Irreversibility Analysis and Numerical Simulation in a Finned-Tube Heat Exchanger Equipped with Block Shape Vortex Generator

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Abstract: In this paper the effect of block shape Vortex Generators (VGs) on an air-water finned-tube heat exchanger has been studied experimentally using exergy analysis method. Also the effect of these VGs on increasing heat transfer rate has been simulated numerically and the results indicate a good agreement with the experiments. In this research a wind tunnel is used to produce wind flow over heat exchanger in the range of 0.054 *kg/s* to 0.069 *kg/s*. The steady volume flow rate of hot water is 240 *L/h* with the temperature of 44^{o}C to 68^{o}C. These experiments have been carried out with and without VGs on the heat exchanger. Results demonstrate that using the VGs has reduced Air Side Irreversibility to Heat transfer Ratio (ASIHR). To reveal the effect of VGs on heat exchanger performance with respect to reducing ASIHR, a quantity is used namely Performance of Vortex Generator (PVG). The calculated PVG values are less than 15% to over 35% which confirm the satisfactory effects of VGs on heat exchanger performance.

Keywords: Heat Exchanger, Irreversibility, Steady Flow Rate, Vortex generator

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1 INTRODUCTION

Finned-tube heat exchangers have wide application in various fields such as power generation, refrigeration, air conditioning systems, automobiles and chemical industry applications. In this type of heat exchanger, in the air region, the air resistance is an important part of the overall thermal resistance [1]. Therefore, to improve the efficiency of heat exchangers, improving heat transfer from the fin surface is essential. The method of improving heat transfer at the air side of heat exchanger is to apply different types of vortex generator on the fins in the stream direction. Vortex generators are small bumps that form in different ways, such as molding, punching or multiple operations are being installed on the surface of fins in the flow direction. These VGs make longitudinal vortices which are parallel to the main flow direction that lead to rotation of the flow and increase the mixture in the downstream areas. Additionally VGs also determine secondary flow patterns. The improvement of heat transfer is associated with a relatively low pressure drop.

Biswas et al., provided numerical investigations to improve heat transfer in a channel with circular tubes, and Winglet VGs [2]. They showed that in the absence of the winglet type vortex generator, relatively little heat transfer takes place in the downstream of the circular tube which is a recirculation region with low velocity fluid. However, in the presence of a winglet type longitudinal vortex generator in the wake region behind the cylinder, heat transfer in this region can be enhanced as high as 240%, where this may be a great help to reduce the size of the heat exchanger.

Chen *et al.* studied the effects of delta winglet shaped vortex generators that have been punched on a heat exchanger, in two different arrangements: Inline and Staggered arrangement [3], [4]. In in-line arrangement they found that the longitudinal vortices (LVs) of incoming flow intensified the LVs downstream of the second and the third winglet. For Re = 300 and Fi = 500, the ratios of heat transfer enhancement to flow loss penalty were around 1 whiles Winglets in staggered arrangement bring larger heat transfer enhancement than in in-line arrangement since the longitudinal vortices from the former arrangement influence a larger area and intensify the fluid motion normal to the flow direction.

Kotcioglu et al., studied the second law of thermodynamics for cross-flow heat exchangers in the presence of a new block shape convergent – divergent longitudinal vortex generator [5]. They found that increasing cross flow velocity improves heat transfer rate and reduces heat transfer irreversibly and at high Reynolds numbers these VGs improve mixture within the heat exchangers channel. They recorded 15% to 30% increase of heat transfer and 20% to 30% increase of pressure drop in heat exchanger with and without VGs.

Wang et al., presented their results from enlarged fin – and – tube heat exchangers with and without the presence of vortex generators [6]. They examined three samples of fin – and – tube heat exchanger having inline arrangements including one plain fin and two wave– type vortex generators. The presence of vortex generators significantly increase the vertical motions of the horseshoe vortices hitting on the tubes and a much better mixing characteristics is seen by introducing the vortex generators. The penalty of pressure drops of the proposed vortex generators relative to plain fin geometry is relatively insensitive to change of Reynolds number. The frictional effects due to using of VGs increase 25% to 55% compared to flat surface fins.

Tian et al., studied air-side heat transfer in finned tube heat exchanger with wavy fins and delta vortex generators in two styles, Staggered and inline arrangement [7]. They showed that the longitudinal vortices not only reduce the wake size but also increase flow velocity. They stated that heat transfer of heat exchangers were increased 80% for Staggered VGs arrangement and 95% in Consecutive arrangement compared to the heat exchangers without VGs.

Fiebig et al., investigated the effect of wing-type vortex generators on heat transfer and pressure drop of a fin-and-tube heat exchanger [8]. Four fin-and-tube configurations were tested; inline arrangement with plain fins, inline arrangement with fins with a pair of vortex generators behind each tube, staggered arrangement with plain fins, and staggered arrangement with fins with a pair of vortex generators behind each tube. For the inline tube arrangement the vortex generators increase the heat transfer by 55-65% with a corresponding increase of 20-45% in the apparent friction factor. Results indicate that the vortex generators have the potential to reduce considerably the size and mass of heat exchangers for a given heat load.

