Experimental Investigation Of Thermal And Hydraulic Enhancement Convection Heat Transfer Through V And Trapezoidal Corrugated Channel

Lect. Dr. Damiaa Saad Khudor
Mechanical Engineering Dept., College of Eng.
Al-Mustansiriya University, Baghdad, Iraq
damiaasaad@yahoo.com

Abstract:

A steady developing fluid flow and heat transfer through a wavy corrugated square cross-sectional passage (V-shaped and trapezoidal) was experimentally studied for a fluid with a Prandlt number of 0.7. In the present study, the experimental investigation on the heat transfer and flow distributions in the channel with various geometry configuration wavy plates under constant heat flux conditions of 600 W/m² and 1100 W/m² is considered. Effects of geometry configuration of wavy plates, wavy plate arrangements, and air flow rates on the temperature and flow developments are considered. Reynolds number is range from (2.19 - 3.3) $\times 10^3$. The results indicated that the enhancement of heat transfer coefficient factors are obtained from the wavy plate structures were essentially independent of the operated heat flux. The enhancement of heat transfer coefficient was ranged between the (1.036 to 1.062) and (1.042 to 1.066) for heat fluxes (600) W/m² and (1100) W/m² respectively for the V-shaped. While the enhancement of heat transfer coefficient for trapezoidal it was ranged between the (1.052 to 1.077) and (1.058 to 1.082) for heat fluxes (600) W/m² and (1100) W/m², respectively. The corrugated channel pressure drop was higher than the plain smooth surface. The deterioration percentages of pressure drop for the corrugated plate when compared with that of flat plain surface was ranged between (15.4%) and (24.7%) for the V-shaped and (18.2%) and (27.5%) for trapezoidal with heat fluxes of 600 W/m² and 1100 W/m², respectively.

تحقيق عملي للتحسين الحراري والهيدروليكي لانتقال الحراره بالحمل خلال قنال متموج ذو شكل V وشبه منحرف

الخلاصه

تمت دراسة تطور انسياب المائع وانتقال الحرارة خلال قنال متموج ذو مقطع مربع (شكل V وشبه منحرف) كانت هذه الدراسة دراسة عمليه للمائع ذو رقم (براندلت)0.7 . في هذه الدراسه، تم الاخذ بنظر الاعتبار التحقيق التجريبي لانتقال الحراره وانسياب المائع خلال قنال مع وجود اشكال متغيره للسطوح المتعرجه بوجود فيض حراري ثابت 600 W/m² و1100 W/m². تم الاخذ بنظر الاعتبار تاثير شكل السطوح المتعرجه وترتيبها وتغيير معدل جريان المائع على توزيع السرع وتطور جريان المائع.

رقم رنولد يتراوح بين $10^3\times$ (2.19 - 3.3). ان النتائج العمليه اعطت انطباع لتحسين معدل انتقال الحراره من السطوح المتعرجه بوجود الفيض الحراري. تحسين معدل انتقال الحراره يتراوح بين (1.036 to 1.062) و (1.042 to 1.066) لفيض حراري ثابت 600 W/m² و1100 W/m² على الترتيب مربع (شكل V). بينما تحسين معدل انتقال الحراره للشكل شبه المنحرف كانت تتراوح بين (1.052 to 1.077) و (1.058 to 1.082) لفيض حراري ثابت 600 W/m² و 1100 W/m² على الترتيب . ان هبوط الضغط خلال القنال المتعرجه كانت اعلى من السطح الاملس. ان النسب المئويه لهبوط الضغط خلال السطوح المتعرجه ومقارنتها مع السطح الاملس للقنال كانت تتراوح بين (15.4%) و (24.7%) للشكل (شكل V) وللشكل شبه المنحرف كانت (18.2%) و (27.5%) لفيض حراري ثابت 600 W/m² و1100 W/m² على الترتيب.

