Journal of Engineering and Sustainable Development
Vol. 21, No. 02, March 2017
ISSN 2520-0917
www.jeasd.org

EXPERIMENTAL INVESTIGATION FOR ENHANCEMENT HEAT TRANSFER IN A CHANNEL WITH ANGLE-RIBBED TAPE AT VARIOUS ATTACK ANGLE

Dr. Rafea A.H. Albaldawi[1], Dr. Aseel K. Shyaa[2], *Suha Abd Al-Ameer[3]

1) Assist Prof., Mechanical Engineering Department, Al-Mustansiriayah University, Baghdad, Iraq.
2) Lect., Mechanical Engineering Department, Al-Mustansiriayah University, Baghdad, Iraq.
3) MSc, Mechanical Engineering Department, Al-Mustansiriayah University, Baghdad, Iraq.

Abstract: This paper presents the results of an experimental investigation for heat transfer and pressure drop characteristics in a square duct inserted with angle-ribbed straight diagonal tape. The straight tape used in the current study inserted in diagonal side of 44 mm hydraulic diameter square test section duct has uniform heat-fluxed walls. Different angled ribs were repeatedly placed at similar locations on both opposite sides of the inserted tape. The investigation covers a range of Reynolds numbers from (3400 to 20,800) according to hydraulic diameter and air is employed as working fluid. For studying the heat transfer dependency on ribs inclination angle, six different inclination angles (α) of 10°, 20°, 30°, 45°, 60°, 90° are examined at constant value for both rib blockage ratio (BR) and pitch spacing ratio (PR) of 0.2 and 1 respectively. The angle-ribbed diagonal tape generate a pair of longitudinal vortex flow through the heated duct and it is apparent that these vortices helps to induce impingement flows on the duct walls leading to drastic increase in heat transfer rate over the duct. The experimental results showed that the insertion of ribbed diagonal tape leads to a significant increase in Nu and f over smooth duct with no tape. Ribs at $\alpha = 45^{\circ}$ provides higher heat transfer and friction losses as compared with other angles about (77.78, 98.42) % respectively more than smooth duct while ribs at 10°provides higher thermal enhancement factor (TEF) by about 1.297 at lower Re.

Keywords: *square duct, angled ribs, pressure drop, heat transfer*

دراسة عملية لتحسين انتقال الحرارة في قناة بوجود شريط مزعنف بزوايا مختلفة

الخلاصة: هذا البحث يعرض نتائج التحقيقات العملية لخصائص انتقال الحرارة والاحتكاك في قناة مربعة المقطع مدرجة بشريط مستقيم قطري مزعنف. الشريط المستقيم المستخدم في هذه الدراسة مدرج بالجانب القطري لقناة اختبار مربعة المقطع قطرها الهيدروليكي 44 ملم جدرانها مسخنة بفيض حراري منتظم. زعانف مثبتة بشكل متكرر و دوري بمواقع متشابهة على كلا الجانبين المتعاكسين للشريط المستقيم المدرج بميلان عند زوايا ميل مختلفة. في هذا التحقيق تم استخدام الهواء كمائع اختبار و بمعدلات عدد رينولد تتراوح بين 3400 – 20,800. لغرض دراسة اعتمادية انتقال الحرارة على زاوية ميلان الزعانف المثبتة، تم اختبار اشرطة مزعنفة بستة زوايا ميلان مختلفة هي (90°, 60°, 45°, 30°, 20°, 10°) عند خصائص ثابتة ل (PR) = 1. and (BR) = 0.2 الشريط القطري المزعنف بزوايا ميل يولد زوج من الدوامات الطولية خلال القناة المسخنة ومن الواضح ان هذه الدوامات تساعد على تسليط جريان البثق على جدران الانبوب وبالتالي تؤدي الى ارتفاع حاد لمعدلات انتقال الحرارة خلال القناة. بينت النتائج العملية زيادة ملحوظة في معامل انتقال الحرارة وكذلك زيادة في معامل الاحتكاك عند ادراج الشريط المزعنف بشكل قطري عند مقارنته مع القناة الفارغة بدون شريط، تم الحصول على اقصى زيادة في معامل انتقال الحرارة ومعامل الاحتكاك عند الزعانف المائلة بزاوية 45° عند مقارنتها بباقي

*Corresponding Author suhaabdalamer@yahoo.com

الزوايا بمقدار % (98.42,77.78) على التوالي فوق القناة الفارغة بينما الزعانف المائلة بزاوية 10° تعطي افضل اداء حراري بمقدار 1.297 عند اقل قيمة لعدد رينولد.

1. **Introduction**

Heat transfer enhancement is simply any process objective to improve the thermal efficiency of a heat system through utilization of various techniques. Vortex generation is one of the promising techniques for heat transfer enhancement in which a suppressing turbulent flow separation happened through generating of vortex flow leading to stronger fluid mixing and impingement flows and finally a significant increase in heat and momentum transfer occurred [1]. Helical or twisted tapes, coiled wires are vortex generation devices which are effectively inserted in circular tubes while ribs, fins, baffles and winglets are suitably employed for channels. High thermal loads and decreased dimensions of ribs or baffles make them attracted many researchers. Sutapat Kwankaomeng et al. 2010 [2]: attempt to carry out an experimental investigation for heat transfer and friction loss through a square channel fitted with 45°inclined baffles on the two lower and upper walls of the duct in staggered array. Re ranging from (4000 to 25,000) and different of baffle blockage ratio (BR) from 0.1 to 0.4 were examined. The experimental results showed that baffles with blockage ratio of 0.4 perform higher Nu and f than other tested baffles. Pongjet Promvonge 2010 [3] presented experimentally the effect of employing ribs and winglet vortex generator type (at entrance of test section) together on thermal performance of channel with constant heat flux walls in Re from (5000 to 22,000). Measurements taken for different winglet attack angle $(\alpha = 60^{\circ}, 45^{\circ}, 30^{\circ})$ with fixed blockage and pitch ratios (e/H) = 0.13, (P/e) = 1.33 respectively with inline and staggered arrangements. Results showed that higher Nu and f performed in larger winglet attack angle and inline rib arrangement but highest thermal performance in staggered ribs array with 30° winglet. Giovanni Tanda 2011 [4] studied experimentally the effect of rib spacing on thermal behavior of a rectangular channel. Angled rib of 45° inclination angle mounted on one or two of the channel walls were examined with different rib pitch to height ratios (P/e) 6.66, 10, 13.33, and 20 at Re from (9000 to 35,500). It was found that pitch to height ratio (P/e) of 13.33 yields the optimum performance for one ribbed wall channel and for two ribbed wall channel the optimum heat transfer occurred at (P/e) = 6.66–10. The thermal performance was studied numerically for isothermal wall square channel attached by 45° inclined baffles mounted repeatedly on one and two opposite walls in inline arrangement by Pongjet Promvonge et al. 2010 [5],[6]. These two studies conducted for one purpose which is the influence of varying baffle blockage ratio on heat transfer and pressure loss with only one pitch ratio (PR) fixed at 1 for the two studies. The domain examined in [5] is (BR) = 0.1 to 0.5 and Re = 100 to 1200 whereas in [6] the examined (BR) = 0.05 to 0.3 and Re from (100 to1000). Based on the computational results of these two studies it is apparent that the increase in baffle height lead to increase in both Nu and f. The maximum value of thermal enhancement about 2.6 at BR=0.2 and