Valencia et al., investigated heat exchanger element with (1) One tube, and with (2) a Flat tube and a Pair of Winglets [9]. A fin-tube element with a flat tube gives smaller Nu and friction factor than the element with a round tube. Longitudinal vortices generated by a pair of mounted delta winglets can substantially improve the performance of the element with a flat tube. With vortex generators, fiat tubes can give a superior performance to the round tubes with or without vortex generators, since heat transfer in both cases are the same but the pressure loss is much less for flat tubes.

Joardar and Jacobi, calculated experimentally the potential of triangular vortex generator to improve the heat transfer rate at the air side with a fin-tube heat exchanger inside the wind tunnel tests [10]. This heat exchanger has seven rows of consecutive pipes where double VGs with intermittent one-row and three-row style were studied. These fins were installed in the upstream region of flow to improve the heat transfer of the wake. Results showed that three-rowed arrangement has better heat transfer than one-rowed arrangement but has more pressure drop that decrease with Reynolds number increase. They found that the heat transfer coefficient for one-rowed arrangement changes from 16.5% to 44% when increase in pressure drop is less than 12%. Also for the three-rowed arrangement, heat transfer coefficient increase 26% to 68.8% while the pressure drop increases from 26% to 87.5%.

In this study a finned-tube heat exchanger with block shaped vortex generator has been studied numerically and experimentally, using exergy method. VG's were located at wind flow upstream. Thermal and exergy transfer efficiency of heat exchanger in two modes were compared. A mode is without VG's installation and the other mode is with VG's installation. To reveal the effects of this VG on heat exchanger from an irreversibility approach, ASIHR reduction has been expressed by PVG that is the ASIHR reduction percentage in the modified heat exchanger with the installation of VG's compared to initial heat exchanger.

2 EXPERIMENTAL SET UP PREPARATION

Figure 1 shows the schematic of a part of used experiment tool. As it is observed in Fig. 1, it consists of a wind tunnel that is used to generate air flow with different velocities over the heat exchanger fins by a variable speed fan that is located at the end of tunnel. The inlet air channel has rectangular cross section that is 0.0317 m^2. The parameters that should be calculated are air mass flow and pressure drop during air passing over the heat exchanger, so two differential micromanometers have been used. One of them is for mass airflow measurement by using an orifice system and another one to record the air pressure drop. To flow out the hot water a centrifugal pump is used. This pump takes the water from the distribution network and keeps it in a water tank with a floating valve, then sends it to an electric heater to heat the water that is needed. This is done by an electronic thermostat heater.

A rotameter is used to measure the mass flow rate of water with the accuracy of 5 L/h. To regulate and control the water flow and fluctuations, a short connection tap is used beside the pump. To register water pressure drop during passing through the tube, a U-shaped mercury manometer with accuracy of 0.1 cm Hg is used. To determine the rate of heat transfer and irreversibilities, it is necessary to record temperatures for air and water at entry and exit of heat exchanger. So 100pt thermo resistance thermocouples are used. These kinds of thermocouples have a good accuracy in large range of temperatures. Thermocouples used here are linked to a digital temperature recording device with an accuracy of 0.1 °C and temperature changes up to 99.9 °C.

In Fig. 2, heat exchanger geometry is shown. This exchanger consists of six flat rectangular plate fins with 1mm thickness and sixteen copper tubes in two rows with inline arrangement. The thickness of the copper pipes is 1 mm and the outer diameter is 12.8 mm. Three collectors made of copper at the top and bottom of exchanger with outer diameter of 28.7 mm, are the hot water channels. Exchanger length is 185 mm and its width is 170 mm and the distance between any two consecutive fin is 20 mm. Figure 3 shows the vortex generator dimensions used in this study. VGs have been made of MDF wood and will be located on the fins (All dimensions are in millimeters).

