Wasit Journal of Engineering Sciences
VOLUME: (6), NO. : (3) 2018

Numerical and Experimental Investigation of Heat Transfer Features in a Square Duct with Internal Ribs

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Submitted: 16/10/2017
Accepted: 6/12/2017

Abstract: In this work, experimental investigation and numerical prediction were completed to study the heat transfer for disturbed flow in square passage heated from upper and lower aspect through the use of technology to heighten the heat transfer action by the use of different ribs shapes (semicircle, triangular and square) with the same surface area and height, distributed on the superior and inferior aspect of the heated channel. The experimental part included designing and manufacturing test apparatus and run the tests through operational parameters (Reynolds number 10460-40500) and heat flux (138-6888) W/m². While the design parameters included different ribs shapes (circular, triangular and square). Numerical investigation was done by using ANSYS FLUENT14.0 program, depending on Navier–Stokes equations, application of turbulence model k-ε and energy equation for the air were implemented for the same experimental conditions for all ribs type. In conclusion, the additions of ribs make noticeable enhance of the heat transfer compared to without ribs. Square ribs provide the best enhancement in heat transfer by (28.22-58.23) % compared to without ribs. Then comes the triangular and circular ribs that provide enhancement by (20.14-45.62) (10.87-28.13) % respectively. In addition to that, arrange the ribs in staggered pattern on the upper and lower aspect give the best heat transfer.

Key words: ribs, numerical simulation, heat transfer enhancement, square channel.

تحقيق نظري وعملي لخصائص انتقال الحرارة في مجرى مربع مع زعانف داخلية

علي شكر غلام

الخلاصة: في هذا العمل تم اجراء التحقيقات العملية والعددية لخصائص انتقال الحرارة لجريان مضطرب في مجرى مربع المقطع مسخن الجانبين العلوي والسفلي باستخدام تقنية لتحسين الاداء الحراري المتمثلة أضلاع مختلفة الاشكال (دائري، مثلث، مربع) بنفس المساحة السطحية والارتفاع موزعة على السطح العلوي والسفلي المسخن للمجرى. الجانب العملي تضمن تصميم وتصنيع جهاز اختبار ويتم عمل الاختبارات ضمن المتغيرات التشغيلية (رقم رينولد من 10460 الى 40500 ، الفيض الحراري (138- 6888)W/m² اما المتغيرات التصميمية تتضمن أضلاع مختلفة الاشكال (نصف دائري، مثلث، مربع) تم إجراء المحاكاة العددية باستخدام برنامج ANSYS FLUENT14.0 ,بالاعتماد على معادلات نفير-ستوك تم تطبيق نموذج الاضطراب k-ε ومعادلات الطاقة طبقت للحمل القسري للهواء لنفس الظروف التجريبية لجميع انواع الاضلاع. من خلال النتائج نلاحظ ان ادراج الاضلاع يؤدي الى زيادة ملحوظة في معامل انتقال الحرارة مقارنة مع المجرى الفارغ. الاضلاع المربعة تقدم افضل تحسين في معدل انتقال الحرارة بنسبة زيادة % (28.22-58.23) مقارنة مع المجرى الفارغ. يليها الاضلاع المثلثة والدائرية بنسبة تحسين لانتقال الحرارة بمعدل (10.87-28.13) % (20.14-45.62) على التوالي. كذلك لوحظ ان ترتيب الاضلاع بشكل متعاقب على السطح العلوي والسفلي يعطي افضل انتقال للحرارة.

الكلمات الرئيسية: الاضلاع(الزوائد)،محاكاة نظرية،تحسين انتقال الحرارة ، مجرى مربع .

1. INTRODUCTION

Heat transfer enhancement is basically any method intended to increase the performance of a heat system or rise in heat transfer coefficient by using different techniques. Heat exchangers, fins, ribs, or baffle tabulators in the channel are frequently used to raise the rate of convective heat transfer which lead to the compact heat exchanger and intensify the efficiency. Due to their great thermal loads, rib tabulators were used in great -performance thermal systems for decades. Several ribs with in the channels will increase turbulence intensity of cooling or heating levels in comparison with the plane wall channel. The chief explanations aimed at heat transfer improvement in such channels are due to the interruption of thermal and hydrodynamic boundary layers as down through each rib the flow detached, re-circulates, and encroaches on the channel walls. Not only improvement in heat transfer by the use of ribs will occur, also the pressure loss significantly. Many parameters affect the thermal performance and the heat transfer rate, particularly the ratio of rib to channel great, rib geometry, and the ratio of rib pitch to great. Numerous researchers have investigated the heat transfer features in straight channels with different ribs shape. The research in[1] studied computationally, the Nusselt number and the local heat transfer in divergent/convergent four-sided square of the developed turbulent flow have been studied. In this study, only the convergent four-sided square with ribs been taken for conducting the computational analysis of this study. A good agreement has been found when comparing the performance of the convergent ducts with that of the experimental data reported in the heat transfer. Characteristics of local heat transfer in the convergent duct are unlike those of straight duct due to the acceleration of stream wise flow. The research in [2] This experimental investigation illustrated significant effects of turbulent characteristic and pressure drop at different configurations and various arrangements of ribs. A good agreement showed when comparing the experimental results with numerical analysis. The authors in [3] numerically and experimentally investigated the heat transfer characteristic in circular pipes with ribs compared to without ribs (plain case). In conclusion, adding the square ribs in the pipes offer the best performance factors for turbulent flow. The authors in [4] this numerical study investigated the effect of using triangular ribs at different angles on the heat transfer to fluid flow. It can be seen that at constant temperature, as the Reynolds number (Re) rise, the Nusselt number (Nu) will increase too for every cases. in[5] A numerical study investigated the effect of using different shapes of ribs on one wall in four-sided rectangular on the heat transfer while using different fluids. This study showed that injection of mist in small amount in ribbed channels will lead to enhancement of heat transfer. in [6] used ANSYS FLUENT v12.1.software to investigate the influence of using ribs on a four-sided rectangular on fully established turbulent flow. The study showed that, higher enhancement ion square ribs when compared with circular rib. The authors in [7] a numerical investigation to study the influence of using different shapes of ribs and perforations on the characteristic of heat transfer. It is concluded, that the ribs perforations have great effect on the heat transfer. in [8] an experimental investigation to study the thermal characteristics of square channel with angled ribs. It is found that higher heat transfers will occur on angled ribs and V" ribs in comparison with continuous ribs. All the above researchers discussed the effect of using different ribs shapes on the properties of heat transfer without focusing on the equal surface area and height of the object to be studied. The current research provides a theoretical and practical study to test the effect of using different ribs (semicircle, triangular and square) with the same surface area and elevation within a square duct on the properties of heat transfer, as well as studying the effect of distribution and number within the duct of the optimum type.

2. NUMERICAL SIMULATION PROCEDURE

2.1 GEOMETRICAL MODEL

Square channel roughed with ribs on three dimension on two opposing wall were formed. Physical representation of the system as shown in figure (1). Gambit modeling software was used for meshed and work software. Figure (2) shows the tetrahedron mesh that was used in the model. Various grid sizes were tested to choose the mesh optimum grid size and now transported to software program (FLUENT 14.0). The effects of the tested grids temperature distribution were compared to select the optimal grid size in this study. This research includes different types of rib (semicircle, triangular, and square) with the same arrangements of surface area used for each setup.

2.2 MATERIAL USED

Galvanized steel sheet was selected for square channel and incompressible air is the fluid. The material that's use for the channel is the same as for rib.

2.3 BOUNDARY CONDITIONS

In this investigation,(1, 1.7, 2.4, 3.1 and 3.8) m/s are the variation on inlet air velocity respectively as the Reynolds number varies as 10460, 17950, 25400, 33000 and 40500. Room temperature which is 35 °C presumed to be the fluid inlet temperature. Fluent has boundary condition with extended range that allows the flow to exit and enter in the physical model. Due to the density being constant, inlet boundary condition is not essential to be in usage in incompressible flows. Accordingly, in the present study, the velocity inlet is the physical model inlet and the out flow is the outlet. Inlet boundary situations have static the mass flow. Bouncing the solid regions and the fluids is done by the wall boundary conditions. In this study, the attachments of perforated ribs are on two opposite wall where the heat flux is applied on the square channel, while the other two sides are insulated. As shown in table (1).