Re = 1000 for using the 45° inline baffles on two walls. P. Promvonge et al. 2012 [7] made experimental work to examine the effect of different fin blockage and pitch ratios of (0.1 to 0.3) and (1 to 3) respectively on thermal performance. This fins inclined with single 30° attack angle and attached in tandem on two sides of tape inserted diagonally in uniform heat flux walls of square duct. This work conducted for Re number from (4000 to 23,000). The work reports that the best thermal performance about 1.8 occurred at lower Reynolds number for blockage ratio BR=0.2 and pitch ratio PR=1. K.Yongsiri et al. 2014 [8] performed simulations to analyze the characteristics of heat transfer and air flow in a channel ribbed with inclined detached-ribs pointed on the lower wall of the channel at different attack angles (α) of 0°, 15°, 30°, 45°, 60°, 75°, 105°,120°, 135°, 150°, and 165° for turbulent flow with Re range from the 4000 to 24,000 of Re. Simulation results showed that detached-ribs at attack angle $(\alpha) = 60^{\circ}$ produce comparable heat transfer rate 1.74 times of the smooth channel and rib with $(\alpha) = 120^{\circ}$ yield higher thermal performance factor of 1.21 than other angles.

From the previous experimental and numerical works most of these investigations focused on studying the heat transfer and pressure drop characteristics for employing inclined baffles or ribs mounted on the upper or lower or both upper and lower walls of the channel with different blockage and pitch ratios furthermore usage of combination from two or more types of vortex generators together.

In the present work it is intended to enhance heat transfer in a square duct by means of vortex generation technique through insertion of diagonal straight tape attached with angled ribs at two opposite sides in a square duct. The diagonal tape that used in this work suitable for square ducts because of it's characteristically location in diagonal side of square duct which considered as the shared base of two imaginary isosceles triangular ducts. Combination of all benefits of rib, baffle, winglet and coil wire are represented by this tape which form a situation where all enhancement vortex devices that mentioned above work in unison to induce a secondary swirl flow. Where angled ribs (small height projections) that mounted repeatedly on the tape on its double sided helps to better flow mixing between the core and wall regions with low pressure drop penalty and the tape looks like baffles (thin elements of greater heights than ribs) which can disturb the bulk flow, promote mixing of flow and finally give a significantly high heat transfer rate. Furthermore this tape induce vortex flow with ease of practical use and low cost like twisted tapes also wash up trapped flow at duct corners that occurred when inserting twisted tape or coil wire in a duct.

The main objective of this study is to investigate the influence of inclination angle (α) of rib with flow direction on thermal enhancement performance through testing six different ribs inclination angles (α) of 10°, 20°, 30°, 45°, 60°, 90° in turbulent flow with a range of Re from 3400 to 20,800. Optimum ribs blockage ratio b/H=0.2 and pitch ratio P/H=1 are taken from [7] also an empirical correlation develops for heat transfer and pressure drop as a function of the inclination angle (α) for range studies.

2. Experimental Setup

The experimental test rig is shown in figure (1) as a photograph image and figure (2) as a schematic diagram. The test rig conducted in an open-loop air flow circuit and consists of the following component: air supply unit, control valves system, air duct, plenum duct, test section, tested straight tapes, power supply unit and measurement equipment. High pressure blower connect to control valves system (bypass valve) and a horizontal square duct, which include: inlet section (100 cm), plenum section (15 cm), test section (80 cm) and exit section (50 cm). In order to get the correct Nusselt number values of turbulence flow regime, the flow should be consider fully developed flow, so that the length of inlet section was taken 100 cm according to $10 \leq L/D \leq 60$ [9]. The plenum duct section constructed from aluminum square duct with 3mm thickness. Its dimension is 15cm in length and 5 cm wide. A 4mm honeycomb cells 15cm in length was inserted inside the duct according to British standard 1042 [10] its length to diameter is $L/D \geq 20$ to reduce the turbulence level in the incoming air flow through the duct. A test section constructed from aluminum square duct with 5 cm height (H), 3 mm thickness (t) and 80cm long (L). It consist of four heating walls was heated by electrical heater tape with resistance 7 ohm/m, 0.5 mm thickness and 3mm width attached on the outer surface walls for all side of square duct in regular distribution to provide uniform heat flux. To avoid electrical short an electrical insulation was fixed between wall surface and heater tape also thermal paste with good thermal conductance was applied between electrical insulation and wall surface to ensure good thermal coupling and heat dissipation which due to reduce contact resistance. The outer surface of test section was well insulated by using fiber glass with thermal conductivity 0.04 $W/m.k$ and thickness 2 cm to minimize heat loss by conduction. For measuring the temperature distribution along the upper, lower and two sides wall of test section, 32 thermocouples type K with junction diameter 0.8 mm installed in a drilled holes in the thickness of test section four walls with respective junction positioned with 2 mm of the inside wall and axial separation was 11 cm. To measure the inlet and outlet air bulk temperature two thermocouples were fixed on the up-stream and down-stream of the test section. Additionally one thermocouple were positioned to outer surface of fiberglass insulation and the second to ambient to measure the heat loss to surrounding. All thermocouples output voltages were fed into 3 data loggers and then download to personal computer via 3 SD cards. Two pressure taps were located on the top wall in the centerline of the duct for measuring the pressure drop across the test section one of these taps was one positioned at up-stream of test section leading edge while the other fitted at down-stream of test section leading edge. The pressure drop was measured by using curve tube manometer with fluid of 0.826 specific gratify to ensure accurate measurement of low pressure drop encountered low Reynolds Number. The straight tapes that used in the current study was made of aluminum plate with its dimensions of ($62.2\times800\times0.9$) mm inserted diagonally through the test section. Angled ribs with optimum blockage ratio $BR = (b/H) = 0.2$ and optimum pitch ratio $(PR) = P/H = 1$ [7] made from aluminum sheet thickness of (e) = 0.4 mm were repeatedly placed at similar locations on both opposite sides of the inserted tape by using a superglue adhesive as shown in figure (3). Six tested model tapes were prepared with six inclination angles for ribs which is (α) = (10°, 20°, 30°, 45°, 60°, 90°) all tested models are shown in figure (4). Air as the test fluid in this experimental study was directed into the system by a blower which operates by a single-phase electric motor (0.8 KW). The operation speed of the blower was varied by using flow control valves (bypass valve) to provide the desired air flow rate. The speed of air in the system was measured by using hot wire anemometer. For each test recorded the data of temperature duct wall, speed of air flow and pressure drop at state condition. Recorded every 15 min. until it reach to steady state (when the temperature changed about 0.1 °C over this period).