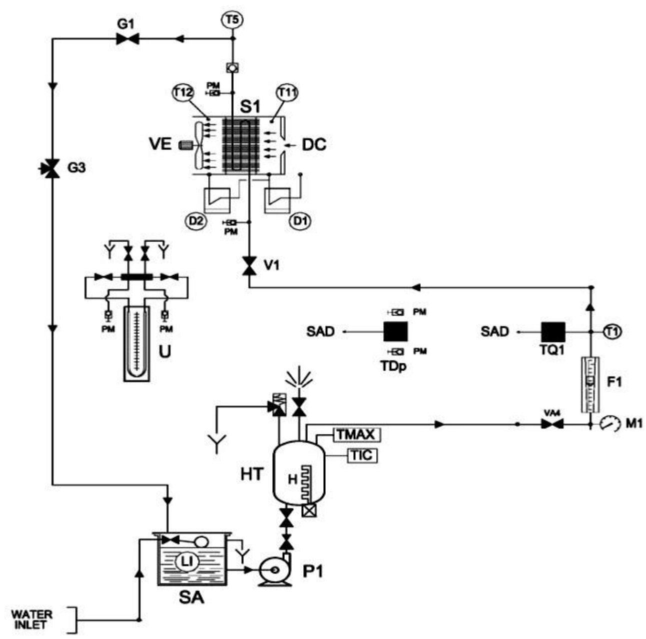

Fig. 1 Experimental setup: (DC) Calibrated diaphragm for the measurement of air flow; (D1) Differential micro manometer for the measurement of air flow from DC; (D2) Differential micro manometer to measure air pressure drop; (F1) Hot water flow meter; (G1) Hot water flow manual control valve; (G3) Three way hot water recirculation manual valve; (H) Hot water resistor type electric heater; (HT) Hot water accumulator; (LI) Float valve; (M1) Hot water pressure gauge; (PM) Pressure measurement tap; (P1) Hot water centrifugal pump; (SA) Water reservoir; (S1) Radiator type water/air heat exchanger; (TIC) Hot water electronic thermostat; (TMAX) Maximum hot water temperature safety thermostat; (T1..T12) pt100 thermo resistance; (U) Mercury differential manometer to measure pressure drops of water flow; (VE) Variable speed fan for S1; (V1) Hot water circuit S1 selection valve.

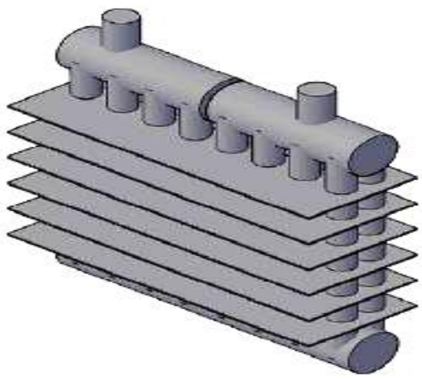

Fig. 2 Overall view of heat exchanger

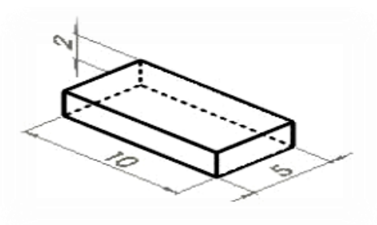

Fig. 3 Geometrical shape and dimensions of VG

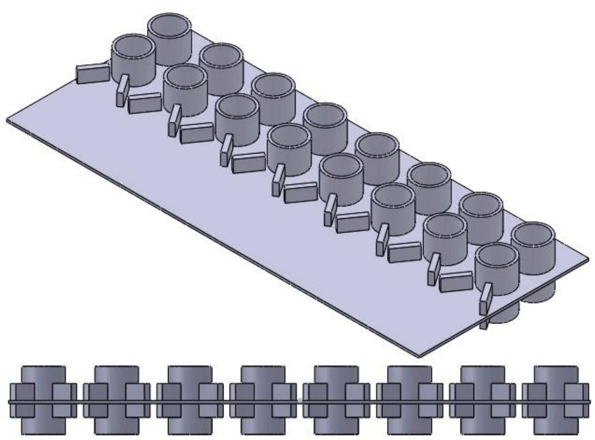

Fig. 4 Cropped view of heat exchanger with VGs.

Figure 4 shows cropped view of a heat exchanger on which vortex generators are mounted. The location and arrangement of VGs on the fins are illustrated. The VGs and horizontal axis have angle of 45 degrees. When the air flows over these VGs, longitudinal vortex is induced in the downstream. These vortices make an appropriate region for heat transfer because of generating turbulence in stream and consequently heat convection coefficient will be increased. The flow around VGs edges makes secondary vortexes that have a circulation in the opposite direction of the initial vortices and the results are small horseshoe-shape vortices which move toward the place that the VGs are linked to the wall.

In all tests the ambient air were passed over the heat exchanger fins with two different mass flow rate 0.054 kg/s and 0.069 kg/s using a fan. While hot water runs inside the pipes of heat exchanger with constant flow rate 240 L/h, while the Inlet temperature of the hot water varies from 44°C to 68°C passing through the heat exchanger and the data are recorded subsequently. In addition to measuring input and output air and water temperature, pressure drop in both fluid and ambient temperature and pressure are also recorded.

3 DATA ANALYSIS

For calculation of heat transfer rate in both of air and water sides Eqs. (1) and (2) are used respectively [1].

$$q_{air} = \dot{m}_{air} c_p \left(T_{ao} - T_{ai}\right) \tag{1}$$

$$q_{water} = \dot{m}_{water} c \left(T_{wi} - T_{wo}\right) \tag{2}$$

To calculate the heat transfer rate, Eq. (1) has been used that is related to the heat transfer rate of air. The air temperature change is much more than the water flow temperature changes in a heat exchanger because the heat capacity of air is much smaller than the water heat capacity. Therefore, the error in reading digital temperature indicator on the air side of the heat transfer has less effect on calculated heat transfer rate. Efficiency of the heat exchanger is obtained using Eq. (3) [1]:

$$\varepsilon = \frac{q}{q_{max}} \tag{3}$$

In which the heat transfer rate in the numerator is the heat transfer on the air side and, q_{max} is the highest heat transfer rate that may occur. In a case that inlet and outlet fluids have the same temperature, considering the $C_{water} > C_{air}$, Eq. (4) is obtained [1]:

$$q_{max} = \dot{m}_{air} c_p \left(T_{wi} - T_{ai}\right) \tag{4}$$

Assuming a steady state for fluid flow and heat transfer rate and constant physical properties of the fluids, and considering that heat transfer to the surrounding is negligible, flow specific exergy is calculated as follows [1]:

$$e = \left(h - h_\infty\right) - T_\infty \left(s - s_\infty\right) \tag{5}$$

In which, "h" is enthalpy, "s" is entropy of fluid and index "∞" refers to the surrounding conditions. According to Eq. (5), using thermodynamic relations and inlet and outlet temperatures recorded for the air and water, the exergy change rates of air and water can be written respectively as follows [11]:

$$\Delta\dot{E}_{air} = \dot{m}_{air}[c_p\left(T_{ao} - T_{ai} - T_\infty ln\frac{T_{ao}}{T_{ai}}\right) - T_\infty \tilde{R}\frac{\Delta P_a}{p_{ai}}] \quad (6)$$

$$\Delta\dot{E}_{water} = \dot{m}_{water}[c\left(T_{wo} - T_{wi} - T_\infty ln\frac{T_{wo}}{T_{wi}}\right) - \vartheta\Delta P_w] \quad (7)$$

In these equations, $\hat{R}$, θ, ΔP are respectively the gas constant, specific volume of water and the flow of water or air pressure drop (that is positive). For heat exchangers that are used in higher temperature than ambient temperature if the pressure drop is not zero, exergy transfer effectiveness is as follows [11]:

$$\varepsilon_{exergy} = \frac{T_{ao} - T_{ai} - T_\infty ln\frac{T_{ao}}{T_{ai}} - \frac{I\,\Delta p}{c_p}}{T_{wi} - T_{ai} - T_\infty ln\frac{T_{wi}}{T_{ai}}} \quad (8)$$

In this equation, for ideal gas $I = T_\infty \tilde{R} / p_i$, and for an incompressible fluid I=θ. The irreversibility rate is obtained as below [11]:

$$\dot{I} = -\Delta\dot{E}_{water} - \Delta\dot{E}_{air} \quad (9)$$

Total irreversibility rate of the air side can be calculated as follows [11]:

$$\dot{I}_{AS} = q\left(1 - T_\infty / T_J\right) - \Delta\dot{E}_{air} \quad (10)$$

To determine total irreversibility rate of the water side the following equation can be used [11]:

$$\dot{I}_{ws} = q\left(1 - T_\infty / T_J\right) - \Delta\dot{E}_{water} \quad (11)$$

Which $T_J = (T_{wi} + T_{wo})/2$, and q is the heat transfer rate that is a positive number. ASIHR dimensionless number that is the ratio of the rate of irreversibility on the air side to the heat transfer within the heat exchanger is defined by the following equation [11]:

$$ASIHR = \dot{I}_{AS} / q \quad (12)$$

Dimensionless number PVG that is used to evaluate vortex generators and is defined as modified heat exchanger ASIHR reduction by VGs compare to the initial heat exchanger will be calculated as the following equation [11]:

$$PVG = \frac{100\left(ASIHR_{without\,VG} - ASIHR_{with\,VG}\right)}{ASIHR_{without\,VG}} \quad (13)$$

And the Reynolds number of the flow is determined from the following equation [11]:

$$Re_{AS} = \frac{\dot{m}_{air} d_{tube}}{A_{duct}\,\mu_{air}} \quad (14)$$

In which μ_{air} is achieved in the average air flow temperature.

4 RESULTS AND DISCUSSIONS

Variations of heat transfer rate versus mean temperature difference have been represented in Fig. 5 for various air mass flows. It can be observed that heat transfer rate increases with inlet water temperature because of increasing mean temperature difference between water and air. Also an increment in air mass flows has a similar effect on heat transfer rate by increasing convection heat transfer coefficient which is a consequence of increasing Reynolds number and subsequently Nusselt number on air side.

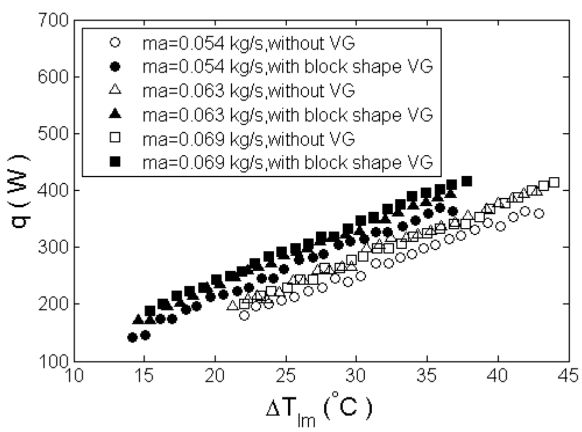

Fig. 5 Variation of heat transfer rate vs. mean temperature difference in heat exchanger.