NAME	TYPE
Air inlet	Velocity inlet
Top wall	Heat flux
Air outlet	Out flow
Bottom wall	Heat flux
Right wall	insulated
Left wall	insulated
Ribs	wall

Table (1) Boundary conditions.

2.4 GOVERNING EQUATIONS

Fluid flow differential governing equation is given by mass conservation equation or continuity, energy equation and Navier Stokes equations or momentum conservation equation, as follows:

1) Continuity Equation

$$\nabla(\rho U) = 0 \quad \text{...... (1)}$$

2) Energy Equation

$$\vec{\nabla}(\rho h U) = -\vec{\rho}\nabla U + \nabla(k\nabla T) + \emptyset + S_h \quad \text{...... (2)}$$

3) Momentum Equations

$$\vec{\nabla}(\rho U u) = -\frac{\partial P}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} + \beta_x \quad \text{...... (3)}$$

$$\vec{\nabla}(\rho U v) = -\frac{\partial P}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} + \beta_y \quad \text{...... (4)}$$

$$\vec{\nabla}(\rho U w) = -\frac{\partial P}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} + \beta_z \quad \text{...... (5)}$$

3. EXPERIMENTAL SET UP

Test rig presented in figure (3) as a schematic and in figure (4) as a photographic image. Detailed description of the experimented test rig that's used to study the heat transfer enhancement in square duct included in this section. In addition to that, tested the effect of adding different ribs shapes (semicircle, triangular, and square) with the same surface area (0.0378 m^2) attached periodically aligned on two opposite side of wall of the duct. The experimental device used in this study directed with an air flow open-loop circuit, comprises the next parts: unit of air supply, system of control valves, ribs, test section, measurement equipment and power supply unit.

- *Unit of Air Supply*

Air to system is provided by blower of centrifugal fan, which works by an electricity with power of (0.5) kW single-phase motor. The flow rate is ranged from (6-45) m^3/hr with velocity varies from (1 to 6.3) m/sec. in order to determine the influence of mass flow rate on heat transfer rate.

- System of Control Valves

To modify the preferred air flow rate in the test rig, manually controlled regulators systems are used.

- Test Section

A test unit made from galvanized steel sheet horizontal square duct with (2) mm thickness, (0.18) m height, and (1) m long. A test section includes, two heating walls were heated by electrical heater (1 kW) connected on the upper and lower aspect part of test section to deliver even heat flux. To prevent electrical short, electrical insulation (Mica) was secured between heater and wall surface. In addition to that thermal paste with decent thermal conductance was used between the wall surface and electrical insulation to guarantee decent thermal connection and heat dissipation which owing to the reduction of contacts resistance. Fiberglass with thickness (2) cm and thermal conductivity (0.04) W/m. K well insulated the outer test section surface which reduces heat loss by conduction. For the intention of calculation the temperature spreading along the lower, and upper sides wall of test section, (6) type K thermocouples secured in the thickness of the two walls of test section. (2) thermocouples were fixed on down-stream and upper-stream of the test unit intended for calculation the outlet and inlet air temperature. In the main center line of the test section (5) thermocouples were fitted. All the voltages of thermocouples output were fed into data logger.

- *Ribs Shape*

Three kinds of ribs shapes (semicircle, triangular and square) with the same surface area (0.0378 m^2) attached periodically inline on two opposite side of duct wall.

- *Power Supply Unit*

Control voltage regulator was used for the purpose of having power with constant voltage value supply to the electrical heater a (variac, staco variable transformers). The current range (0-10) A and the voltage in value of (0-240) V , using a probes to attach the power supply unit to heater wire for decent conductance .

- *Measurement Instruments*

(Data logger type (BTM-4208SD), hot wire anemometer type (YK-2005AH), multi meter type MT 87, thermocouples of type k, and manometer) measurement instruments that were used in the current study.

3.1 CALCULATIONS PROCEDURE

To achieve the calculations procedure, the following steps have been applied:

A) Air Properties

For internal flow, the bulk mean temperatures for the air properties are taken, as follows:

$$T_{bm} = \frac{T_{inlet} + T_{outlet}}{2} \quad \text{...... (6)}$$

B) Hydraulic Diameter

$$D_h = \frac{4\,A}{P} \qquad \text{...... (7)}$$

C) Reynolds Number

Reynolds number is stated for internal flow and defined as the ratio of fluid inertia powers to viscous forces, which is a dimensionless amount:

$$Re = \frac{\rho\,U\,D_h}{\mu} \qquad \text{...... (8)}$$

D) The Temperature of Average Walls Surface

Calculation the temperature of average walls surface, the next equation can be used:

$$\bar{T}_s = \sum \frac{T_s}{6} \qquad \text{...... (9)}$$

E) The Coefficient of Heat Transfer(h)

Calculation the coefficient of heat transfer, the next equation can be used:

$$h = \frac{q_{conv.}}{\bar{T}_s - T_{bm}} \qquad \text{...... (10)}$$

F) Nusselt Number(Nu)

The Nusselt is defined as the relationship of the fluid conductive to the convective thermal resistance. The next equation can be used for calculation:

$$Nu = \frac{h\,D_h}{K_a} \qquad \text{...... (11)}$$

G) Friction Factor

The friction factor can be calculated from the following equation:

$$f = \frac{2\,\Delta P\,D_h}{L\,\rho\,U^2} \qquad \text{...... (12)}$$

3.2 EXPERIMENTAL PROCEDURE

(80) Tests of different ribs shapes are experimented at different air flow rate, different heat flux (heat flux range is 138-6888 W/m^2) and Reynolds number range is (10460- 40500). All the parameters were recorded after reaching a steady state condition and they included the current, the voltage, temperature of insulation, air velocity, air inlet and outlet, ambient, and the temperature of test unit surface walls.

- The ribs put on heater wall sides inside the square duct.
- The blower was switched on, and air velocity was adjusted by manually controlled valves.
- The electrical heater was switched on, and the voltage is control by using the regulator.
- After (35 minutes) the steady state condition was accomplished. The current, voltage, and temperatures were recorded.
- Repeat steps above with various air velocities Reynolds number (10460- 40500) and various heat fluxes heat flux (138 - 6888 W/ m^2).
- All steps above were repeated for various ribs types.

4. RESULTE AND DISCUSSIONS

4.1 Experimental and Numerical Models Rapprochement

To reach reasonable results, it is highly important to validate the numerically formed model. After, creating the numerical model, the results were compared with experimental data. Comparison between experimental and numerical axial temperature is done for square duct without ribs under similar operating conditions (Reynolds number 40500, heat flux 138-6888 W/m^2) as shown in figures (5). It is worth noting that a good agreement between the axial temperature and numerical results. The discrepancies for temperature values are not exceed 11%

4.2 Numerical Results Analysis

The results of the numerical part of the present work were obtained by the use of ANSYS FLUENT 14.0 program. The results were appeared as temperature contours and velocity vectors. Before starting discussing the result of the work, it's very important to make it clear that all tests were done at Reynolds number 40500, heat flux 6888 W/m^2 to study the improvement of heat transfer through square channels with and without the addition of different ribs shapes that has the same surface area.

The numerical tests were done in two stages

- *The first stages*

Study the effect of adding different shapes of ribs with the same surface area on the heat transfer through square channel, fixed number of ribs were tested (7) distributed on the upper and lower part of the channel in a staggered pattern.

- *The second stages*

Study the influence of ribs numbers and their arrangements for the best chosen ribs depending on the results of the first stage and after determining the best type of ribs on heat transfer enhancement through the square channel.

