular winglet) as shown in Figure (2) on local heat transfer and drag of plat fins and compared the results between these types of vortex generators and in the Reynolds number range (2000-9000) with angle of attack range of (30°-90°). The results show that the vortex generators increase heat transfer and also 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 pair of delta winglet performs slightly better than the pair of rectangular winglets.

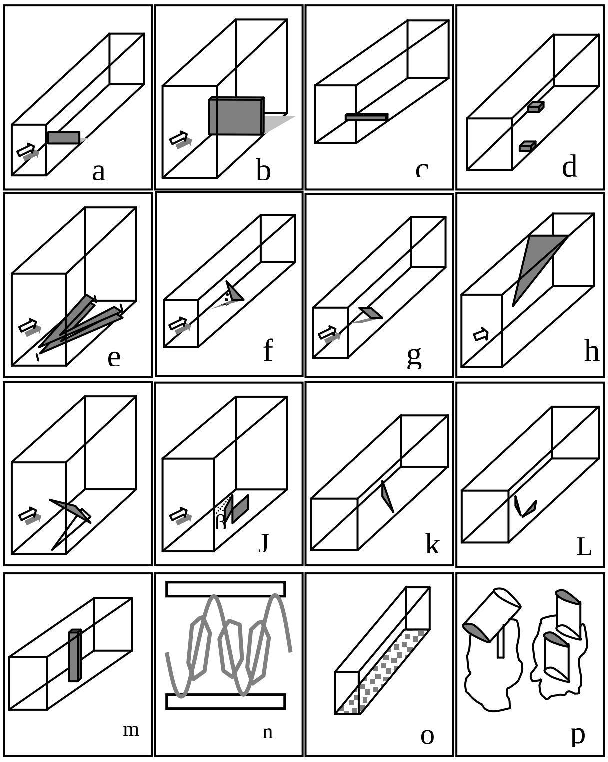


Fig. 2. Triangle and Rectangle Wings and Winglets.

2.3. Vortex Phenomenon

Davis and Moore [22], presented numerical solutions for two-dimensional time-dependent flow about rectangles in infinite domains . The numerical method utilizes a third order upwind differencing for convection and a Leith type of temporal differencing. An attempted use of a lower-order scheme and its inadequacies are also described. The Reynolds-number regime investigated is from 100 to 2,800. Other parameters that are varied are upstream velocity profile, angle of attack, and rectangle dimensions. The initiation and subsequent development of the vortex-shedding phenomenon was investigated. Passive marker particles provide an exceptional visualization of the evolution of the vortices both during and after they are shed. The properties of these vortices are found to be strongly dependent on Reynolds number, as are lift, drag, and Strouhal number. Computed Strouhal numbers were compared well with those obtained from a wind-tunnel test for Reynolds numbers below (1,000).

Yasuo Mori et al, [23], studied experimentally the mechanism of symmetrical vortex shedding behind a cylinder in a uniform, upward flow by heating the wake with fine wires or by using a splitter plate or mesh. In the case of heating the wake the vortices are gradually modified from Karman vortex to symmetrical vortices with increase of heat input. Similar symmetrical vortex shedding is also observed by the increase of the splitter plate length or mesh number behind the cylinder.

An experimental investigation of a longitudinal vortex/boundary layer system was made by Eibeck and Eaton [24], to understand the effects of the vortex on local convective heat transfer coefficients. Measurements in the presence of a single longitudinal vortex embedded in an otherwise two-dimensional turbulent boundary layer included local Stanton number distributions, momentum and thermal boundary layer profiles, and skin friction distributions. The local Stanton number varied, with an increase of (22) % over flat plate values in the downwash region of the vortex, and decreases of (12) % in the up wash region. The vortex imposed spanwise variations of boundary layer parameters such as thickness, wall shear, and profile shape. In spite of this, the heat transfer process was locally dominated by two-dimensional mechanisms, as evidenced by the existence of a log-region in the boundary layer, as well as the applicability of the Reynolds analogy.

The heat transfer effects on an isolated longitudinal vortex embedded in a turbulent boundary layer were examined experimentally by Eibeck and Eaton [25], for vortex circulations ranging from (0.12 to 0.86). The test facility consisted of a two-dimensional boundary-layer wind tunnel, with a vortex introduced into the flow by a half-delta wing protruding from the surface. In all cases, the vortex size was of the same order as the boundary-layer thickness. Heat transfer measurements were made using a constant-heat-flux surface with 160 embedded thermocouples to provide high resolution of the surface-temperature distribution. Three-component mean- velocity measurements were made using a four-hole pressure probe. Span wise profiles of the Stanton number showed local increases as large as 24 percent and decreases of approximately 14 percent. The perturbation to the Stanton number was persistent to the end of the test section, a length of over 100 initial boundary-layer thicknesses. The weakest vortices examined showed smaller heat transfer effects, but the Stanton number profiles were nearly identical for the three cases with circulation greater than $\Gamma/U_\infty\delta_{99}$= 0.53. The local increase in the Stanton number is attributed to a thinning of the boundary layer on the downwash side of the vortex.