It can be seen that using VGs increases heat transfer from 14% to 30% for all mass flows. This can be explained by increasing heat transfer from fins and tubes surfaces of heat exchanger due to creation of longitudinal vortexes which is caused by sticking VGs to the fin surfaces at upstream. These vortices create a secondary flow which prevents from boundary layer growth, so heat transfer between fin and fluid flow improves. Also they postpone the boundary layer separation behind the tubes. With extending of boundary layer and elimination of low heat transfer regions in wake region, heat transfer from fins will be improved.

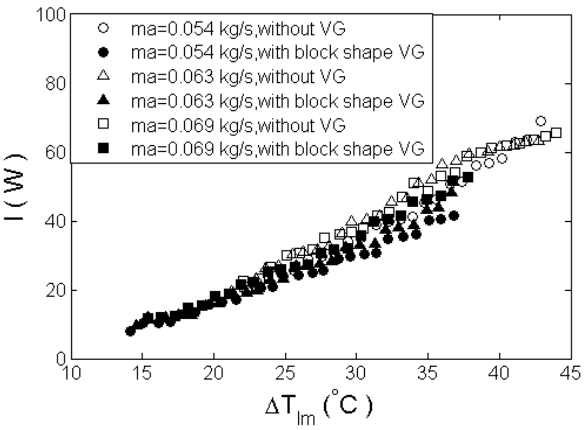

Fig. 6 Variation of total irreversibility rate of heat exchanger against mean temperature difference.

Figure 6 shows the irreversibility rate of heat exchanger against mean temperature difference. It is understood from the figure that increasing air mass flow causes an increment in total irreversibility in heat exchanger because of enhancing mean temperature difference between air and water. Total irreversibility rate has been reduced using VGs which improves heat transfer and reduces mean temperature difference between air and water.

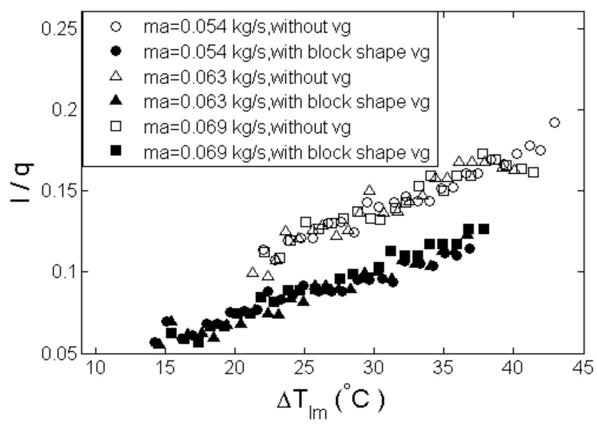

Fig. 7 Variation of dimensionless irreversibility of heat exchanger vs. mean temperature difference.

Figure 7 is used to investigate simultaneous effects of VGs on heat transfer rate and total irreversibility of heat exchanger which has become dimensionless with heat transfer and plotted against mean temperature difference. Using VGs causes a reduction in irreversibility per heat transfer rate unit of heat exchanger as it is observed.

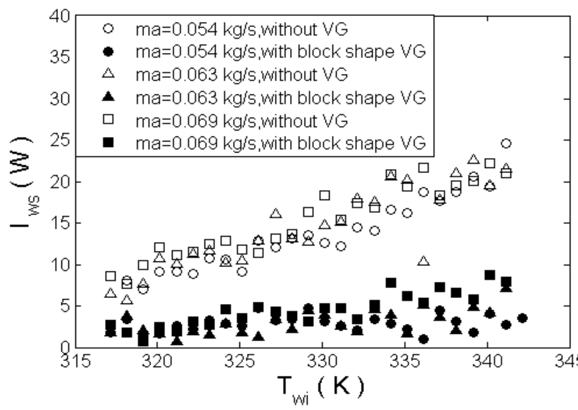

Fig. 8 Variation of irreversibility on water side vs. inlet water temperature.

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Variation of irreversibility on water side against inlet water temperature has been represented in Fig. 8. In comparison with heat exchanger without VGs, heat exchanger equipped by VGs has less irreversibility on water side. Increasing heat transfer rate and reduction of exergy variation due to decrement of finite temperature difference on water side, may be the two reasons for decreasing of irreversibility.

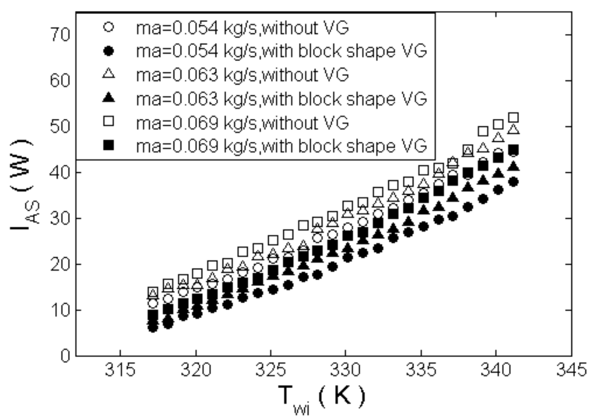

Fig. 9 Variation of irreversibility on air side vs. inlet water temperature.