Figure (6) demonstrate temperature contours in transverse planes alongside the axial center plane (A-B) at (x=0.09m) of the duct with and without ribbed. Starting with figure (6a) (duct without ribs) indicated that the higher temperature happened at wall duct however the lower happened at central region. As of Figure (6b) (duct with semicircle ribs) It is observable that with uniform distribution, the lowest change temperature value. With triangular ribs, as illustrated in figure (6c), there is a key change in temperature field in the channel. Figure (6d) (duct with square ribs) demonstrated that the maximum temperature happened at square ribs and core region which will provide good mixing and will improve the overall heat transfer rate between air flow and duct walls. Figure (7) illustrate temperature contours of the channel using different numbers of square ribs. From the results above, it can be concluded that the use of the square ribs through channels will give the best heat transfer, and that's why it was chosen to study the effect of ribs numbers (1, 3, 5,7 and 8) and their arrangement either opposite of each other or in a staggered pattern. From the result of temperature distribution it can be noticed that the best heat transfer occur by the use of 7 square ribs distributed in a staggered pattern while the lowest heat transfer through the channel occur when using 8 square ribs distributed opposite to each other. The physical explanation will be given later while discussing the velocity vector results. Figure (8) illustrate velocity vector, the effect of ribs on the air flow inside the channel, it can be noticed that the square ribs in a staggered pattern generates the best air vortex that helps to transfer the heat form the wall to the center resulted in best heat transfer. In addition to that, soberer fluid mixing between the core area and the wall will occur when using square ribs that carry the main air flow from center area to near wall. All this will lead to great temperature gradient over the heating channel wall. The presence of ribs will raise the heat transfer rate due to the creation of main longitudinal vortex flows as helical flow due to stronger fluid mixing between the wall and core regions and that's lead to high vortex strength, longer flow path and impingement flows. Two pairs of counter rotating vortex are generated by the square ribs. The tip of square ribs divides the flow into sets, convey the maximum flow from central to near wall region, so as to soberer fluid mixing and increasing the overall performance of square shape ribs in heat transfer rate compared with all other ribs with the same surface area.

4.3 Experimental Results

A) The Influence of Reynolds Number

Figures (9) to (12) show the influence of Reynolds number on Nusselt number (Nu), heat transfer coefficient (h) for all type of ribs (semicircle, triangular, square). These figures displayed that increase of Reynolds number will lead to increase in the number of Nusselt (Nu), the coefficient of heat transfer (h) as a result of the increase in turbulence intensity due to increase in turbulence kinetic energy. In comparison with to the smooth duct, the enhancement in heat transfer of the roughened duct (with ribs) as well increases with an increase in Reynolds number. For all different Re values, square ribs provides the best (h,Nu) enhancement with increase of (28.22 to 58.23) % percentage other than duct without ribbed depend on Re values, this is due to the induction of soberer fluid mixing between duct wall and the main flow by the square ribs (Concluded from Ansys-fluent analysis). Triangular ribs produces enhancement more than duct without ribbed in (h, Nu) around (20.14 to 45.62) % depending on Re values. Circular ribs delivers lower value of (h, Nu) enhancement than duct without ribbed for all different Re values with increased of (10.87 to 28.13) % percentage depend on Re values. This is due to ribs shape, which causes reduction in the direct impact between fluid flow and ribs. Also, weaker vortex flow strength will developed due to the fact that the fluid flow nearly parallel with ribs over the other ribs and led to weaker fluid mixing between wall and core area.

B) The Influence of Heat Flux

Figures (13) to (17) demonstrate the influence of heat flux on Nusselt number (Nu), heat transfer coefficient (h) for all different type of ribs (semicircle, triangular, square). These figures indicate the influence of the heat flux on the heat transfer coefficient (h) for different type of ribs. As heat flux increases, the coefficient of heat transfer (h) rise for all type ribs. In addition to that, it is shown that the largest heat transfer coefficient (h) can be reached in the square ribs in comparison with other types. As well as form these figures, it is seen that as the heat flux increased, the Nusselt number is increased for all type of ribs. It is obvious that the increase of Nusselt number is due to the increase in the amount of heat transfer. It is obvious that the increase of Nusselt number is because of the increase in heat transfer coefficient that was affected by increase of air flow rate.

5. CONCLUSIONS

Both experimental investigation and numerical prediction were included in this work of heat transfer characteristics in a heated square duct with ribs snug repetitively on opposite side of the wall duct .The following observations could be concluded:

1- In the numerical simulation, the results attained were in a good agreement with the experimental results for the similar operative conditions that was considered for this study.
2- For all ribs type, as heat flux and Reynolds number are increased, the heat transfer coefficient (h) and Nusselt number (Nu) are increased.
3- Square ribs offer the best (h, Nu) enhancement, more than duct without ribs, for all different Re values with increase of (28.22 to 58.23) % percentage.
4- Triangular ribs produce enhancement more than duct without ribbed in (h, Nu) around (20.14 to 45.62) % respectively.
5- Circular ribs provides the lowest rate of (h, Nu) enhancement, more than duct without ribbed, for all different Re values with increase of (10.87 to 28.13) % percentage.
6- The best heat transfer occur by the use of 7 square ribs distributed in a staggered pattern while the lowest heat transfer through the channel occur when using 8 square ribs distributed opposite to each other.

NOMENCLATURE

A= the area of heat transfer, m^2
D_h= hydraulic diameter, m
h = the coefficient of connection heat transfer, W/m^2 °C
k = thermal conductivity, W/m^2 °C
Nu = Nusselt number
Re = Reynolds number
H = channel width, m
T = temperature, °C
V = velocity, m/s
Q = heat transfer rate, W
m = mass Flow Rate of Air, kg/s
u = velocity of flow, m/s
T_{bm} = bulk mean temperature, °C
T_i= inlet temperature of duct, °C
T_o= outlet temperature of duct, °C
T_s= surface temperature, °C
L = length of duct, m.
F = frictional factor
u , v, w = axial velocity
x , y, z = cartesian coordinates, m

GREEK SYMBOLS

ρ = density, kg/m^3
μ = dynamic viscosity, N s/m^2
ε = Percentage improvement of fins
δ = delta function
τ = Wall shear stress, kg/m^2

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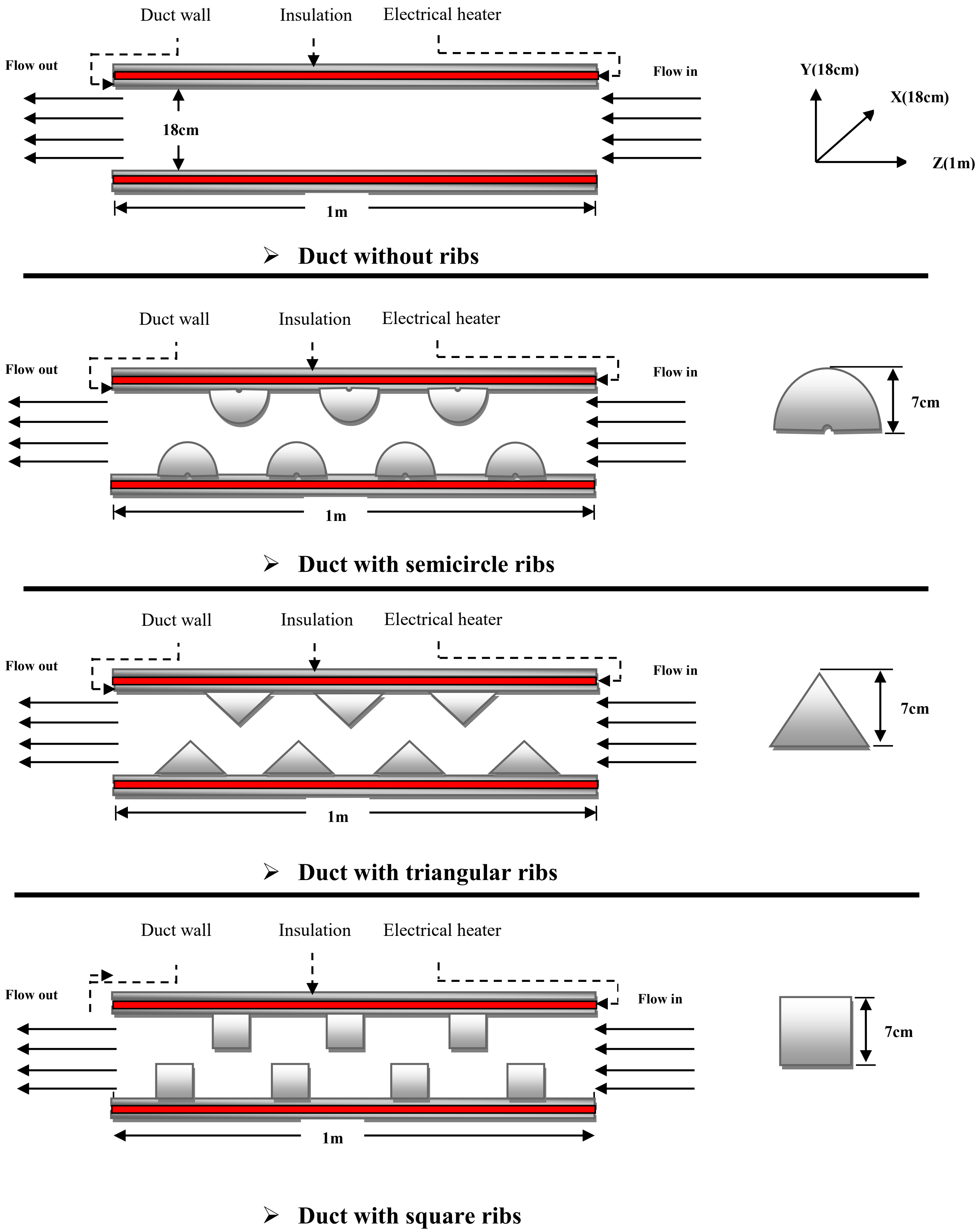

Figure (1) Physical representation of the system.