Keywords: Forced Convection, Enhancement, Hydraulic, Corrugated channel

1- Introduction

To improve the performance of plate heat exchangers, efficient heat transfer surfaces which do not induce much pressure loss and flow obstruction must be implemented. The wavy geometries are known to enhance the heat transfer by breaking and destabilizing the thermal boundary layer. Therefore, corrugated surfaces act as turbulence promoters to increase the local heat and mass transfer throughout the flow zone.

Abed and Ahmed (2010) [1] studied numerically laminar forced convection heat transfer and fluid flow characteristics in V-corrugated channel with corrugated angles from 0° to 60° , and Reynolds number range from 500 to 2500. The results showed that heat transfer and pressure drop increase with increasing wavy angle at same Reynolds numbers. The most important finding of this work was that the optimum values of the heat transfer enhancement and pressure drop are (3.6) and (1.11) times higher than those from the plane channel at wavy angle (40º), respectively. **Promvonge and Eiamsa (2007) [2]** studied the influence of V-nozzle turbulator inserts in conjunction with a snail entry on heat transfer and friction loss characteristics in a circular duct through which air as the test fluid is passed. For Reynolds number ranging from 8000 to 18,000, the results revealed that the use of Pitch ratio (2) leads to higher Nusselt number and friction factor values than that of Pitch ratio (4 or 7).

Naphon (2009) [3] postulated a numerical investigation on the heat transfer and flow distributions in the channel with various geometry configuration wavy plates under constant heat flux conditions. The results showed the vital importance of the wavy plate geometry configuration for efficient design heat exchanger to enhance thermal performance. **Fabbri (2000) [4]** studied numerically the heat transfer and fluid flow characteristics in a channel composed of a smooth and a corrugated wall under laminar conditions. The results demonstrated that, when no constraint is imposed either on the wall volume or the pressure drop, a corrugated wall profile only maximizes the heat transfer when both the Reynolds and Prandtl numbers are not too low. **M. M. Awad & Y. S. Muzychka (2011) [5]** A detailed review and analysis of the thermal-hydrodynamic characteristics in air-cooled compact wavy fin heat exchangers was studied. New models are proposed which simplify the prediction of the Fanning friction factor f and the Colburn j factor. These new models are developed by combining the asymptotic behavior for the low Reynolds number and laminar boundary layer regions. In these two regions, the models are developed by taking into account the geometric variables such as: fin height, fin spacing, wave amplitude, fin wavelength, Reynolds number, and Prandtl number. Fanning friction factor *f* and Colburn *j* factor in air cooled compact wavy fin heat exchangers at different values of the geometric variables obtained from the published literature were presented to show features of the asymptotes, asymptotic analysis and the development of the simple compact models. **Bahaidarah (2009) [6]** developing a numerical studied for a two-dimensional steady fluid flow and heat transfer through a periodic wavy passage with a Prandlt number of 0.7. Identified horizontal pitch to module length ratio and Reynolds number as the independent parameters. Four different types of wavy geometry, triangular without horizontal pitch and triangular horizontal pitch were considered. Results were compared with those for corresponding triangular wavy channel. Triangular wavy channel without horizontal pitch provide lower normalized pressure drop values when compared to triangular wavy channel with horizontal pitch and it keep increasing as the horizontal pitch increases. The module average Nusselt number monotonically increases as Reynolds number increases. **Maryam Mirzaei, eet al. (2014) [7]** Large Eddy Simulation (LES) of turbulent flow and convective heat transfer over a half-corrugated channel was studied. Simulations are performed for various ranges of the normalized wave amplitudes, AM = 0 − 0.15 (the ratio of wave height to wave length). The Reynolds number based on the bulk velocity is chosen as Re_b = 10 000 and the Prandtl number is Pr=0.71. A comparison between the Direct Numerical Solution (DNS) and LES results of a plane channel (AM = 0) at Re_ = 395 is also performed. The obtained results indicate that the region of recirculating flow depends strongly on the wave amplitude. This study shows that the Nusselt number (Nu) increases by increasing the wave amplitude until a specific value then it remains approximately constant. The thermal performance parameter (JF) is used as a measure for the heat transfer enhancement relative to the pressure drop and it is found that the maximum values of Nu and JF appear at AM = 0.1, which hence correspond to the optimum value of the wave amplitude. **Mohammad Mehdi Rashidi , et al. (2016) [8]** the effect of phase shift between the upper and lower wavy wall on the laminar flow and heat transfer characteristics of a copper-water nano-fluid was numerically investigated. The governing equations associated with the required boundary conditions were solved using finite volume method based on the SIMPLE technique. The predicted results for average and local Nusselt number along the wall had great agreement with the published data. Moreover, the effects of phase shift, wavy amplitude, Reynolds number and nano-fluid volume fraction on the local and average Nusselt numbers were declared in this research. Results showed that the heat transfer increases with the increase in the wavy amplitude, Reynolds number and nano-fluid volume fraction.