Figure 9 shows variation of irreversibility on air side versus inlet water temperature. As it is observed, increasing mass flow and temperature difference between air and water cause an increment in irreversibility rate. Using VGs irreversibility on air side decreases due to reduction of finite temperature difference between air and water. It seems that this factor has a greater effect on irreversibility rather than increasing heat transfer rate.

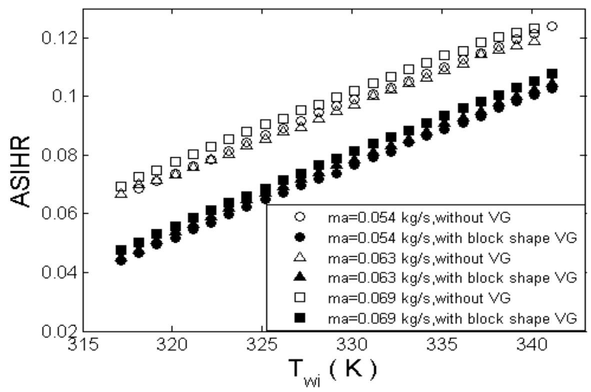

Fig. 10 Variation of dimensionless irreversibility on air side vs. inlet water temperature.

Variation of dimensionless irreversibility on air side against inlet water temperature has been shown in Fig. 10 for different conditions. It can be understood from this figure that although heat transfer rate increases by increasing inlet water temperature, increment of irreversibility rate on air side is more than heat transfer rate enhancing. As it is observed, using VGs dimensionless irreversibility on air side has been reduced. Using VGs not only improves heat transfer rate but also reduces temperature difference of air flow which has caused a decrement in irreversibility rate on air side.

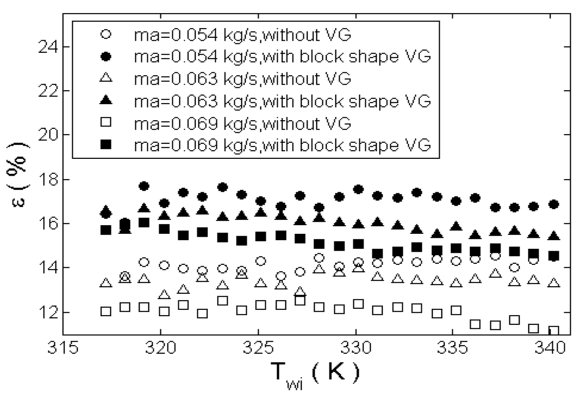

Fig. 11 Variation of heat exchanger effectiveness against inlet hot water temperature.

Variation of heat exchanger effectiveness against inlet hot water temperature has been displayed in Fig. 11. As it is expected, using VGs and increasing heat transfer rate to the air side of heat exchanger consequently, heat exchanger effectiveness has been enhanced greatly. Also decreasing air mass flow causes an increment in heat exchanger effectiveness because of providing more time for the air to get closer to the inlet hot water temperature.

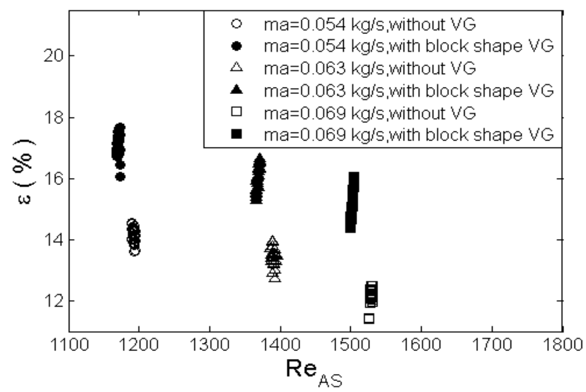

Fig. 12 Variation of heat exchanger effectiveness versus Reynolds number of inlet air flow.

Figure 12 represents variation of heat exchanger effectiveness versus Reynolds number of inlet air flow in various statuses. This figure can be a confirmation for Fig. 18 and shows increasing of heat exchanger effectiveness due to the use of block shape VGs for a certain Reynolds number.

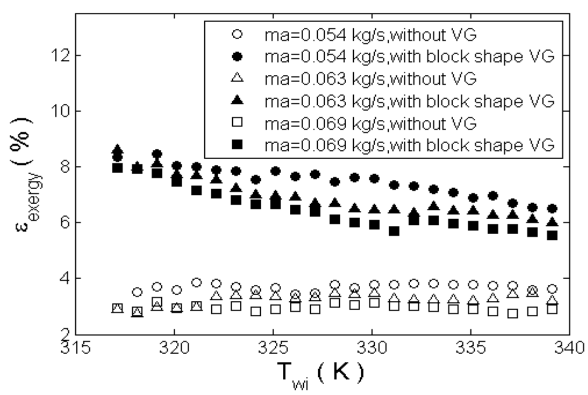

Fig. 13 Variation of exergy transfer effectiveness vs. water inlet temperature.