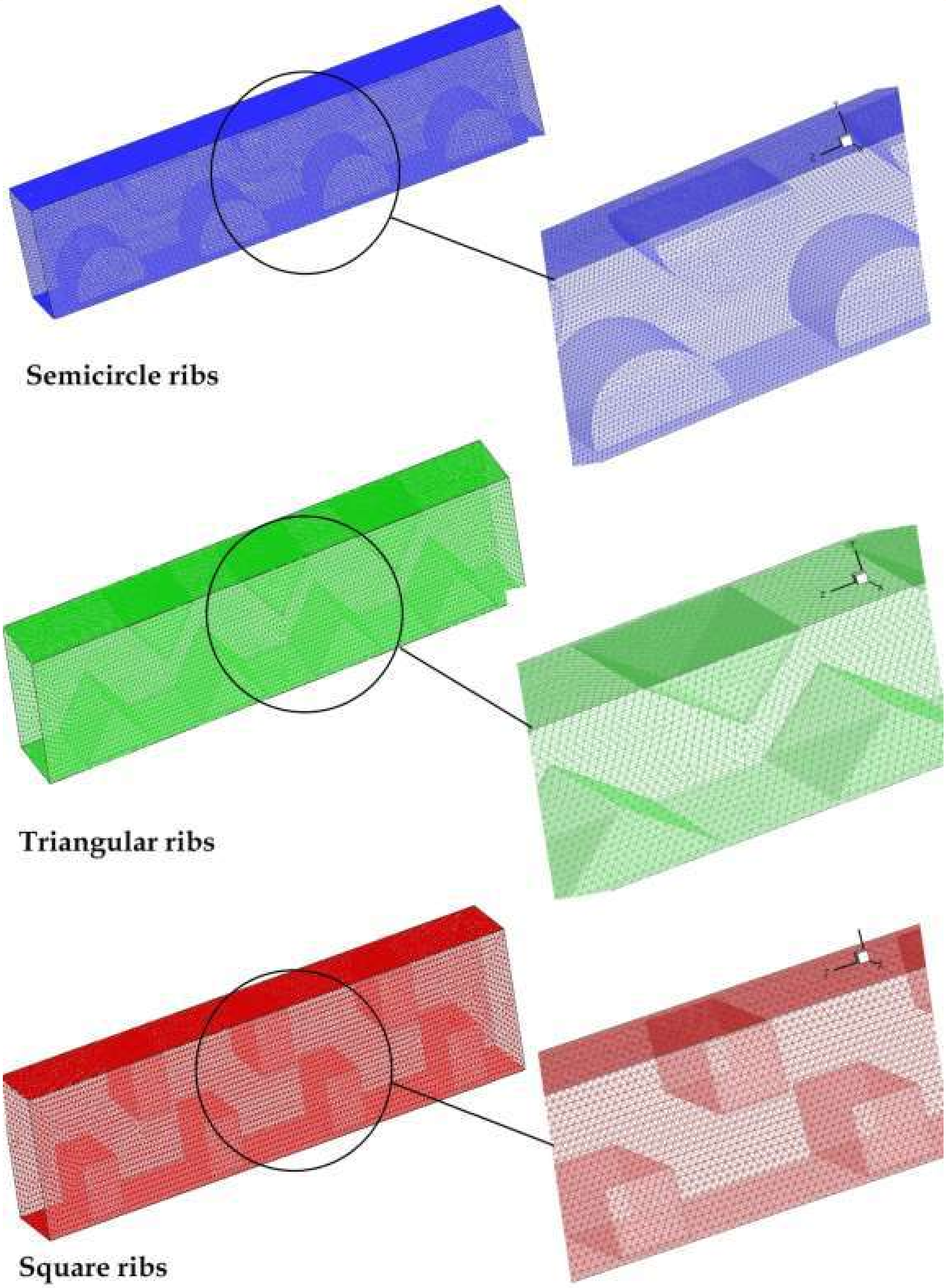

Figure (2) Mesh of present model

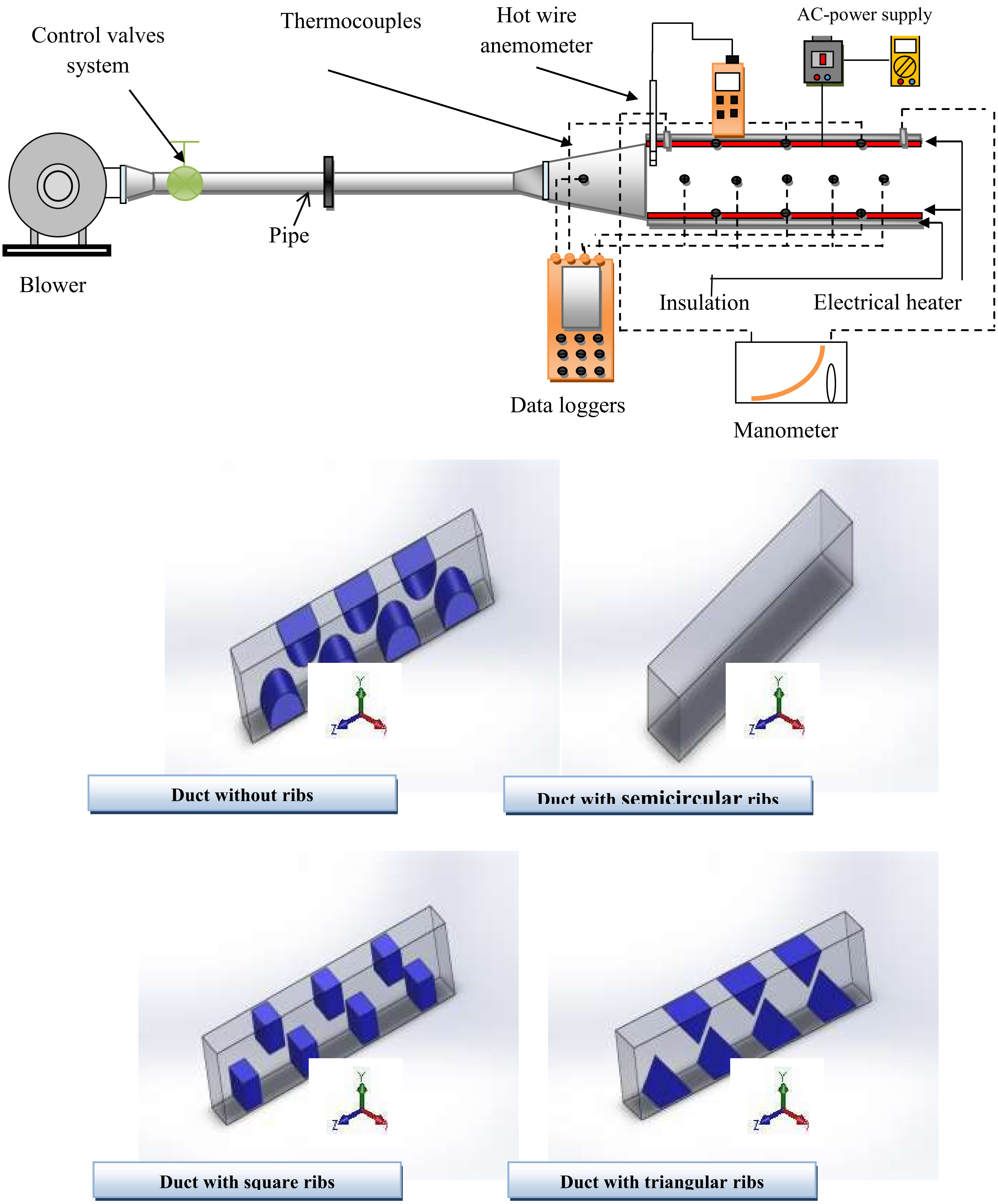

Figure (3) Description of the test rig.