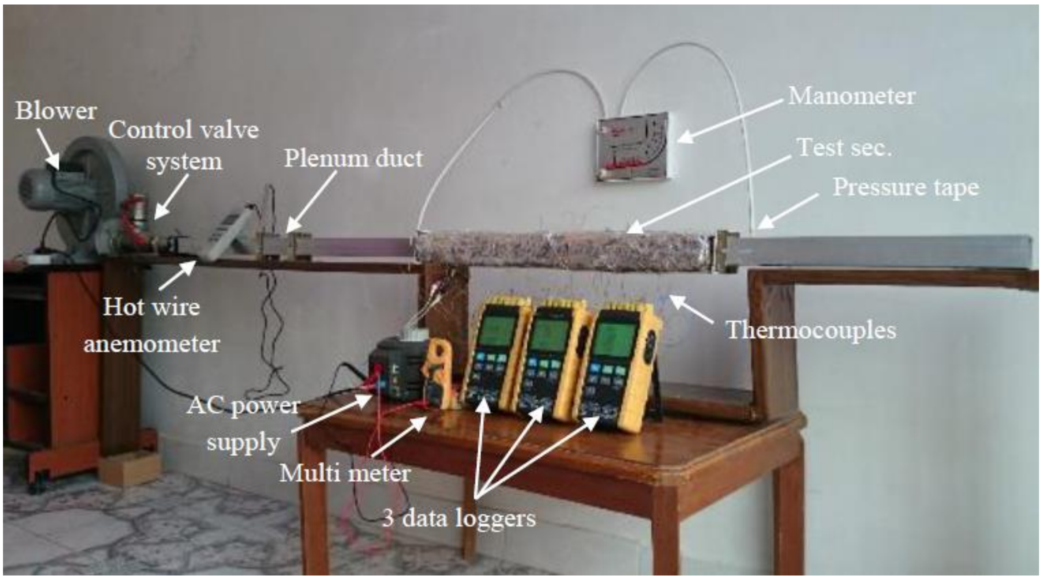


Figure (1): The experimental test rig

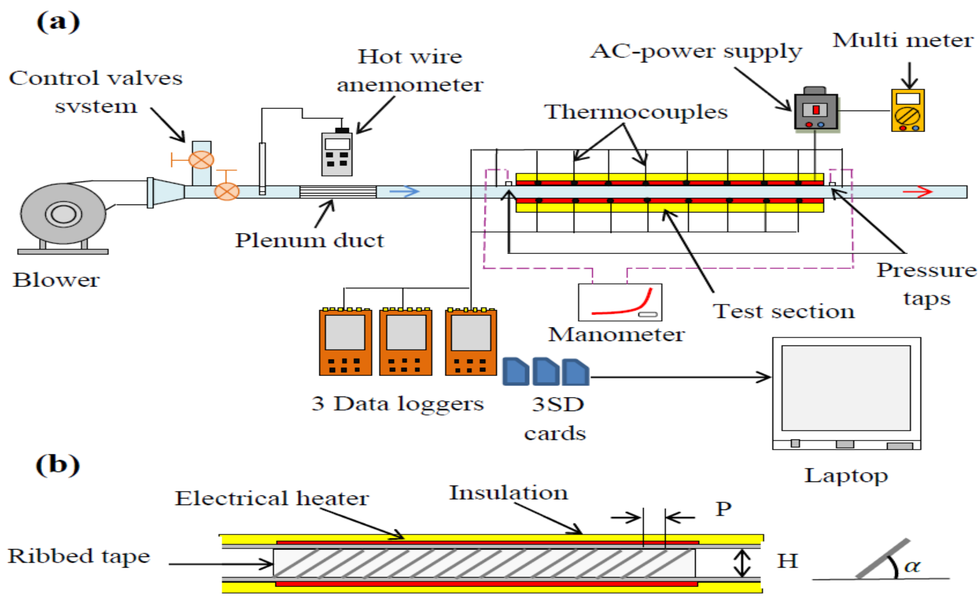


Figure (2) (a) Schematic of experimental apparatus. (b) Test sections inserted with ribbed tapes.