In the present study an experimental investigation on the heat transfer and hydrodynamic aspects in a channel with (V-shaped and trapezoidal) wavy corrugated is considered. It is important to point out that the main objective of the present investigation is to figure out the best corrugation geometry that possesses the most attractive practical performance at low Reynolds number turbulent zone. Accordingly, there was no consideration given for the implementation of high heat fluxes in these tests. The enhancement of heat transfer and deterioration of pressure drop due to the implementation of corrugated plates in a narrow zone of Reynolds number were considered. Four different geometry configurations of wavy plates were tested at Reynolds number range of (2.19 - 3.3) $\times 10^3$ for corrugate channel at constant heat flux conditions of (600), and (1100) W/m².

2-Experimental apparatus and method

2-1 General

A schematic diagram of the experimental setup is illustrated in Fig. 1. The test rig consists of entrance section, a test section, orifice and outlet air portion. The air stream flows from a 5 kW blower and then passed through a calming section duct. The air is passed to the entrance section through calming section comprises of a conical section having 75° angle with horizontal. This was aimed to damp the turbulence in flow and helps smoothing of pressure fluctuations in the inlet air, as shown in Fig. 1. This measure was taken to assure a stability of the air stream. Then it passes through the heat transfer test section, orifice meter to measure the flow rate, and then discharged to the atmosphere.

2-2 Test Section

The test section is the corrugated channel with two opposite (V-shaped and trapezoidal) corrugated plates on both of the upper and lower surfaces of the duct. The air duct is a square cross section having dimensions of (10 × 10) cm and a length of 100 cm as shown in Fig. 1. The duct is insulated with coated fiberglass standard sheet to minimize the heat loss from shell to the ambient. The wavy corrugated plate metal was made of aluminum with thickness (0.5mm) and surface area (0.224 m²); it's with four geometry is illustrated in table 1. Figure (2-a) shows the geometrical configuration of triangular wavy channel with two corrugated angle (32º and 45º). Figure (2-b) shows the geometrical configuration of trapezoidal wavy plate with two corrugated angle (47.5º and 63.4º). The corrugated sheet was fitted tightly with the wall by compression to prevent its movement and air leakage.

2-3 Instrumentation

2-3-1 Air Flow Rate

The air flow rate to the test rig was measured by an orifice designed according to B.S.1042, with (1.23) cm diameter, as shown in Fig. 3. The orifice meter was calibrated using a pitot tube with an accuracy of 2% of full scale are employed to measure the air volumetric. There are two pressure taps on each wall upstream and downstream of the test section and the orifice plate. Carbon tetra-chloride is the working fluid in the U-duct manometer used to measure the pressure drop across the orifice meter with specific gravity of 1.588. This measure was taken to ensure a reasonable accurate measurement of the low pressure drop encountered at low Reynolds numbers. The volumetric air flow rates from the blower were adjusted by varying motor speed through an inverter, situated before the inlet of test duct. Also, the pressure drops across the heat transfer test section were measured with two pressure gauges with an accuracy of (±0.02) of full scale are employed to measure air pressure.