Figure (4) Photograph of the test rig.

WJES Wasit Journal of Engineering Sciences
VOLUME: (6), NO. : (3) 2018

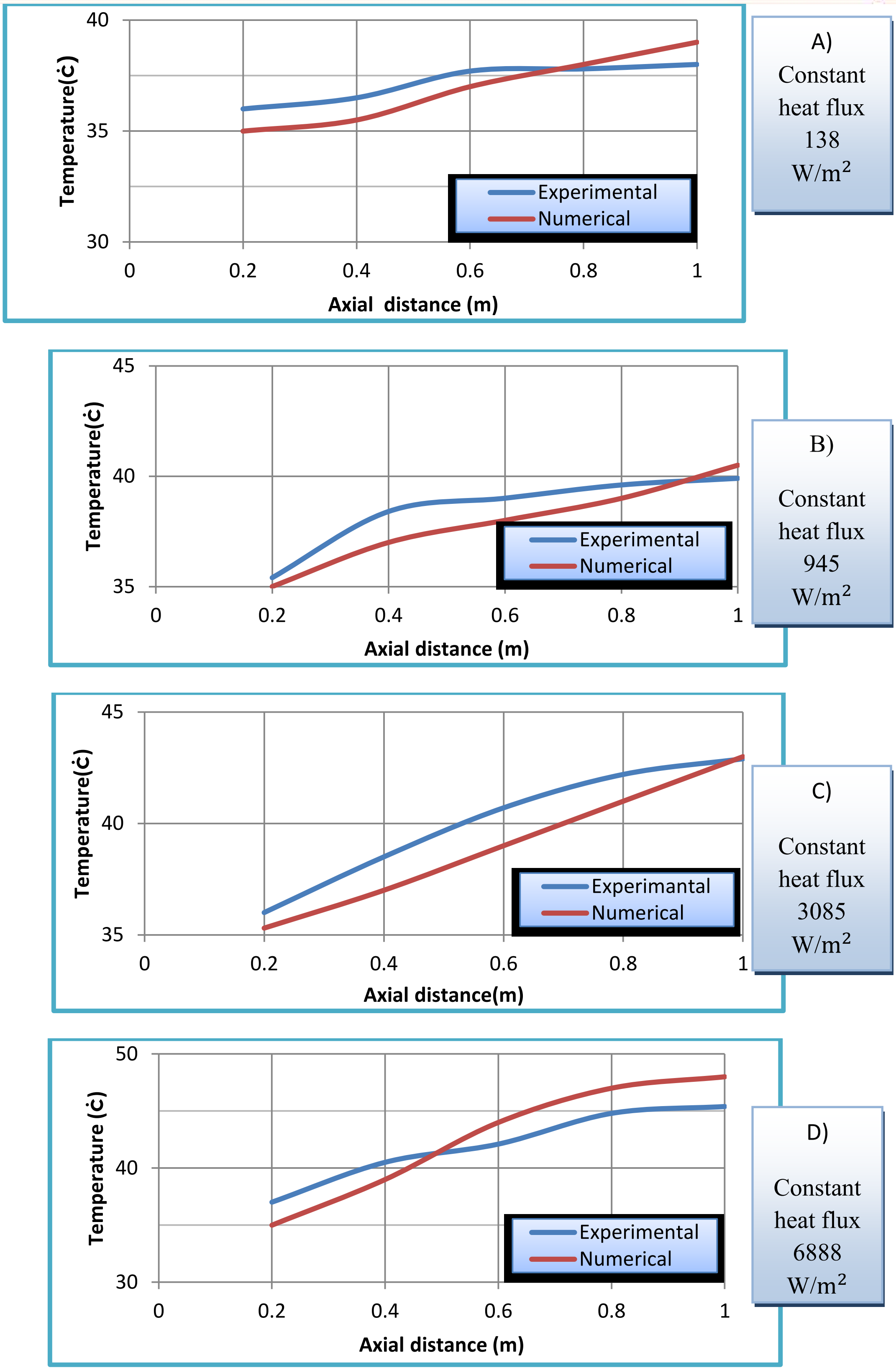

Figure (5) Comparison of experimental and numerical data.

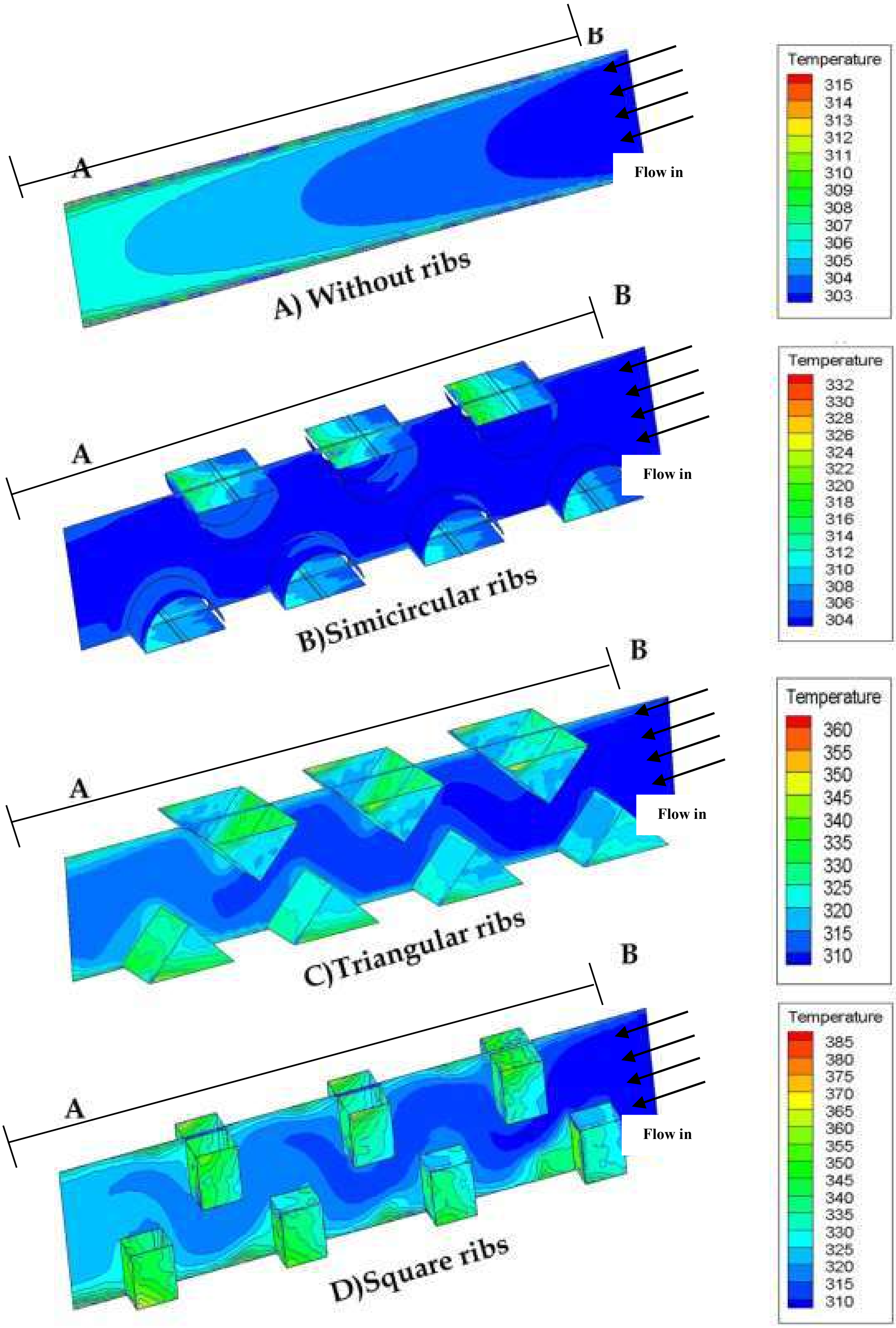

Figure (6) Temperature contour for all ribs type (x=0.09m).