Experimental studies on bodies of revolution at high angles of attack have shown that the forces and moments developed are greatly affected by the formation of rolled up vortex cores. Thus it is felt that an accurate model of the vortex would aid in the design of fuselage fore bodies or slender bodies in general. Due to geometric simplicity several mathematical models of the vortex flow over a slender, sharp-edged, delta wing have been formulated. However, these models generally ignore the entrainment effect of the vortex core, and are found to yield results which are not in agreement with experiment, thus, their extension to the more general case would be of little value. A technique referred to as the leading-edge suction analogy, has been found to yield extremely accurate results when compared with experiment. Paul [26], indented to propose a mathematical model of the vortex, which incorporates the previously ignored entrainment effect, and leads to an expression similar to the leading-edge suction analogy.

2.4. Scope of the Present Study

The aim of present study is to investigate turbulent flows in rectangular ducts using two-shapes of VGs and different locations and dimensions, to know the effect of these small bodies on flow and heat transfer characteristics. Experimental investigations are to be undertaken in the present study. In the experimental part, none of the previous studies have tackled the problem of constant wall temperature along the duct.

Therefore, a rig is to be built to study the velocities and temperatures for a range of Reynolds number (62000-125000) so that the flow become fully development, using accurate tools and modern measurement equipments and devices that can be calibrated accurately.

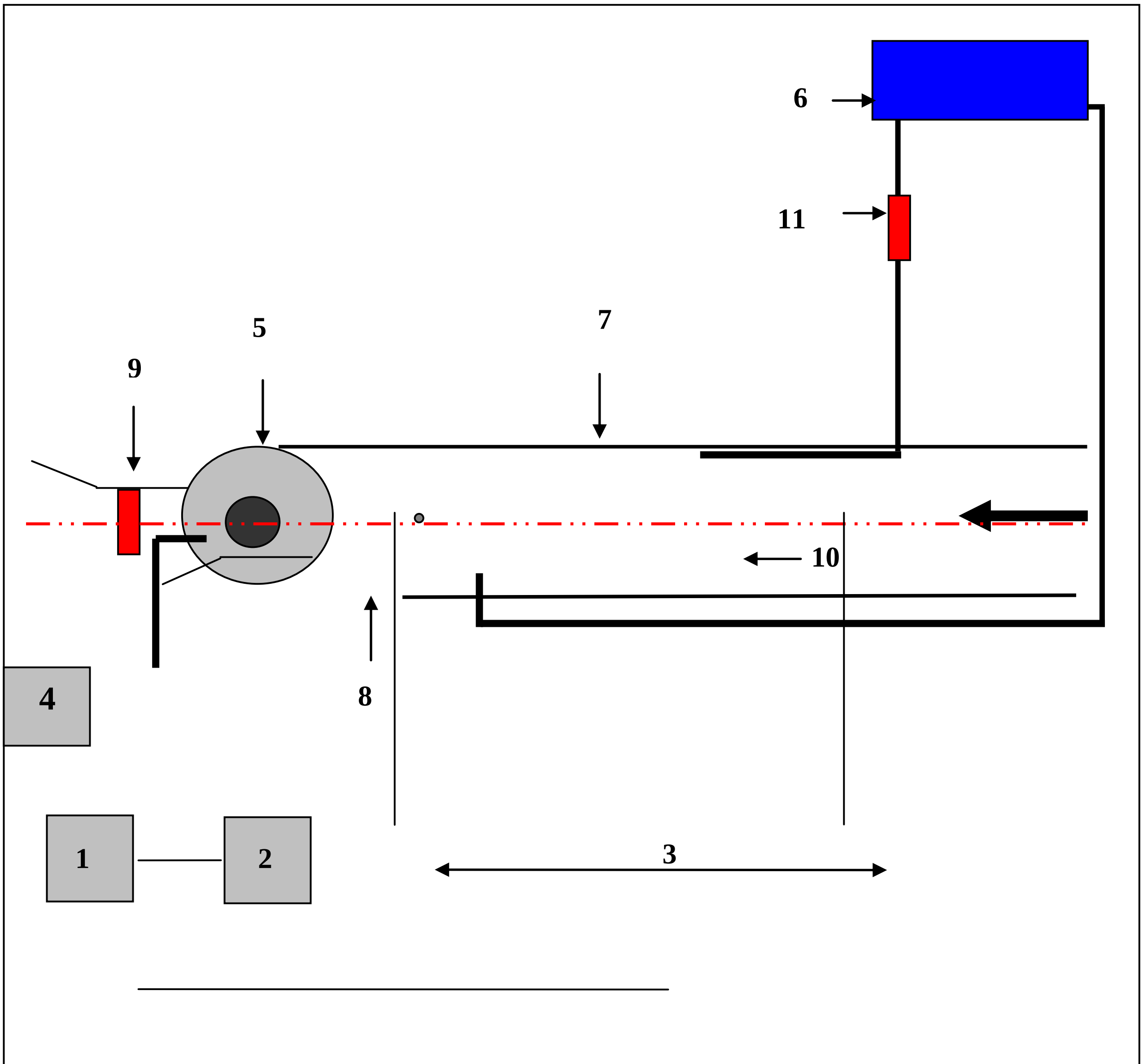
Also we found correlation between the average Nusselt number and Reynolds number (62000-125000) in different vortex generator shapes (circular, square).