2-3-2 Temperature

Six thermocouples were situated along the test section wall surface, embedded in wavy groove duct surfaces. The mean wall temperature was determined by means of a calculation based on the readings of the K-type thermocouples. It was necessary to measure the temperature at 7 stations at the surface of the heat transfer test section to find out the average Nusselt number. The K-type copper-constantan thermocouples with an accuracy of 0.1% of full scale are employed to measure air temperature. The surface temperature was measured at selected points with a multi-channel temperature measurement unit in conjunction, Fig. 1. In the apparatus setting above, the inlet air at 27 Cº from a blower and passed to the heat transfer test section.

2-3-3 Power Supply

The corrugated sheet was heated by continually winding flexible electrical wire provided a uniform heat flux boundary condition. The electrical output power was controlled by a Variac transformer to obtain a constant heat flux along the entire length of the test section and keeps the current to be less than 3 Amperes. This was a standard limitation from the manufacture of the heating wire..

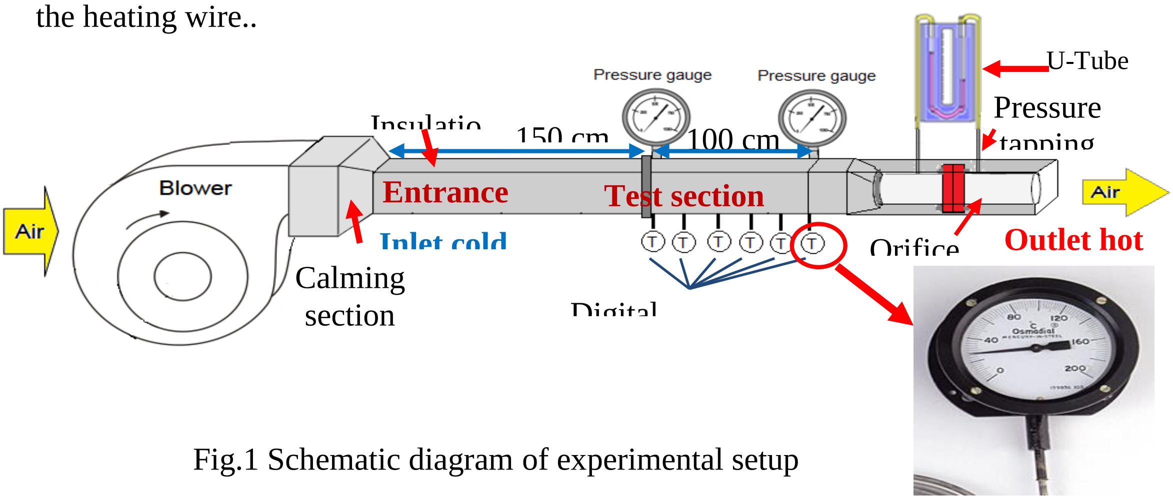


Fig.1 Schematic diagram of experimental setup

V- shaped dimension								
Case	L (mm)		Hmax (mm)	Hmin (mm)	N	α (°)	P (mm)	a (mm)
1	32		100	80	31	32°	32	10
2	20		100	80	50	45°	20	10
	Trapezoidal corrugated dimension							
Case	L (mm)	ℓ (mm)	Hmax (mm)	Hmin (mm)	N	α (°)	P (mm)	a (mm)
3	28	6	100	76	31	47.5°	34	12
4	16	4	100	76	50	63.4°	20	12