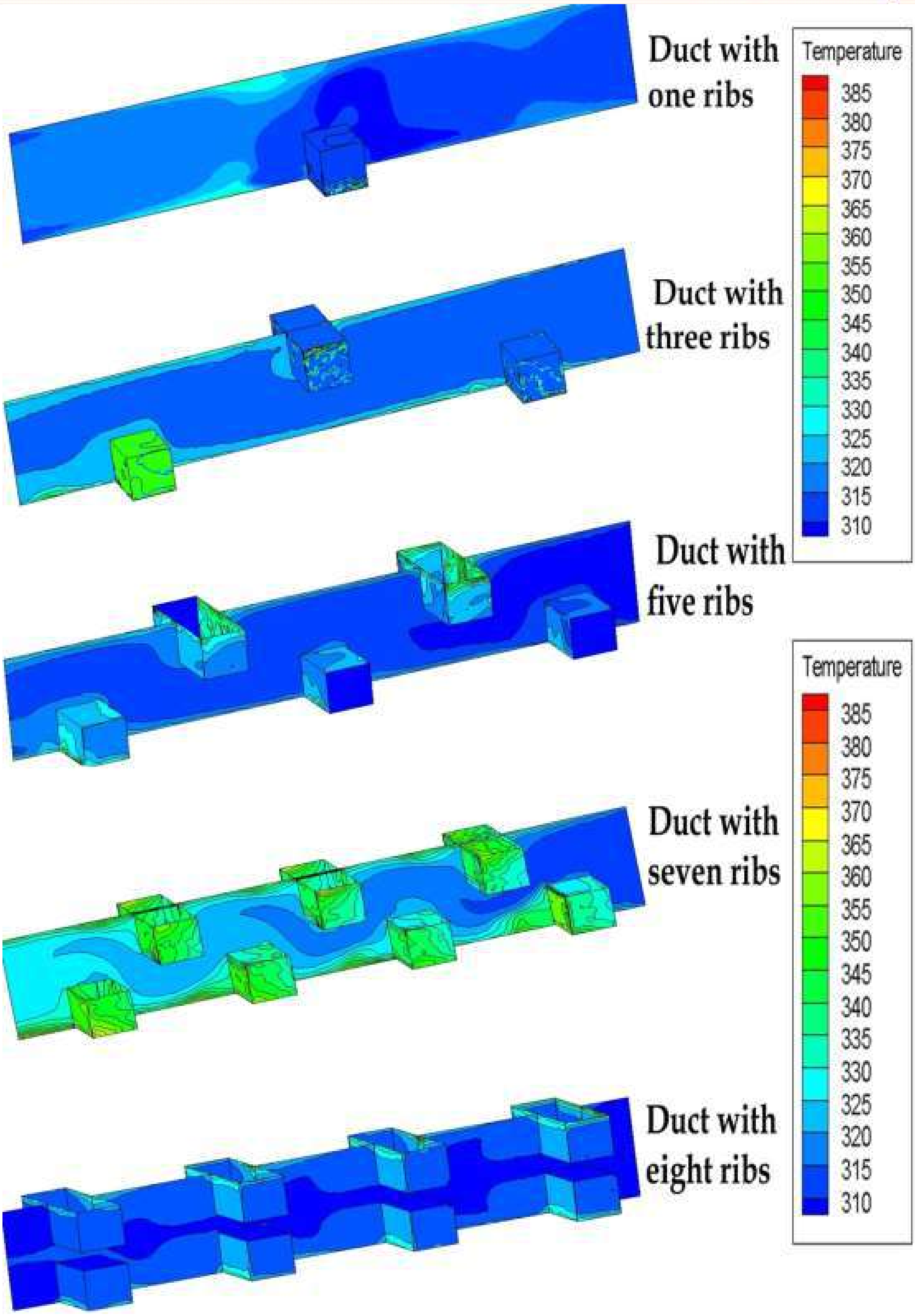

Figure (7) Temperature contours of the channel using different numbers of square ribs.

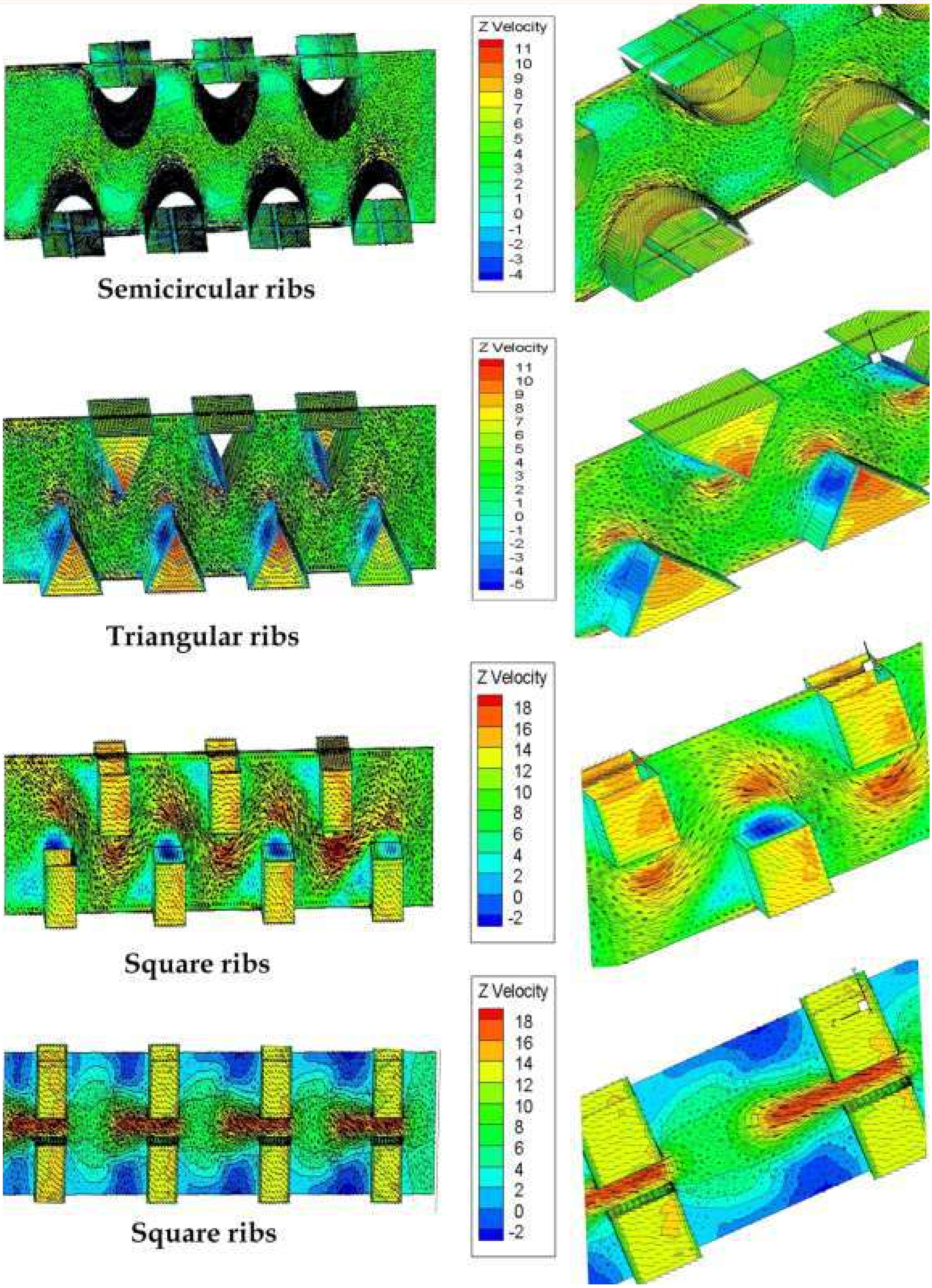

Figure (8) Velocity vector contour for all ribs type.