Thus, the layout of research can be demonstrated as follows:

1. Set up an experimental rig to measure the velocity, temperatures.
2. Study the effect of vorticities on the heat transfer on the heat exchanger.
3. Study the obstruction that can produce maximum heat transfer.

2.5. Summary

A conclusion of the available data in literatures for effect of vortex generators on heat exchanger, both wing and winglet – type vortex generators and vortex phenomena was presented. All data related to study friction factor, pressure drop, heat transfer from heat exchanger, and relation between Nusselt number and Reynolds number. From this summary and to the author's knowledge no study of the heat transfer and pressure drop through the heat exchanger by using vortex generators, circular and square shapes was conducted.



1- Thermometer calibrator
2-Digital Thermometer
3-Thermocouples
4- Digital anemometer
5- Fan

6- Electrical heater
7-Plian tube heat exchanger
8-Rectangular duct
9-Throttling valve
10- Vortex generators
11- Flow meter

Fig. 3. Scheme Diagram.

3. Experimental Work

3.1. Rig Requirement

A schematic diagram given in Figure (3) shows the experimental arrangement. The experiment was carried out at the heat transfer laboratories of the University of Technology using plain tube heat exchanger with six rows tube staggered and vortex generator. The test model was constructed from rectangular wood duct the dimensions of duct were 0.3m height and 0.15m width and 3m length, and the length of duct before the heat exchanger is 0.3 m.

3.2. Rig Description and preparation

The test rig shown in Figure (4) is designed and manufactured to fulfill the requirements of the test system for finned duct. The experimental apparatus consists basically of:

1-The airflow rates supply section.
2-The test section. (Heat exchanger).
3-The heating and control sections.
4-The measuring instruments.
5-Vortex generators.

Most of these parts are designed and manufactured during the current work, and care was taken to prevent any air leakage between the connected parts; the rig was completely damaged and during the current work so it was rehabilitated reconstructed according to the acceptable wind tunnel standards.

Fig. 4 Rig Description.

3.3. Air Supply Equipment

Suction type axial fan shown in Figure (5) was used to supply the flow of air to the rig working section. The Axial fan [made by Tecquipment in England, se. no. w/4/20029,] is driven by A.C. motor, the flow of air is controlled by using an adjustable throttling valve mounted just at the mouth of the air intake. Since there is no need for any connection in the fan inlet and the throttling valve being fully open to obtain maximum air flow rate , the air delivered by the fan should has a uniform velocity profile with minimum turbulence level.

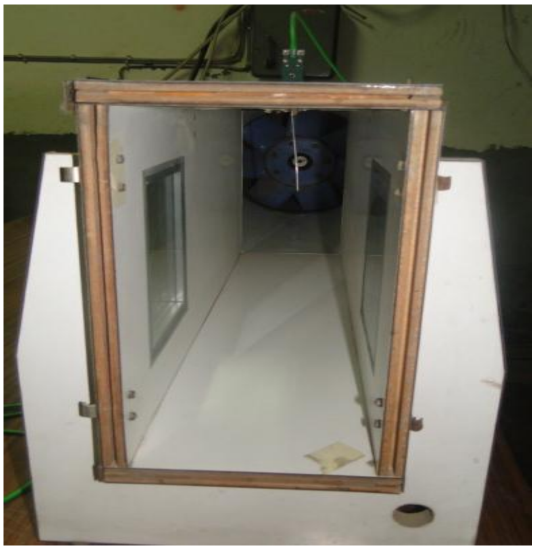


Fig. 5. Axial Fan.

3.4. Air Duct

It consist of a rectangular wood duct as shown in Figure (5) which is constructed of four section clipped tightly together. Identical entry and exist section are separated by a plain center section which can easily be replaced with the optional heat exchanger .The apparatus is supplied with the axial fan connected to the exit . The fan discharges directly to the atmosphere thought an adjustable throttle plate which can be used to vary the volume flow rate. Air flow rate measurement is computed by using a pitot static tube and manometer or anemometer, the dimensions of duct were 0.3m height and 0.15m width and 3m length.