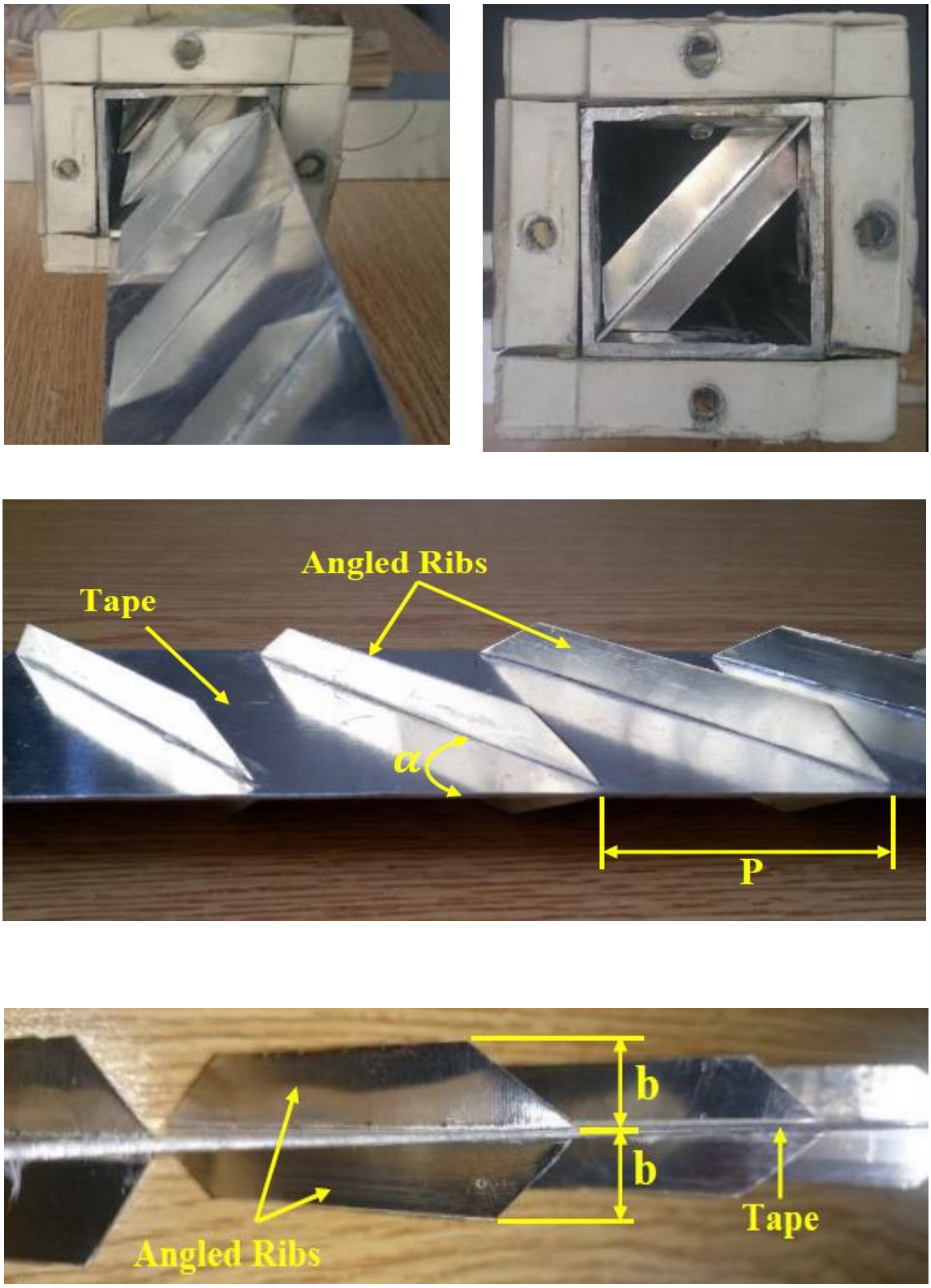


Figure (3): Angled ribbed tape inserted diagonally in the test section with vision of geometry parameters.

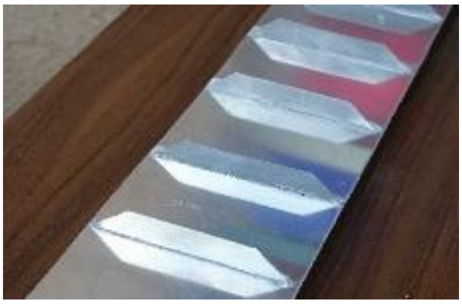
$\alpha = 10^{\circ}$

$\alpha = 20^{\circ}$

$\alpha = 30^{\circ}$

$\alpha = 45^{\circ}$

$\alpha = 60^{\circ}$



$\alpha = 90^{\circ}$

Figure (4): Angled ribbed tape with different angles.

3. Data Reduction

The following procedures has been followed to perform the steady state heat transfer rate absorbed by the cooling air that flowing inside the tested duct.

3.1. Air properties

For internal flow, the air properties are taken at the bulk mean temperature as follows:

$$\mathrm{T_{bm}} = \frac{\mathrm{T_{inlet}} + \mathrm{T_{outlet}}}{2} \quad (1)$$

3.2. Hydraulic Diameter

For non-circular duct the hydraulic diameter are calculated as follows:

$$D_h = \frac{4\,A}{P} \quad (2)$$

For square duct

$$D_h = \frac{4\,(a*a)}{4\,a} = a \quad (3)$$

3.3. Reynolds Number

The Reynolds number define as the ratio of fluid inertia forces to viscous forces which is a dimensionless quantity, and is expressed for internal flow

$$Re = \frac{\rho\,U\,D_h}{\mu} \quad (4)$$

3.4. Mass Flow Rate

The mass flow rate of air can calculated from the following equation.

$$\dot{m} = \rho U A_c \quad (5)$$

3.5. Rate of Heat Transfer

The conservation of energy equation for the steady flow of a fluid in square duct which is the sensible heat gained by the fluid can be expressed as:

$$Q_1 = \dot{m}\,cp\,(T_{outlet} - T_{inlet}) \quad (6)$$

3.6. Average Walls Surface Temperature

The average walls surface temperature can be calculated from following equation:

$$\bar{T}_s = \sum \frac{\mathrm{T_s}}{32} \quad (7)$$

3.7. Power Input

The total heat generated by the electrical heater is calculated as:

$$Q_t = (I \times V) \qquad \text{Watt} \tag{8}$$

In order to get the actual heat flux values supplied from electrical heater to test section, the heat losses from input power must be calculated. The heat input from the electrical heater is:

$$Q_2 = Q_{net} - Q_{losses} \tag{9}$$

3.8. Heat Losses

1. The heat losses due to extra electrical wire length, It can be calculated according to the following equation [11]

$$Q_{net} = Q_t \times \left(\frac{\text{total electrical wire length} - \text{extra electrical wire length}}{\text{total electrical wire length}}\right) \tag{10}$$

2. Calculation the heat losses caused by radiation by the following equation.

$$Q_{rad} = \epsilon\sigma A_s\left(T_{ins}^4 - T_{surr}^4\right) \qquad \text{Watt} \tag{11}$$

3. Calculation the heat losses caused by conduction from the heated walls through the fiber glass thermal insulation layer by the following equation.

$$Q_{cond.} = K_{f.g}\, A_s K_{f.g}\, A_s \left(\frac{\bar{T}_s - T_{ins}}{x}\right) \qquad \text{Watt} \tag{12}$$

3.9. Heat Transfer by Convection

The net convection heat transfer calculated by the following equation.