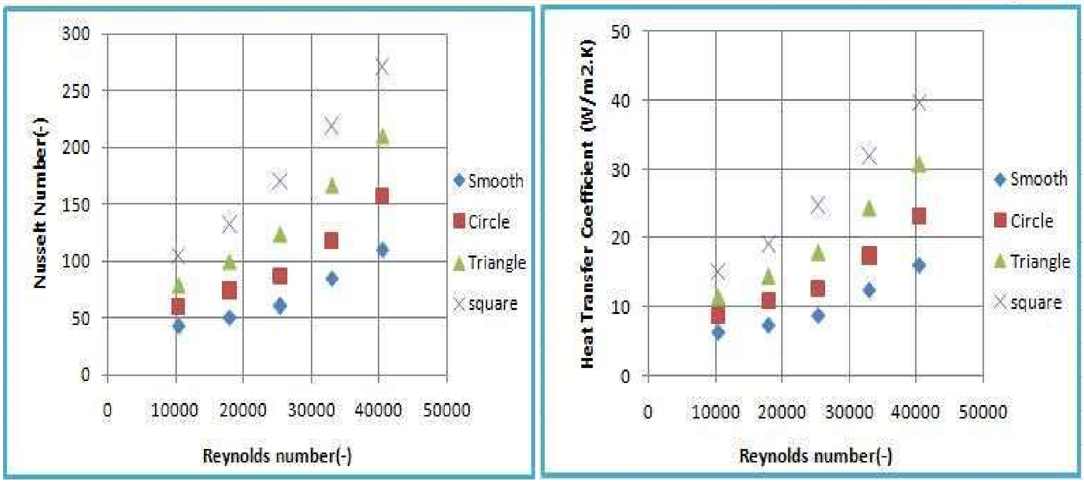

Figure (9) (h), (Nu) with Reynolds number for all ribs type (constant heat flux 138 W/m^2).

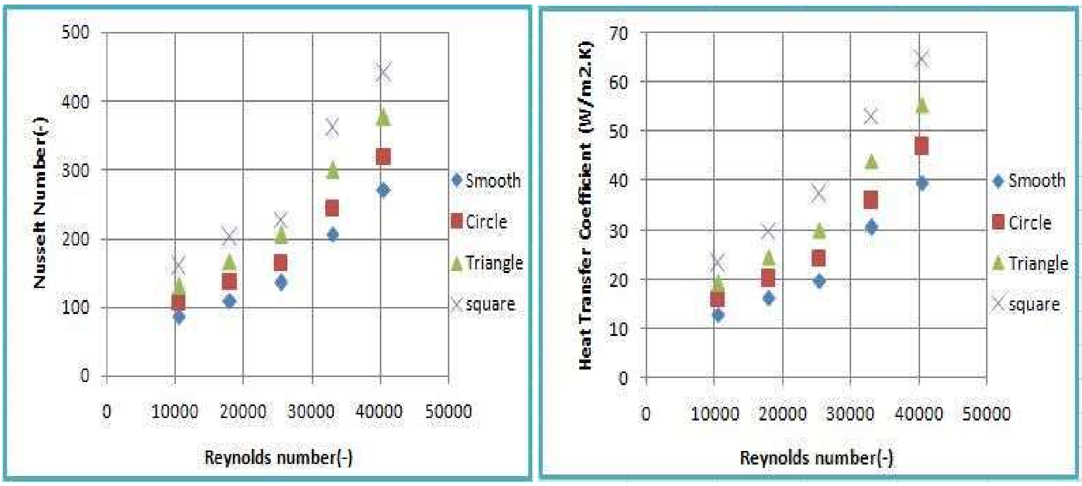

Figure (10) (h), (Nu) with Reynolds number for all ribs type (constant heat flux 945 W/m^2).

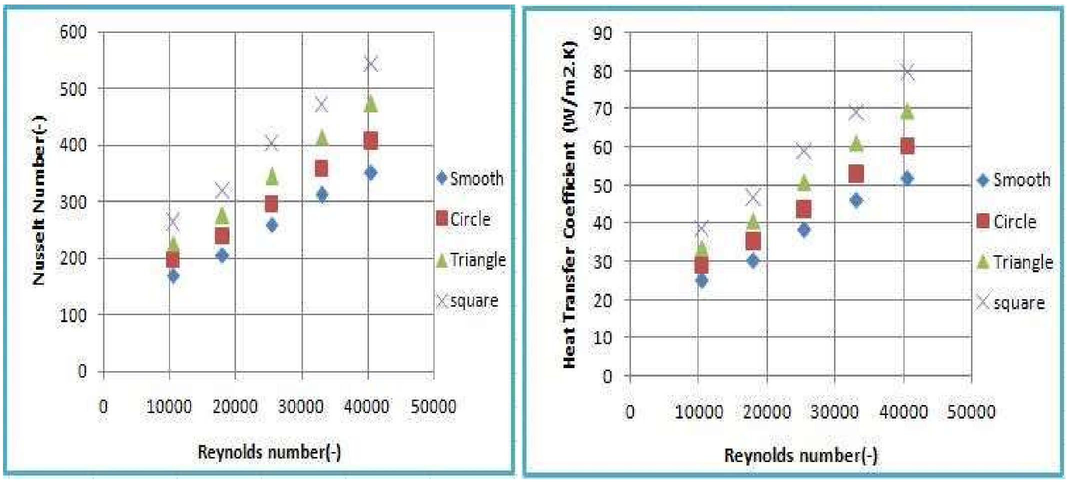

Figure (11) (h), (Nu) with Reynolds number for all ribs type (constant heat flux 3085 W/m^2).

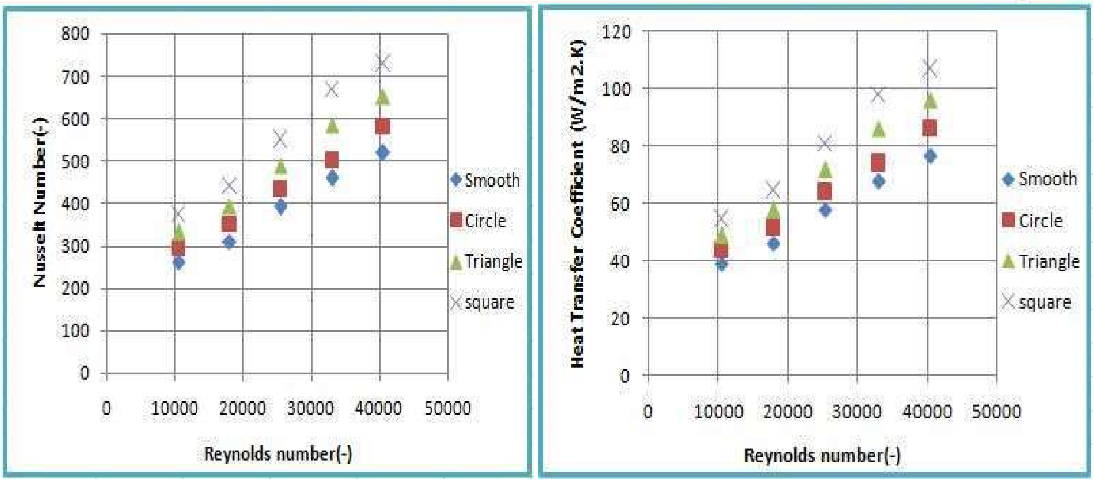

Figure (12) (h), (Nu) with Reynolds number for all ribs type (constant heat flux 6888 W/m²).