T

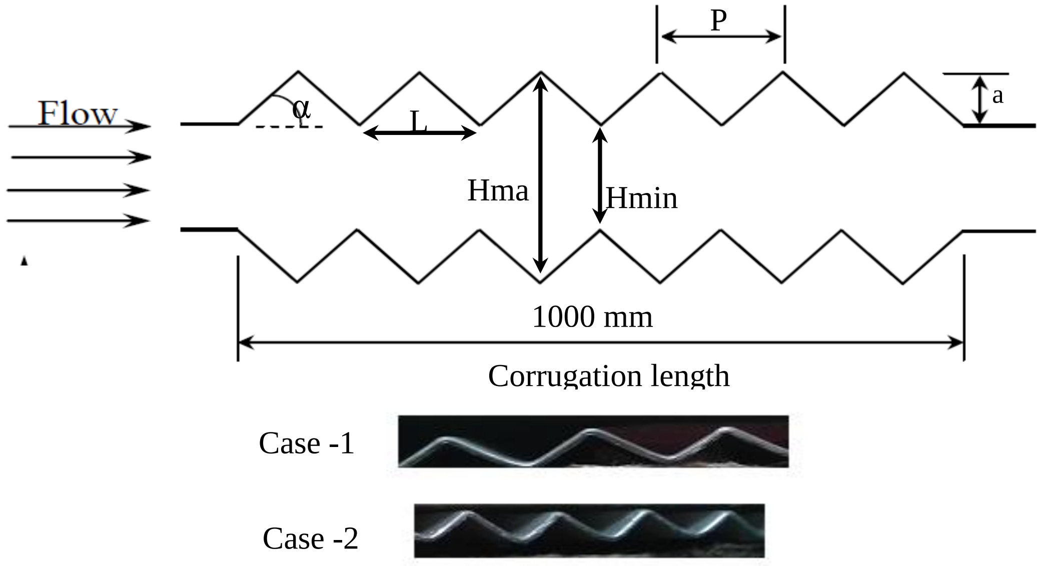


Fig.2-a V- shaped corrugated sheet dimensions

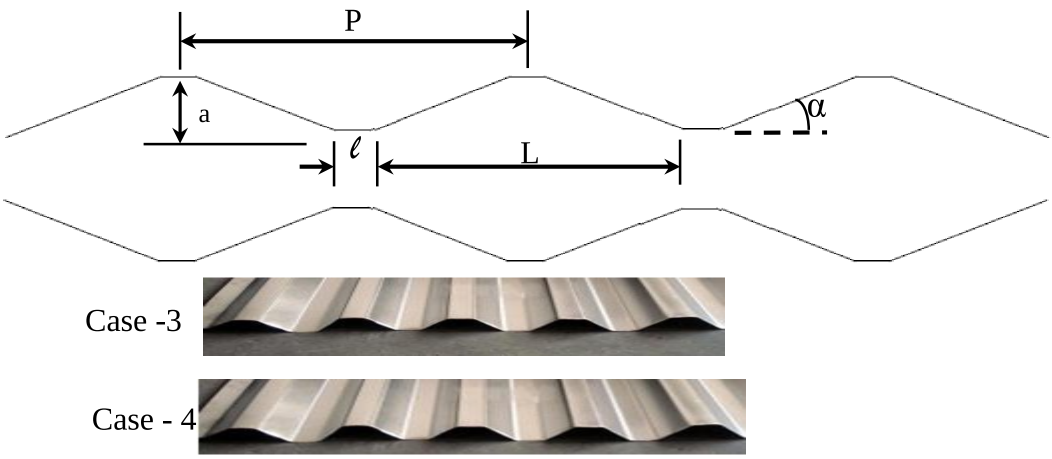
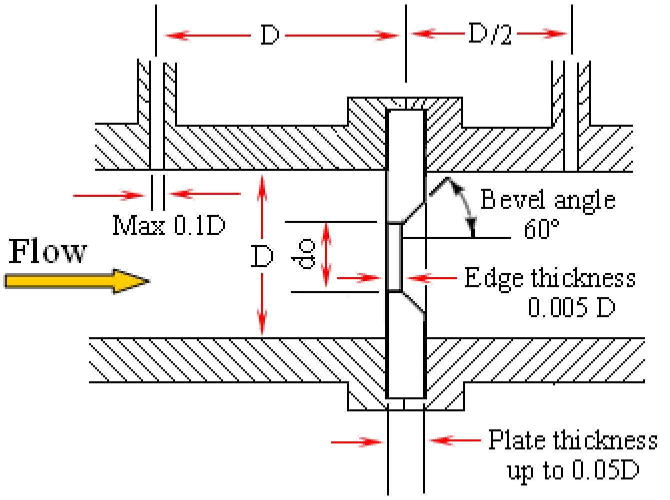