$$Q_2 = Q_{net} - \sum(Q_{rad.} + Q_{cond.}) \qquad \text{Watt} \tag{13}$$

The heat balance between the heat input from the heater Q_2 and heat input to the fluid Q_1 the actual heat flux is then evaluated as

$$Q_{conv.} = (Q_1 + Q_2)/2 \tag{14}$$

Via Newton's cooling law convection heat transfer was termed for the first time and expressed as:

$$Q_{conv.} = h\, A_s\, (\bar{T}_{s} - T_{bm}) \qquad \text{Watt} \tag{15}$$

3.10. Heat Flux by Convection

The constant walls surface heat flux calculated by the following equation.

$$q_{conv.} = \frac{Q_{conv.}}{A_s} \qquad (\text{W/m}^2) \tag{16}$$

3.11. Local Heat Transfer Coefficient

The local heat transfer coefficient is calculated by the following equation.

$$h = \frac{q_{conv.}}{T_{s-}T_{bm}} \qquad (W/m^2 °C) \qquad (17)$$

3.12. Average Heat Transfer Coefficient

The average heat transfer coefficient is calculated by the following equation.

$$h_{average} = \frac{q_{conv.}}{\bar{T}_{s-}T_{bm}} \qquad (W/m^2 °C) \qquad (18)$$

3.13. Nusselt Number

The Nusselt is known as the ratio of the fluid conductive to the convective thermal resistance and calculated by the following equation.

$$Nu = \frac{h\, D_h}{K_{air}} \qquad (19)$$

3.14. Friction Factor

The friction factor across the test section calculated from the following equation:

$$f = \frac{2\, \Delta P\, D_h}{L\, \rho\, U^2} \qquad (20)$$

3.15. Thermal Performance Enhancement factor

The ratio of convective heat transfer coefficient h for an enhanced surface to that of a smooth surface h_o at constant pumping power is the thermal performance enhancement factor (TEF).

$$TEF = \frac{h}{h_0}\Big|_{PP} = \frac{Nu}{Nu_0}\Big|_{PP} \left(\frac{Nu}{Nu_0}\right)\left(\frac{f_0}{f}\right)^{1/3} \qquad (21)$$

4. Results and Discussions

4.1 Validation Results

Verification of heat transfer and friction characteristic of smooth wall square duct is performed in terms of Nu and f. A comparison of Nu and f is obtained from the present work with those from correlations of Dittus-Boelter for heat transfer coefficient and correlations of Blasius for friction factor eq. (22) and (23) found in the literature [9] under a similar operating condition as shown in figure (5) and (6). The present smooth duct results are in agreement within ±20% with correlation data.

Correlation of Dittus-Boelter, for heating

$$Nu = 0.023Re^{0.8} \qquad (22)$$

Correlation of Blasius, for 3000≤ Re ≤20,000

$$f = 0.316\, Re^{-0.25} \qquad (23)$$

4.2 Experimental Results

4.2.1 Heat transfer

The variation of Nu obtained under a different turbulent flow condition with ribs at different angle $(10^\circ, 20^\circ, 30^\circ, 45^\circ, 60^\circ$ and $90^\circ)$ is presented in figure (7) and (8). It is visible that:

1. Nu for all ribs angle with a similar trend in comparison with smooth duct and Nu increases with rise of Re. This is due to increase in turbulent intensity as the Re increases resulting in more destruction of the boundary layer.
2. The presence of inserted tape with flat ribs creates two main longitudinal vortex flows is asymmetry helical flow can help to increase Nu rate in the duct because of stronger fluid mixing between the core and wall regions leading to longer flow path, high vortex strength and impingement flows.
3. Inserted tape with ribs at different angle $(10^\circ, 20^\circ, 30^\circ, 45^\circ, 60^\circ$ and $90^\circ)$ produces enhancement in Nu rate around (45.08-57.7) %, (51.65 -62.4) %, (57.37-67.24) %,(65.57-77.78) %,(62.52-76.06) % and (59.41-71.86) % respectively, more than smooth duct depending on Re values.
4. Ribs at angle 45°provides thebetter Nu enhancement for all different Re values with increased of (65.57 % to 77.78 %) percentage more than smooth duct depend on Re number values this is because the ribs at angle 45° created stronger vortex flow strength over the other angles can induce better fluid mixing between duct wall and the main flow.
5. Ribs at angle 10°provides the lower value of Nu enhancement for all different Re values with increased of (45.08-57.7) % percentage more than smooth duct depend on Re number values.

4.2.2 Friction factor

The variation of friction losses obtained under a different turbulent flow condition with tape and ribs fitted at different angle $(10^\circ, 20^\circ, 30^\circ, 45^\circ, 60^\circ$ and $90^\circ)$ and Re is depicted in figure (9) and (10) it is visible that:

1. The use of inserted tape diagonally with ribs at different angle leads to a substantial increase in f over smooth duct and f shows a slight decrease with the rise of Re values this because presence of inserted tape diagonally with ribs created vortex flow due to high viscous losses near the wall, flow blockage causes extra forces exerted by reverse flow and higher friction of increasing surface area.
2. The inserted tape diagonally with ribs at different angle $(10^\circ, 20^\circ, 30^\circ, 45^\circ, 60^\circ$ and $90^\circ)$ produces increase in f about (79.29- 86.60) %, (87.14- 92.06) %, (92.18- 95.01) %, (97.14- 98.42) % ,(96.88- 98.20538) %, (96.36- 97.32) % respectively, more than smooth duct depending on Re values.

3. Ribs at angle 45° provides higher f losses for all different Re values with increased of (97.14- 98.42) %, percentage over smooth duct depend on Re values.
4. Ribs at angle 10°provides the lower f losses for all different Re values with increased of (79.29- 86.60) % percentage over the smooth duct depend on Re values.

4.2.3 Performance evaluation

Nu ratio, Nu/Nu_0 defined as a ratio of augmented Nu to Nu of smooth channel. Figure (11) and (12) displayed Nu ratio, Nu/Nu_0 obtained under a different turbulent flow condition with tape and ribs fitted at different angle(10°,20° ,30°, 45°,60° and 90°) from the figure, it is visible that Nu/Nu_0 value tends to decrease with the rise of Re for all ribs angles. Ribs angle 45° provides the higher value of Nu/Nu_0 for all Re in comparison with other angles. Maximum Nu/Nu_0 value is found to be about 4.50 for $\alpha = 45^{\circ}$ at Re = 3453.