3.5. Plain Tube Heat Exchanger

A bank of pure copper heat transfer tube arranged vertically consists of 33 tube 5/8 inch outer diameter and 1/2 inch inner diameter tube 30 cm length as shown in Figure (6) . Hot water supply and return duct connection are made by means of flexible rubber hoses and plug in connection through heat exchanger. Water temperature is measured by thermocouples.

Fig. 6. Plain Tube Heat Exchanger.

Fig. 7.a Circular Vortex Generator.

3.6. Vortex Generators

The vortex generator was used to generate the longitude vorticities to make difference in pressure between front surface of heat exchanger flow and the end surface causes enhanced heat transfer from heat exchanger. Most studies used two common shapes of winglets, (Circular and square) shapes as shown in Figure (7a&7b) . In the experimental the following things were fixed:-
There were two types of Grid vortex generator

1- One Grid consisting of (21) pieces of small circular, diameter=2.5 cm. The area of small piece of circular is (4.84) cm^2.
2- The other consisting of (21) pieces of small square, length=2.2cm. The area of small piece of square is (4.84) cm^2.

The purpose of this study is to compare the enhancement heat transfer with vortex generator and without vortex generator.

Fig. 7.b Square Vortex Generator.

3.7. Water Pump

Water Pump of [Q=60 L/min]capacity and a (20) m head is used To circulate Hot Water between heat exchanger and water heater by using flexible pipes as shown in Figure (8).

Fig. 8. Water Pump and Heater.

3.8. Supply Unit of Hot Water

Water tank of (150) liter capacity provided with two heaters each of (3) KW capacity, a thermostat is used to control water temperature at $60^{\circ}C$ as shown in Figure (8).

3.9. The Measuring Instruments

3.9.1. Digital Thermometer

This digital thermometer contains Pt 100 ohm thermometer and thermocouple type K/J/R/E/T thermometer. It contains also microcomputer circuit with high performance wide temperature measuring range. It is used to measure temperature. See Figure (4).

3.9.2. Digital Anemometer

This vane-type probe portable anemometer provides fast accurate readings, with digital readability and the convenience of a remote sensor separately. It is used to measure the average air velocity. The low friction ball bearing design allows free vane movement, resulting in accuracy at both high and low velocities. The sensitive balanced vane wheel rotates. Freely in response to air flows. Conventional twisted vane arms eliminate the source of unreliability, see Figure (9).

Fig. 9. Digital Anemometer.

3.9.3. Thermocouples Circuit

The thermocouple circuit consists of a digital electronic thermometer (type TM-200, serial no. 13528) as shown in Figure (4), connected in parallel to the thermocouples by leads through a thermometer, and digital thermometer Calibration (type TM-300, serial no.13645) by using only calibration digital electronic thermometer. Thermocouple (type K), this type can be used in temperature range from -200 to $1300^{\circ}C$,(Chromel 90 % Ni- 10% Cr ; Alumel 95%Ni- 2% Al- 3%Mn).

3.9.4. Flow Meter

To measure the average flow rate of hot water , a Rota meter which is a flow meter containing a calibrated glass tube and a float [0.27 m^3/hr] and a globe valve was used to control the average flow rate of hot water, see figure (10).

Fig. 10. Flow Meter.

3.10. Calibration of Instruments

3.10.1. Thermometer Calibration

This is a two in one device including type K thermometer and type K thermocouple calibrator. It is a calibrating process device and measuring process signals. Microprocessor circuit assures high accuracy and provides special function and features. Built-in temperature linearity compensation high precision circuit is fitted with standard K input measuring socket. It is used to calibrate the other calibratable digital thermometers see Figure (4).

3.10.2. Anemometer Calibration

The vane-type rotary anemometer is calibrated just after the duct test section because of the large cross-sectional area of the probe, in the free stream of the outlet region, for the five Reynolds numbers. The static-pitot tube with the manometer is used in that calibration, for checking. For many stations, the average velocity (U_{ave}) is calculated by integration. The average error percentage for all Reynolds numbers Range is around 2%.

3.10.3. Thermocouples Calibration

All thermocouple were used with leads and calibrated using the melting point of ice made from distilled water as reference point and the boiling points of several pure chemical substances. The calibration results are given in Table (1) and Figure (11a &11b).

Table 1,

Experimental Accuracies.

Independent variables (v)	uncertainty interval (w)
Surface to bulk air temperature	± 0.16 °C
Voltage of the heater	± 0.04 volt
Current of the heater	± 0.0003 Amp
Hydraulic diameter	± 0.0002 m
Velocity	±0.2% m/s
Thermometer	± (0.2 % + 0.5 C)