Fig.2-b Trapezoidal corrugated sheet dimensions



Symbol	Dimension mm
D	41
d_o	12.3

Fig.3 A schematic diagram for orifice meter design

2-4 Experimental procedure:

At each test the operation conditions such as temperature, volumetric flow rate and pressure drop of the bulk air at steady state conditions were recorded. During the experimental category of the present work, the inlet air dry bulb temperature was maintained at 27Cº. The various characteristics of the flow, the Nusselt number, and the Reynolds numbers were based on the average of duct wall temperature, inlet and outlet air temperatures. The local wall temperature, inlet and outlet air temperature, the pressure drop across the test section, heat flux and air flow velocity were also measured and recorded. Noting that all of the fluid properties were determined at the overall bulk mean temperature as shown in table 2.

Table 2. Air Properties

Air Properties at 27 ˚C				
Density - ρ - (kg/m^3)	Specific Heat - c_p - $(kJ/(kg\ K))$	Thermal Conductivity - k - $(W/(m\ K))$	Kinematic Viscosity - ν - $x\ 10^{-6}\ (m^2/s)$	Prandtl's Number - P_r -
1.1777	1.005	0.02619	15.761	0.71

3- Data Reduction:

The experimental calculations were presented in terms of pressure drop, heat flux, Reynolds number and Nusselt number:

3-1. Fluid Flow

3-1-1 Reynolds Number:

Reynolds number can be defied as the ratio of inertial forces to viscous forces for the fluid flow, which is expressed as follows:

$$\mathrm{Re} = \frac{\rho_a . u_a . D_h}{\mu_a} \quad (1)$$

$$D_h = \frac{4A_{cross}}{p} \quad (2)$$

For the present work the equivalent hydraulic diameter for the corrugated plate was calculated to be (9.64) cm.

3-1-2 Air flow rate

The volumetric flow rate Q_a in terms of the head differential across the orifice plate Δh, was deduced from the well-known equation presented by **Frank (2003) [10]**:

$$\dot{V} = C_d S_o \sqrt{\frac{2g\Delta h_o}{1-\beta^4}} \tag{3}$$

$$\dot{m}_a = \rho \dot{V} \tag{4}$$

Where: $C_d = 0.72$, $S_o = \frac{\pi}{4} d_o^{\;2}$, $S_o = 1.18822 \times 10^{-4} m^2$

3-2. Heat Transfer Mode:

At steady state, it was assumed that the amount of heat transfer rate from the heating element to the cooling medium to be:

$$\dot{Q}_{av} = \dot{Q}_{conv}$$

Heat transferred to air flowing through the test section, Q_a, is determined from

$$\dot{Q}_a = \dot{m}_a \, Cp_a (T_{ao} - T_{ai}) \tag{5}$$

Heat added to the top and bottom corrugated plate is calculated from measuring voltage and current supplied by electrical winding in the test section as follows:

$$\dot{Q}_{heater} = I.V$$

Neglected the radiation heat transfer due to small value, then the average heat transfer rate, Q_{ave}, used in the calculation is determined from the heat removal by the air stream and the heat supplied to the corrugated plate as follows:

$$\dot{Q}_{av} = \frac{\dot{Q}_a + \dot{Q}_{heater}}{2} \tag{6}$$

The convection heat transfer from the test section can be written by:

$$\dot{Q}_{conv.} = hA(T_w - T_{ab}) \tag{7}$$

In which T_w is the local wall temperature and evaluated at the outer wall surface of the inner duct, thermocouples embedded in corrugated outer surfaces. The thermal resistance of the duct wall can be neglected and therefore the measured T_w can be approximated to be the same as the inner duct wall surface temperature.