Figure (13) and (14) show the variation of f ratio, f/f_0 obtained under a different turbulent flow condition for various angle of ribs. From the figures, it is noted that, f/f_0 tends to decrease with rise of Re. Ribs with $\alpha = 45^{\circ}$ gives the highest value of f/f_0 about 63.588 at Re = 3453.

Figure (15) depicted the variation of thermal performance enhancement factor, TEF, obtain from the Nu and f ratios measurement are compared at an identical pumping power condition plotted against the Re number values for ribs at different angle. It is observed that:

1. TEF tends to reduce with the rising in Re for all angles value . It is interesting to note that TEF tends to increase with reduce in ribs angle. In addition seen that ribs at angle 10° gives the highest TEF at lower Re and is about 1.297
2. For ribs at angle (10°,20° ,30°, 45° and 60°) provides (12.1-24.21) %,(6.98-21.78) %,(5.34-18.6) %, (5.71-8) % and (2.75-2.8) % respectively better TEF over the angle 90° depend on Re values.

The Nu and f were estimated from the experimental results for using the inserted tape diagonally with ribs fitted at different angle are correlated as function of Reynolds number (Re), prandtl number (P_r) and ribs angle (α). This is accomplished according to algebraic expression of the form ($R =$ a $\mathrm{X}_1^{\mathrm{m}_1}$ $\mathrm{X}_2^{\mathrm{m}_2}$…..). The results of empirical correlations are valid for Re from 3400 to 20000 as following:

$$N_u = 0.6269\ (Re)^{0.5761}\ (\sin\alpha)^{0.26066}\ Pr^{0.4} \tag{24}$$

$$f = 212.888\ (Re)^{-0.5459}\ (\sin\alpha)^{1.1181} \tag{25}$$

Figure (16) show the comparison between experimental and predicted Nu for all cases. In this figures, the majority of the measured data falls with ± 20 % for prediction Nu and$f$.

This work presented the experimental investigation of heat transfer and friction factor characteristics in a heated square duct of inserted tape diagonally with ribs fitted repeatedly on opposite side of the tape at different ribs angle (10°,20° ,30°, 45°,60° and 90°) for the turbulent flow, Reynolds number of 3400 to 20,800.The following remark could be concluded:

1. For general observation, the use ribbed tape diagonally causes high pressure drop increase, (f/fo = 63.588) but also provides a considerable heat transfer augmentation in the duct, (Nu/ Nuo = 4.5) depend on ribs angle and Re.
2. The Ribs at angle 45° provides the better heat transfer enhancement increase of 77.78 % over the smooth duct as compression with other angles range in current study.
3. The Ribs at angle 45° produce the high fraction factor losses increase of 98.42 % over the smooth duct as compression with other angles range in current study.
4. The TEF tends to reduce with the rising in Reynolds number for all angles value . It is interesting to note that the TEF tends to increase with reduce in ribs angle. And seen that the ribs at angle 10° gives the highest TEF at lower Re number and is about 1.297

5. Conclusions

This work presented the experimental investigation of heat transfer and friction factor characteristics in a heated square duct of inserted tape diagonally with ribs fitted repeatedly on opposite side of the tape at different ribs angle (10°,20° ,30°, 45°,60° and 90°) for the turbulent flow, Reynolds number of 3400 to 20,800.The following remarks could be concluded:

1. For general observation, the use ribbed tape diagonally causes high pressure drop increase, (f/fo = 63.588) but also provides a considerable heat transfer augmentation in the duct, (Nu/ Nuo = 4.5) depend on ribs angle and Re.
2. The Ribs at angle 45° provides the better heat transfer enhancement increase of 77.78 % over the smooth duct as compression with other angles range in current study.
3. The Ribs at angle 45° produce the high fraction factor losses increase of 98.42 % over the smooth duct as compression with other angles range in current study.
4. The TEF tends to reduce with the rising in Reynolds number for all angles value . It is interesting to note that the TEF tends to increase with reduce in ribs angle. And seen that the ribs at angle 10° gives the highest TEF at lower Re number and is about 1.297

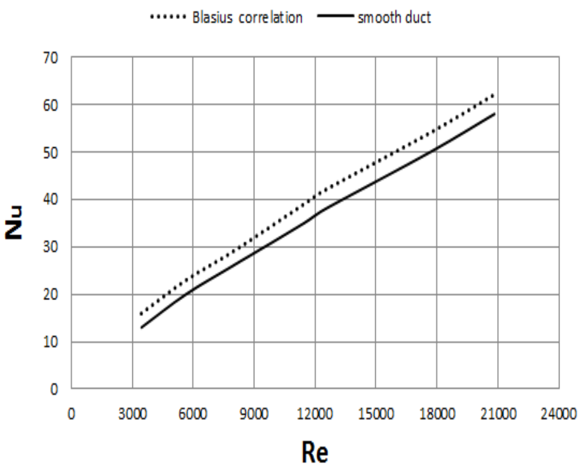


Figure (5): Verification of Nu verse Re for smooth channel.

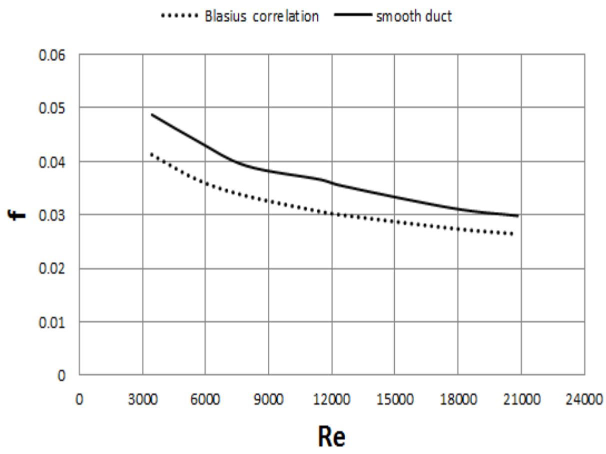


Figure (6): Verification of f verse Re for smooth channel.

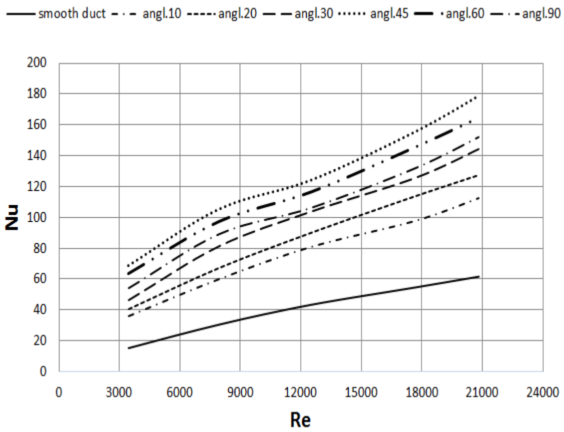


Figure (7): The variation of Nu verse Re with ribs at different angle.