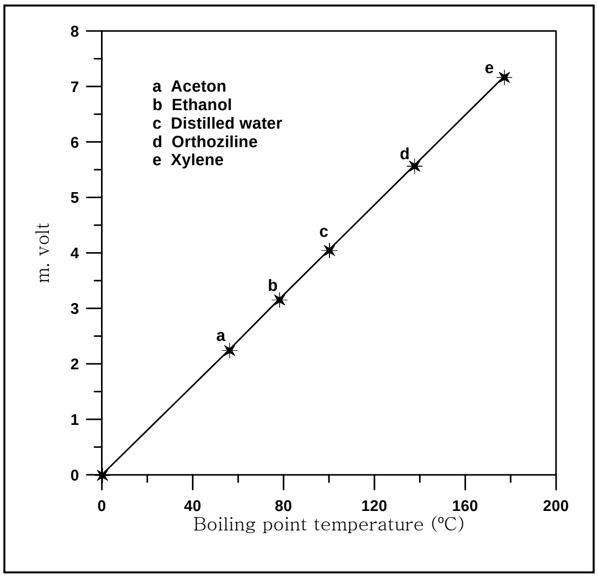


Fig. 11. a. Thermocouple Calibration Results.

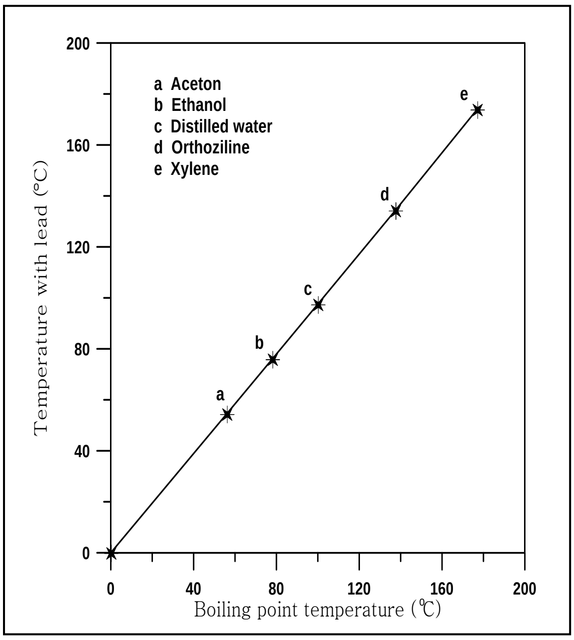


Fig. 11. b. Thermocouple Calibration Results.

3.11. Velocity Measurement

One method to measure and calculate the velocities in this research was the direct measurement by the anemometer, it is used to calculate the streamwise velocity (u_i). The velocity was determined according to [27].

These velocities were used in calculating the Reynolds number and the airflow rate. The mass flow rate for the different flows is calculated from the following:

$$\dot{m} = \int_{x_1}^{x_{Nx}} \int_{y_1}^{y_{Ny}} \rho \cdot u_{in} \cdot dy \cdot dx \qquad \ldots(1)$$

where the u_{in} is the inlet streamwise velocity and it is not constant across the duct height but it has a specific profile.

$$\dot{m} = 1.2 * \int_{x_1}^{x_{Nx}} \int_{y_1}^{y_{Ny}} u_{in} \cdot dy \cdot dx \qquad \ldots(2)$$

So the volumetric flow rate is:

$$\dot{Q} = \dot{m} / \rho \qquad \ldots(3)$$

3.12. Temperature Measurement

Temperature distribution at various stations and points inside the duct at inlet and outlet of water inside heat exchanger . However temperatures distribution before vortex generators and after heat exchanger in the test section, infrared thermometers and type K thermocouple. The temperature measurement devices were calibrated.

3.13. Procedure

1. Connect heat exchanger to hot water supply.
2. Set hot water temperature.
3. 3-Plant two thermocouples in air duct to measure inlet and outlet air temperature.
4. Plant two thermocouples to measure inlet and outlet hot water temperature inside heat exchanger.
5. Set water flow rate as desired.
6. Switch on fan.
7. Switch on circulation pump.
8. 8-Take reading every 50 min after steady state condition.

3.14. Data Analysis

Simplified steps were used to analyze the heat transfer process for the air flow and water in heat exchanger.

The local heat transfer coefficient can be obtained as[28]:-

$$h_x = \frac{q}{(T_s)_x - (T_b)_x} \qquad \ldots(4)$$

$q = \overset{\circ}{m} c_p \Delta T$

q=heat flux

$(T_b)_x$ = Local bulk air temperature.

$(T_s)_x$ = Local tube surface temperature.

All the air properties are evaluated at the mean film air temperature

$$(T_f)_s = \frac{(T_s)_x + (T_b)_s}{2} \qquad \ldots(5)$$

T_f = Local means film air temperature.