Figure 13 shows the exergy transfer effectiveness changes of heat exchanger with water inlet temperature. Using VGs enhances exergy transfer efficiency because the rate of exergy transfer to air increases at a specific inlet water temperature that is because of irreversibility rate reduction within heat- exchanger. Also this figure shows that Lower mass air flow has more exergy transfer effectiveness because inlet air has more time to obtain the highest possible exergy in which the outlet air temperature is equal to the inlet water temperature.

To reveal the effects of block shape VGs on heat exchanger based on air-side irreversibility, a quantity named PVG is used. PVG is the ASIHR reduction percentage of heat exchanger that was improved by block shape VGs divided by initial heat exchanger without using VGs.

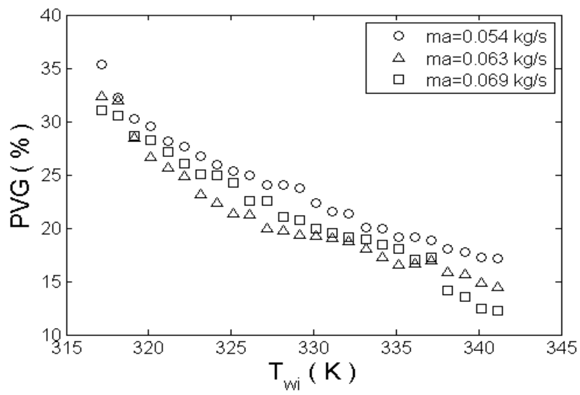

Fig. 14 Variation of PVG versus water inlet temperature.

Figure 14 shows the PVG changes vs. water inlet temperature for different amounts of air mass flow. The results show that for different mass flow rates, PVG changes from 15% to 35%. It means that for modified heat exchanger, ASIHR reduction is more than 15% of the initial heat exchanger. This parameter shows the advantage of using VGs by means of 2nd law of thermodynamics analysis. Also further number of PVG leads to better performance conditions for heat exchangers based on exergy analysis.

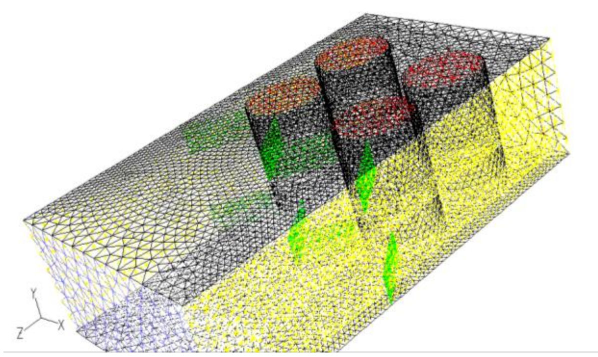

Fig. 15 Heat exchanger configuration mesh.

5 NUMERICAL MODELING OF HEAT EXCHANGER

In simulating the heat exchanger, first a three dimensional configuration of heat exchanger has been provided using GAMBIT software. Then using FLUENT software a numerical analysis has been done with a suitable meshing. Figure 15 represents this heat exchanger mesh. Wall boundary condition is used to introduce VGs and because of problem symmetry, symmetry boundary condition is used in side walls. Mass Flow Inlet and Pressure Outlet are used to satisfy inlet and outlet conditions of heat exchanger respectively. Heat exchanger has been analyzed in both with and without rectangular block shape VGs.

The RNG $k-\varepsilon$ model is used to simulate turbulence in the flow field. This is a widely used model that provides reasonable accuracy and a robust ability to represent a wide range of flow regimes [12]. The features of the airfoil solver are summarized below:

- Finite volume method with a segregated solver;
- RNG $k-\varepsilon$ turbulence model;
- Standard wall functions for near-wall treatment;
- Pressure-velocity coupling – SIMPLE;
- Turbulence kinetic energy – second order upwind;
- Turbulence dissipation rate - second order upwind;

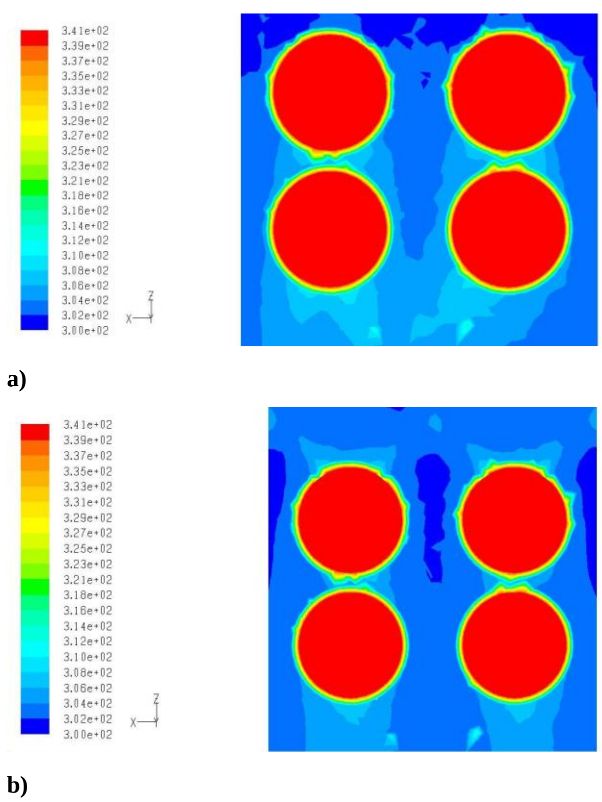

Fig. 16 a) temperature contour without VGs, b) temperature contour with VGs.