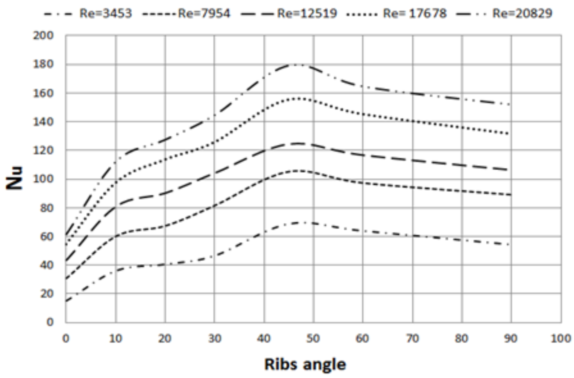


Figure (8): The variation of Nu verse ribs angle at different Re

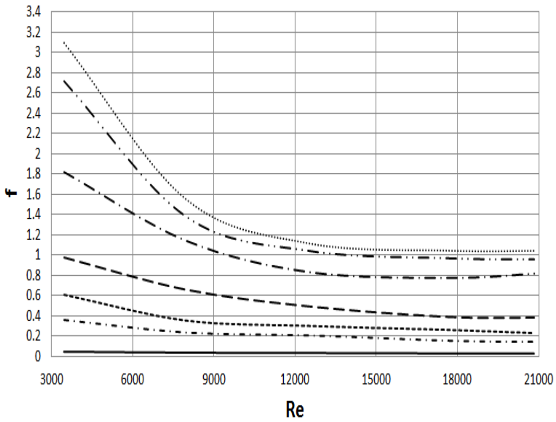


Figure (9): The variation of f verse Re with ribs at different angle.

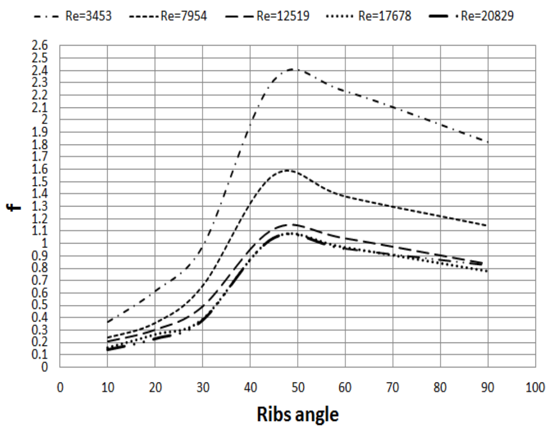


Figure (10): The variation of f verse ribs angle at different Re.

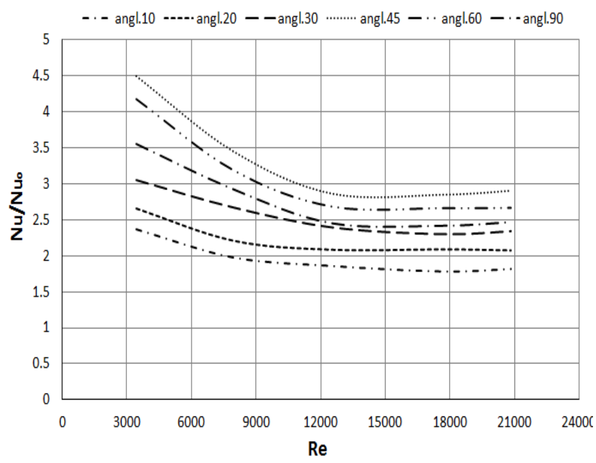


Figure (11): The variation of Nu/Nu_0 verse Re with ribs at different angle.

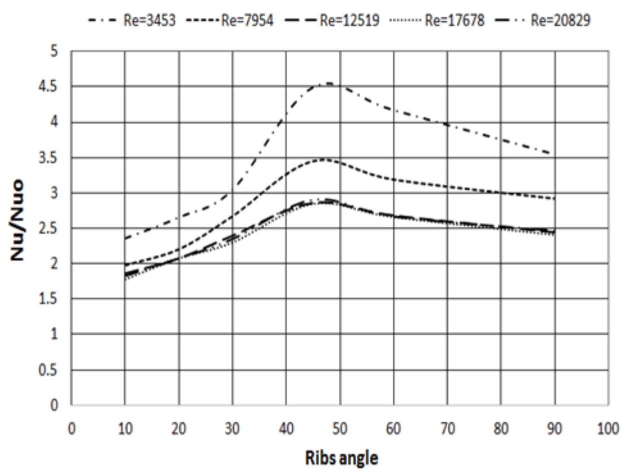


Figure (12): The variation of Nu/Nu_0 verse ribs angle at different Re.

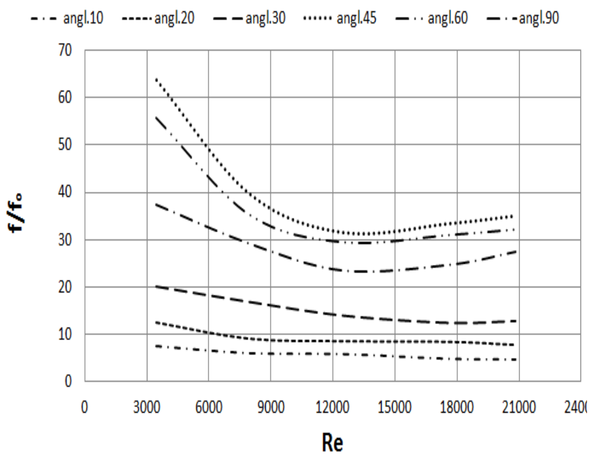


Figure (13): The variation of f/f_0 verse Re with ribs at different angle.