The local Nusselt number (Nu_z) then can be determined as:

$$Nu_x = \frac{h_x \times D_h}{k} \qquad \ldots(6)$$

Where:

K=thermal conductivity of air

D_h=hydraulic diameter of duct

$$D_h = \frac{4A}{P}$$

The average values of Nusselt number Nu_m can be calculated as follows:

$$Nu_m = \frac{1}{L}\int_o^L Nu_x dx \qquad \ldots(7)$$

The average values of the other parameters can be calculated based on calculation of average tube surface temperature and average bulk air temperature as follows:

$$\overline{T_s} = \frac{1}{L} \int_{x=0}^{x=L} (T_s)_x \quad dx \qquad \dots(8)$$

$$\overline{T_b} = \frac{1}{L} \int_{x=0}^{x=L} (T_b)_x \quad dx \qquad \dots(9)$$

$$\overline{T_f} = \frac{\overline{T_s} + \overline{T_b}}{2} \qquad \dots(10)$$

$$Re_m = \frac{\rho \quad u_{in} \quad D_h}{\mu} \qquad \dots(11)$$

Where; $u_{in} = \dot{V}/A$, $A = LH$

All the air physical properties ρ, μ and k were evaluated at the average mean film temperature ($\overline{T_f}$).

4. Results and Discussion

4.1. Introduction

The restriction on shape and position of a duct strongly influences fluid flows along the duct.
The main results from the experimental are used in this research, for a fully developed turbulent flow in a rectangular duct for different Reynolds numbers and they are presented graphically.

4.2. Experimental Results

Experimental study was carried out at Reynolds number equal to (62000-125000) for all shapes of vortex generators. The location of the winglets was changed according to longitudinal distance to get the best location of vortex generators at X_m=0.5 cm at circular shape and X_m=2.5 at square shape.

4.3. The effect parameters on the heat transfer

Finding many parameters effect on the heat transfer coefficient, I will recogniz the Nusselt number reference to enhance the heat transfer from the heat exchanger as shown below.

4.3.1. Vortex generator shape effect

Figures (12,13,14) represent the relation between Nusselt number and flow Reynolds number at location (X=0.5, 1.5, 2.5cm) respectively, as shown Nusselt number increases higher by inserting circular and square vortex generators with respect to without turbulator but the increase in Nusselt number is shown to be highest at Reynolds number Also it indicates that the circular vortex generators at location (X=0.5,1.5 cm) increases Nusselt number with respect to the square vortex generators with ratios (28.5%, 13.5%) respectively, because the turbulence flow in the circular vortex generators greater than the square vortex generators is leading to enhancement the heat transfer in the circular shape better than the square shape . The square vortex generators at location (X=2.5 cm) increases Nusselt number with respect to the circular vortex generators with ratio (29.5%).

The circular vortex generators at location (X=0.5, 1.5, 2.5cm) increases Nusselt number with respect to without turbulator with ratios (56%, 50%, 36%) respectively, and the square vortex generators at location (X=0.5, 1.5, 2.5cm) increases Nusselt number with respect to without turbulator with ratios (39%, 42%, 51%) respectively
For both cases the general equation of this relation is given as equations 12 and 13 for Figures 15 and 16 respectively.

$$Nu= [-21757(X/L)^2- 4693.8(X/L) + 35.454]+ [0.855(X/L)^2 + 0.063(X/L) + 0.002]Re + [-9 \times 10^{-6} (X/L)^2 - 6 \times 10^{-21} (X/L) + 1 \times 10^{-8}]*Re^2$$

(circular) …(12)

$$Nu= [27720(X/L)^2 - 4766.7(X/L) - 187.18] + [-0.45(X/L)^2 + 0.066(X/L) + 0.0069] Re + [2 \times 10^{-22} (X/L) - 2\times 10^{-8}] Re^2$$

(square) … (13)

Where:X: location of vortex generator.
L: the distance before heat exchanger.

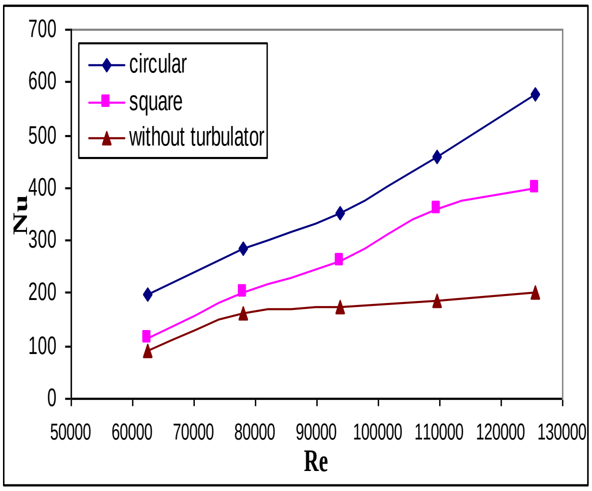


Fig. 12. Experimental Variation of Nusselt Number with Reynolds Number at location (X=0.5 cm) for Vortex Generators.