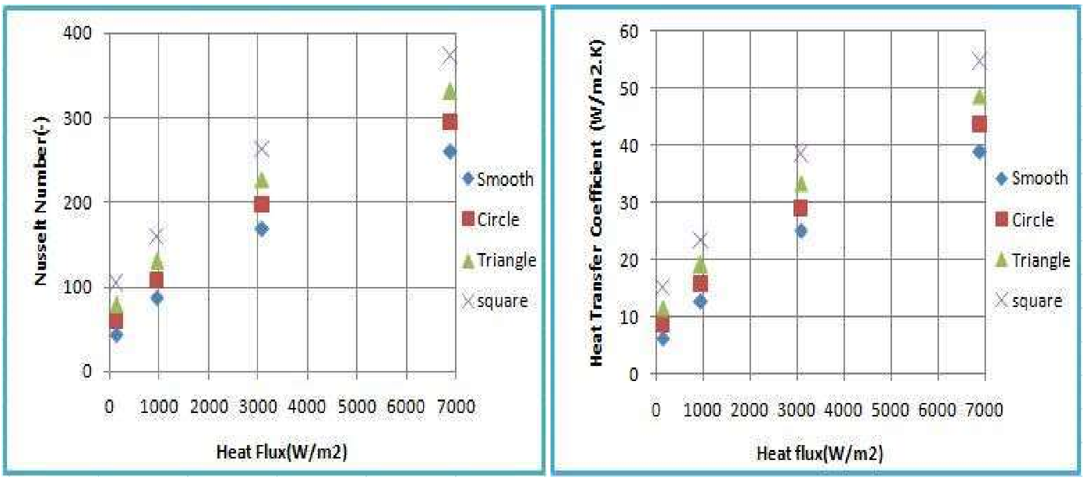

Figure (13) (h), (Nu) with different heat flux for all ribs type (Reynolds number 10460).

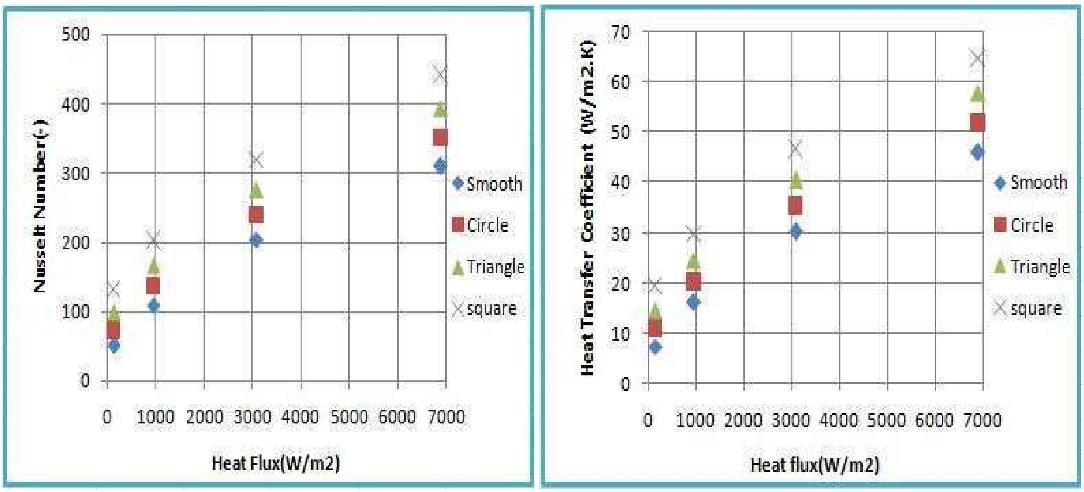

Figure (14) (h), (Nu) with different heat flux for all ribs type (Reynolds number 17950).

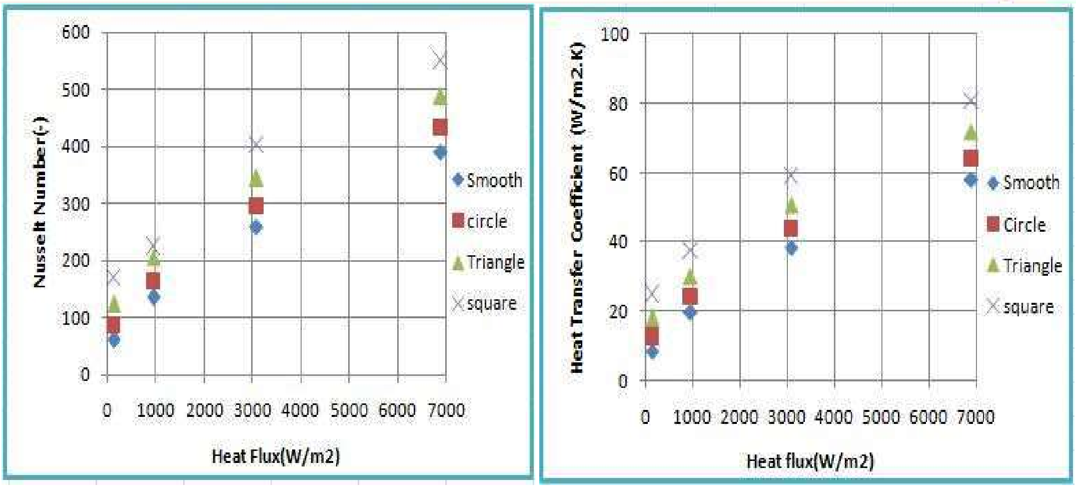

Figure (15) (h), (Nu) with different heat flux for all ribs type (Reynolds number 25400).

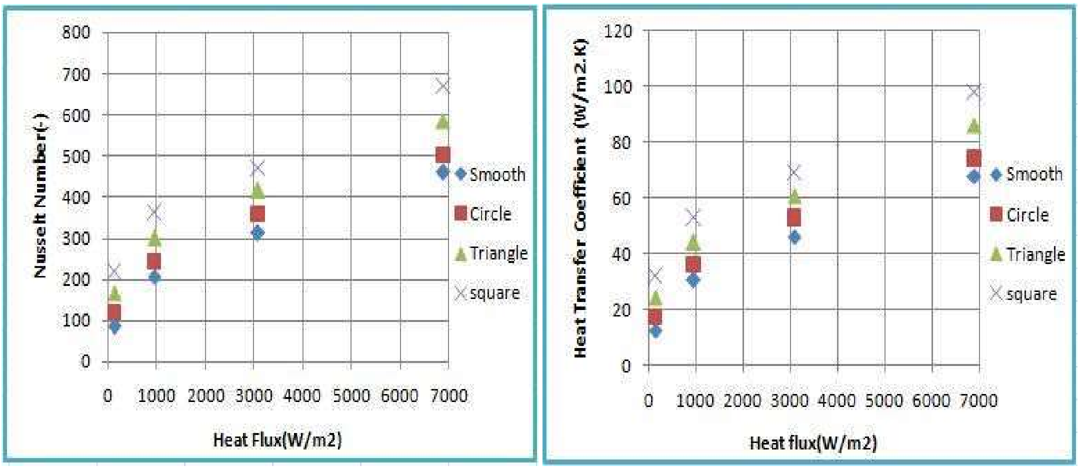

Figure (16) (h), (Nu) with different heat flux for all ribs type (Reynolds number 33000).

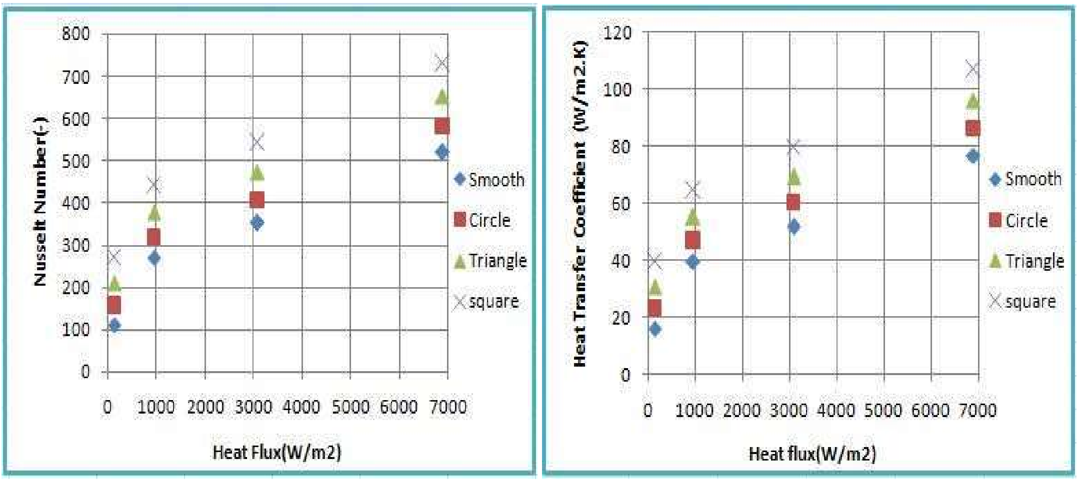

Figure (17) (h), (Nu) with different heat flux for all ribs type (Reynolds number 40500).