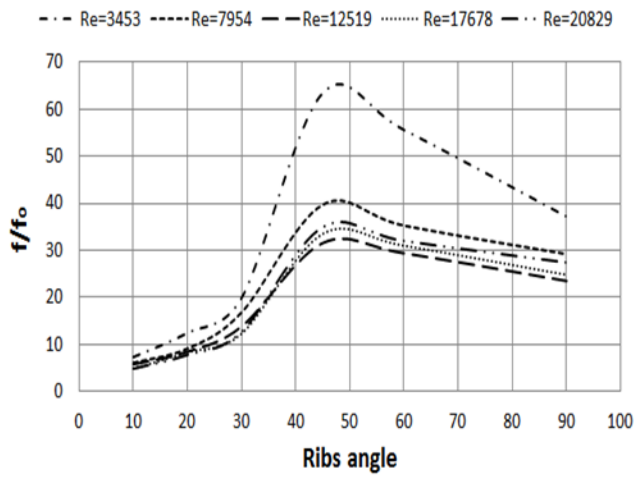


Figure (14): The variation of f/f_0 verse ribs angle at different Re.

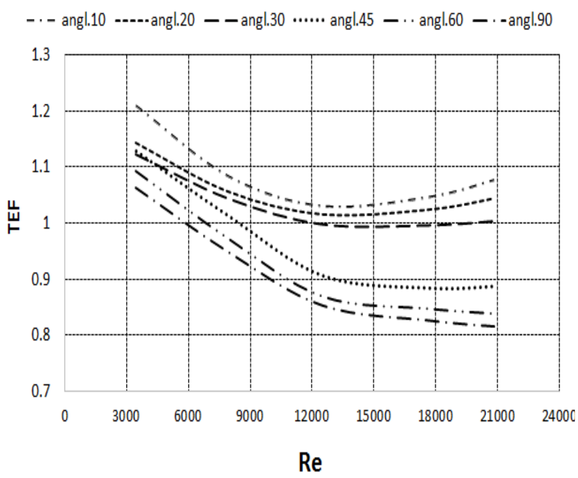


Figure (15): The variation of TEF verse Re with ribs at different ribs angle

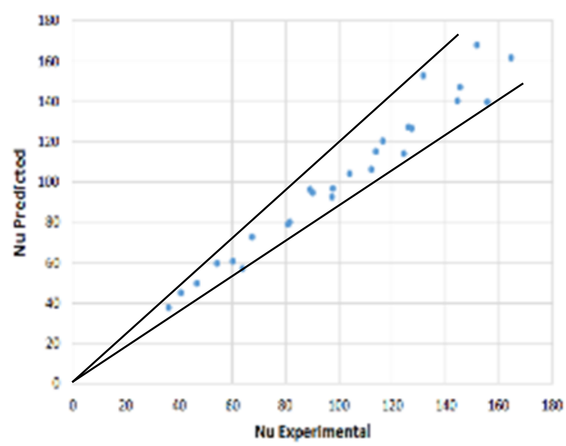


Figure (16): Comparison between predicted and experimental Nu

Abbreviations

A_s convection heat transfer area of duct, m^2
AR aspect ratio of duct, (W/H)
b rib height, m
BR rib blockage ratio, (b/H)
D_h hydraulic diameter of duct, (=H), mm
e rib height, m
f friction factor
H duct height, m
h average heat transfer coefficient, W/m^2K
I current, A
$\dot{m}$ mass flow rate Kg/s
k thermal conductivity of air, W/mK
L length of test duct, m
Nu Nusselt number, (hD_h/K_a)
P rib pitch spacing (axial length of spacing), m
Δp pressure drop, Pa
PR rib pitch to duct height ratio, (P/H)
Re Reynolds number, (UD_h/ν)
Q heat transfer, W
T temperature, K
TEF thermal performance enhancement factor
U average velocity, m/s
V voltage, V
W width of duct
Greek letters
α attack or inclination angle of rib, °
ρ density of air, kg/m3
υ kinematics viscosity, m^2/s

subscribts

b bulk
0 smooth duct
conv. convection
i inlet
o out
pp pumping power
s duct surface

5. References

1. Pongjet Promvonge, Teerapat Chompookham, Sutapat Kwankaomeng, Chinaruk Thianpong (2010) " Enhanced heat transfer in a triangular ribbed channel with longitudinal vortex generators" Energy Conversion and Management 51 1242–1249.
2. Sutapat Kwankaomeng, Sompol Skullong, Thavee Teschareon and Pongjet Promvonge (2010) "Thermal Characteristics in Square Channel with 45° Staggered Baffle inserts", PEA-AIT International Conference on Energy and Sustainable Development: Issues and Strategies (ESD 2010) The Empress Hotel, Chiang Mai, Thailand. 2-4 June.
3. Pongjet Promvonge, Teerapat Chompookham, Sutapat Kwankaomeng, Chinaruk Thianpong (2010) "Enhanced heat transfer in a triangular ribbed channel with longitudinal vortex generators" Energy Conversion and Management 51 1242–1249.
4. Giovanni Tanda (2011)"Effect of rib spacing on heat transfer and friction in a rectangular channel with 45° angled rib turbulators on one/two walls" International Journal of heat transfer and Mass transfer, Volume 54, Issues 5–6, February, Pages 1081-1090.
5. P. Promvonge, S. Sripattanapipat, S. Tamna, S. Kwankaomeng, C. Thianpong (2010) "Numerical investigation of laminar heat transfer in a square channel with 45° inclined baffles" International Communications in Heat and Mass Transfer 37 170–177.
6. Pongjet Promvonge*, Somchai Sripattanapipat, Sutapat Kwankaomeng (2010) "Laminar periodic flow and heat transfer in square channel with 45-inline baffles on two opposite walls" International Journal of Thermal Sciences 49 963-975.
7. Pongjet Promvonge, Sompol Skullong, Sutapat Kwankaomeng, Chinaruk Thiangpong (2012) "Heat transfer in square duct fitted diagonally with angle-finned tape-Part 1: Experimental study" International Communications in Heat and Mass Transfer 39 617–624.
8. K. Yongsiri, P.Eiamsa-ard, K.Wongcharee, S.Eiamsa-ard (2014)"Augmented heat transfer in a turbulent channel flow with inclined detached-ribs" Case Studies in Thermal Engineering 31–10.
9. F. Incropera and P.D.Dewitt (2006), Introduction to heat transfer, 5th edition John Wiley &Sons Inc.,
10. Fluid Flow in Closed Conduits, “British Standard Methods, Part 1 ”, (1981) Pressure Differential Devices, BS No.1042: Section 1.1.
11. Thamir Kh. Ali (2012) " Experimental and Theoretical Study of Inclined Air Jet Impingement on One Side of A Narrow Duct," M.S.C Thesis, University of Al-Mustansiriya, Department of Mechanical Engineering.