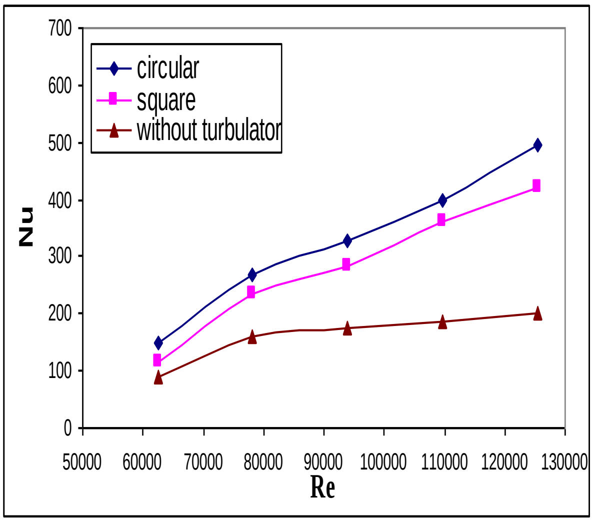


Fig. 13. Experimental Variation of Nusselt Number with Reynolds Number at location (X=1.5 cm) for Vortex Generators.

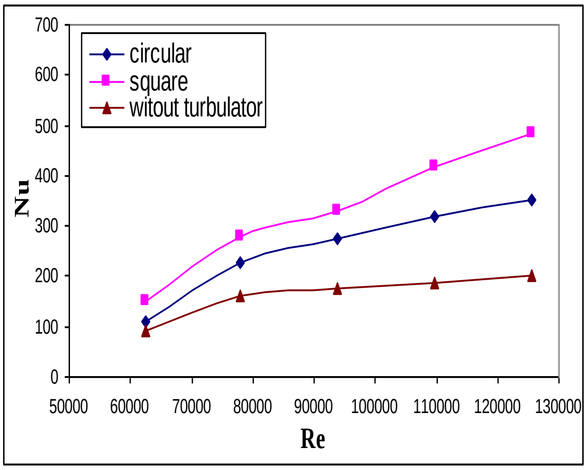


Fig. 14. Experimental Variation of Nusselt Number with Reynolds Number at location (X=2.5 cm) for Vortex Generators.

4.3.2. Location of vortex generator

Figures (15, 16) show the relation between Nusselt numbers (with vortex generators, and without vortex generators) for both cases (circular and square) respectively and flow Reynolds number. As shown in these Figures the behavior is the same as that obtained in Figure (12, 13, and 14). But the heat transfer process in circular shape is better than in square shape at location (X=0.5, 1.5 cm) , the square vortex generators at location (X=2.5 cm) is better than circular shape because the square vortex generators in this location provide turbulence flow better than the circular shape as shown in figures.

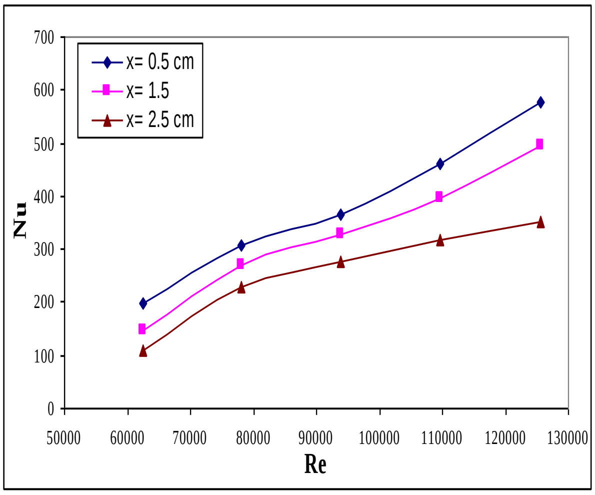


Fig. 15. Experimental Variation of Nusselt Number with Reynolds Number at Circular Vortex Generators.

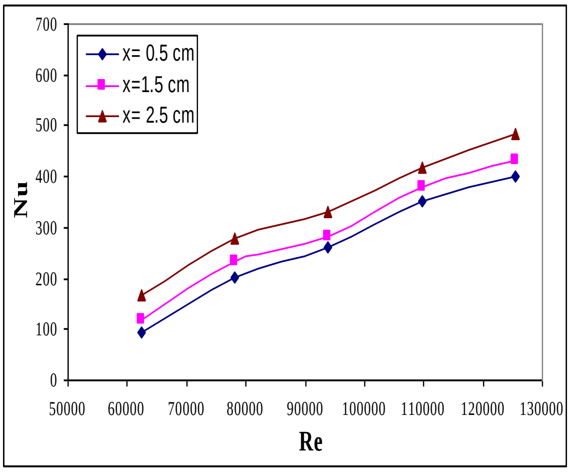


Fig. 16. Experimental Variation of Nusselt Number with Reynolds Number at Square Vortex Generators.

4.3.3. Reynolds Number

Figures (12, 13, 14, 15, and 16) show that Nusselt numbers are enhanced with Reynolds number increasing for all cases due to increase the inlet velocity (u_{in}) and that will increase the heat transfer from the heat exchanger.