Figure 16 shows the temperature contour in two different conditions of heat exchanger fins. It is obvious that using VGs has created a larger temperature gradient between fin surfaces and water tubes and around them. This object declares the great effect of passing air on heat exchanger cooling and increasing heat transfer to the air flow.

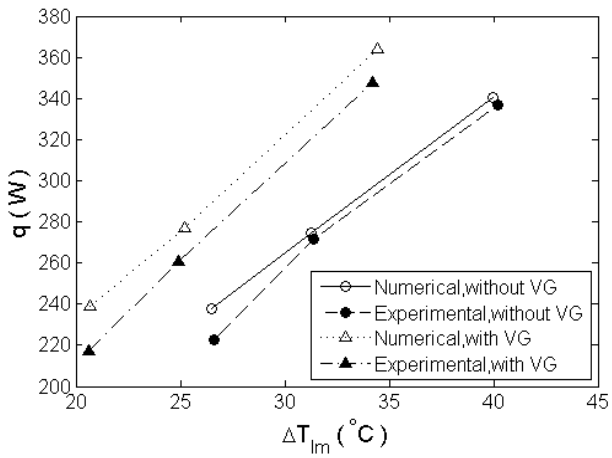

Fig. 17 Comparison of heat transfer rate between simulation and experimental results.

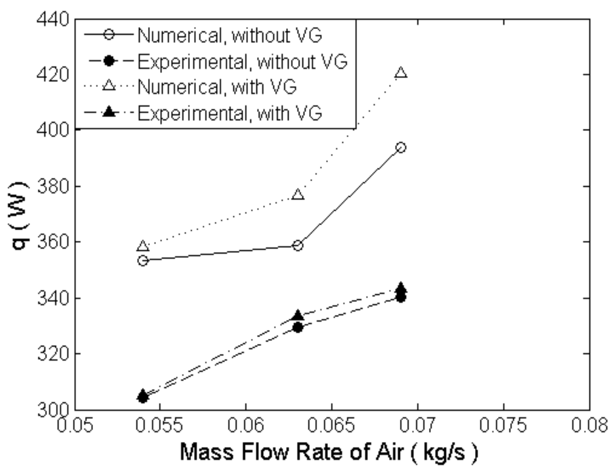

Fig. 18 Comparison of heat transfer rate between simulation and experimental results for different air mass flows.

At last, simulation results have been compared with experimental data. Figure 17 shows heat transfer rate for 0.054 kg/s of air mass flow in two cases, with and without VGs at three logarithmic average temperatures. Heat transfer rate for three specific air mass flows used in experiment and inlet water temperature of 60^{o}C has been represented in figure 18. The curves trend shows that these two figures confirm the results.

6 CONCLUSION

Using VGs improved heat transfer rate from fins and tubes surfaces by inducing longitudinal vortices and preventing from boundary layer growth. This increment in heat transfer rate enhances the heat transfer effectiveness. Also exergy transfer effectiveness of heat exchanger increases because of increasing exergy transfer rate to the air side. PVG values show that how VGs can improve the heat exchanger performance. In other words PVG is a quantity to evaluate the effect of a specific VG on the air side of a heat exchanger. In this paper PVG values change in the range of 15% to 35% for rectangular block shape VGs. This means that the employed optimization procedure may decrease the irreversibility of the air side per unit of heat transfer.

7 NOMENCLATURE

$\dot{m}$	mass flow rate
h	enthalpy
ASIHR	Irreversibility to Heat transfer Ratio of Air Side
c	Specific heat of water (J/kg K)
T	temperature
c_p	Specific heat of air at constant pressure (J/kg K)
$\dot{E}$	Exergy transfer rate (W)
PVG	Performance of Vortex Generator
R	gas constant for air
$\dot{I}$	Irreversibility rate (W)
q	Heat transfer rate (W)
s	Entropy (J/kg K)
VG	Vortex Generator
$\Delta\dot{E}$	Change of exergy transfer rate (W)
ΔP	Pressure drop (Pa)
ΔT	Temperature difference (K)
e	Specific flow exergy (J/kg)

Greek symbols:

ε_{exergy}	Exergy transfer effectiveness (%)
ε	Heat exchanger effectiveness (%)
v	Specific volume of water (m^3/kg)

Subscripts:

a	air
AS	Air Side
i	inlet
max	maximum possible
o	outlet
w	water
with VG	Heat exchanger with mounted

	VGs on its fin surfaces
without VG	Flat finned tube heat exchanger
∞	The surrounding state

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