5. Conclusion and Recommendations

5.1. Conclusion

Experimental study for heat transfer around heat exchanger using vortex generators (circular and square) in turbulent flow has been done. In the experimental study, an apparatus was set up to measure the velocity and temperatures around the heat exchanger with constant heat flux using two shapes of vortex generators at a fixed point.

1- There is an effect for the shapes of vortex generators on heat transfer,Temperatures and velocity distribution.
2- Circular shape is the best shape for enhancing heat transfer and the square shape gives minimum heat transfer in the present work.
3- Heat transfer is enhanced (36-56) % when Circular shapes of vortex generators are used.
4- Heat transfer is enhanced (39-51) % when Square shapes of vortex generators are used.
5- Heat transfer increases by using Circular shape when the distance of vortex generators is equal to (X_m= 0.5, 1.5 cm) before heat exchanger.
6- Heat transfer increases by using square shape when the distance of vortex generators is equal to (X_m= 2.5 cm) before heat exchanger.

5.2. Recommendation for future work

The following points can be recommended for future work.

1. Numerical investigation can be extended to consider 2-D or 3-D problems.
2. Other shapes of vortex generators can be investigated numerically and experimentally (ellipsoidal, trapezoidal, triangle, rectangular)
3. New fixed location for vortex generators and angle (in front of heat exchanger) can be used.
4. The effect of area can be taken into consideration.
5. Replace air with steam and study its characteristics.
6. Compare the turbulence model used in this research with LES (large eddy simulation) turbulence model.
7. Increasing the number of vortex generators fixed in each study.

Notation

Symbol	Description	Unit
A	Tube surface area	m^2
C_p	Specific heat at constant pressure	J/Kg.ºC
D_h	Hydraulic diameter	m
h	Coefficient of heat transfer	W/m^2.ºC
K	Thermal conductivity	W/m.ºC
L	Length of tube	m
m	Volumetric flow rate	m^3/s
T	Air temperature at any point	ºC
Symbol	**Description**	**Unit**
T_b	Bulk air temperature	ºC
T_f	Mean film air temperature	ºC
T_i	Air temperature at tube entrance	ºC
T_s	Tube surface temperature	ºC
u	Axial velocity component	m/s
u_{in}	Axial velocity at tube entrance	m/s
X_m	Longitude distance	Cm

Dimensionless Groups

Symbol	Description	Equation
Nu	Nusselt number	$\frac{hD_h}{k}$
Pr	Prandtal number	$\frac{\mu C_p}{k}$
Re	Reynolds number	$\frac{u_i D_h}{\nu}$

Subscript

Symbol	Meaning
s	Surface
a	Air
i	Inter
h	Hydraulic diameter
x	Local
f	Film
m	Meter

Greek letters

Symbol	Description	Unit
μ	Dynamic viscosity	Kg/m.s
ν	Kinematics viscosity	m^2/s
ρ	Air density at any point	Kg/m^3

Abbreviation

Symbol	Description
B.L	Boundary layer
V.G	Vortex generator
H.E	Heat exchanger
DX	Horizontal Distance of Winglet
DY	Vertical Distance of Winglet

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دراسة عملية عن تأثير شكل وموقع مولدات الدوامات عند مدخل المبادل الحراري

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الخلاصة

دراسة عملية أجريت لفحص تأثير مولد الدوامات (الشبكة الدائرية، الشبكة المربعة) على الجريان و انتقال الحرارة عند مواقع $(X = 0.5, 1.5, 2.5 \text{ cm})$ قبل مبادل حراري ضمن مدى لرقم رينولد $62000 < Re < 125000$ و فيض حراري $3000 \le q \le 8000 \text{ W/m}^2$.في الدراسة العملية تم نصب و صيانة جهاز مختبري لقياس السرعة و درجة الحرارة حول المبادل الحراري.

أظهرت النتائج أن لوجود مولد الدوامات اثر كبير على تحسين انتقال الحرارة ويعتمد هذا على شكل المولد وموقعهِ. الشبكة الدائرية هي الأفضل لعملية نقل الحرارة عند موقع $[X_m=0.5 \text{ cm}]$ قبل المبادل الحراري. الشبكة المربعة هي الأفضل لعملية نقل الحرارة عند موقع $[X_m=2.5 \text{ cm}]$ قبل المبادل الحراري.

قورنت النتائج العملية في حالة وجود مولد الدوامات مع عدم وجود مولد الدوامات. أظهرت هذه المقارنة تحسن في انتقال الحرارة حول المبادل الحراري بنسب(56%, 50%, 36%) عند مواقع (X=0.5, 1.5, 2.5cm) على التوالي بأسنخدام مولد الدوامات الدائري الشكل نسبةً الى عدم وجود عوائق جريان،وانتقال الحرارة حول المبادل الحراري يتحسن بنسب (39%, 42%, 51%) عند مواقع (X=0.5, 1.5, 2.5cm) على التوالي بأسنخدام مولد الدوامات المربع الشكل نسبةً الى عدم وجود عوائق جريان.