The heat transfer coefficient, h and the average Nusselt number, Nu are estimated from the temperature difference between the inlet and outlet of two points as follows, where all thermal properties of the tested fluid are determined at the bulk air temperature:

$$h = \frac{\dot{m}_a\, Cp_a (T_{ao} - T_{ai})}{A(T_w - T_{ab})} \tag{8}$$

$$T_{ab} = \frac{T_{ao} + T_{ai}}{2} \tag{9}$$

$$Nu = \frac{h\, D_h}{K} \tag{10}$$

The hydraulic diameter, D_h, for the duct as a flat plate or corrugated surface was used for the calculation of Nusselt and Reynolds numbers.

3-3Enhancement Evaluation:

A comparison between heat transfer coefficient and Nusselt number of the wavy and p lain plate flows at equal supplied power and heat flux can be made. The respective enhancement factors of mean heat transfer coefficient and Nusselt number at the same heat flux are:

$$h_{Enh,m} = \frac{h_{Corrugated,m}}{h_{Plain,m}} \tag{11}$$

$$Nu_{Enh,m} = \frac{Nu_{Corrugated,m}}{Nu_{Plain,m}} \tag{12}$$

The enhancement percentages for the heat transfer coefficient and Nusselt number of wavy plate at the same heat flux can be written as:

$$\eta_{enh,h} = \frac{h_{corrugated} - h_{Plain}}{h_{Plain}} \times 100\% \tag{13}$$

$$\eta_{enh,Nu} = \frac{Nu_{corrugated} - Nu_{Plain}}{Nu_{Plain}} \times 100\ \% \tag{14}$$

The pressure drop deterioration, Δp_{Det} at constant pumping power and heat flux is the ratio of the mean pressure drop of the duct with corrugated plate to the flat plain duct which can be written as follows:

$$\Delta p_{Det} = \frac{\Delta p_{corrugated}}{\Delta p_{Plain}} \tag{15}$$

The deterioration percentage of pressure drop for the corrugated plate when compared with that of flat plain surfaces is expressed as:

$$\eta_{Det,p} = \frac{\Delta p_{corrugated} - \Delta p_{Plain}}{\Delta p_{Plain}} \times 100\% \tag{16}$$

4- Experimental Results:

Experimental results achieved for Reynolds number ranged (2.19 - 3.3) $\times 10^3$, the corresponding air velocity range was (1.1 to 1.4) m/s. The tests were carried out at constant Prandtl number of 0.71. The main stream air velocity range was (1.1 to 1.4) m/s.

Figure 4 shows the variation of enhancement for the mean heat transfer coefficient, $h_{Enh,m}$ with Reynolds number for V-shaped and trapezoidal corrugated channel. The onset and growth of recirculation zones promote the mixing of fluid in the boundary layer thereby enhancing the convective heat transfer. The duct with a corrugated sheet provides heat transfer coefficient enhancement percentages of the (1.036 to 1.062) and (1.042 to 1.066) for heat fluxes (600) W/m² and (1100) W/m² respectively for the V-shaped. While for trapezoidal it was ranged between the (1.052 to 1.077) and (1.058 to 1.082) for heat fluxes (600) W/m² and (1100) W/m², respectively depending on air Reynolds number. This can be attributed to better and fast mixing of the reverse flow from wavy corrugated and larger surface area.

Figure 5 shows the mean heat transfer coefficient with Reynolds number at heat fluxes (600) W/m² and (1100) W/m² for four cases. It can be seen that the heat transfer coefficient increased with Reynolds number where main flow deflects into the corrugated confined area where the vortices are created vortex grows larger and its center moves to the downstream as Reynolds number increases, and occupies the confined area. The vortices in the confined corrugated area repeat the propagation that they travel to the downstream simultaneously in the lower and upper plates.

Figure 6 and 7 shows a comparison for the enhancement for $Nu_{Enh,m}$ and the mean Nusselt numbers with Reynolds number at heat fluxes (600) W/m² and (1100) W/m². The duct with a corrugated sheet provides Nusselt numbers enhancement of the (6%) and (8%) for heat fluxes (600) W/m² and (1100) W/m² respectively for the V-shaped. While for trapezoidal it was ranged between the (7.8%) and (9.4%) for heat fluxes (600) W/m² and (1100) W/m², respectively depending on air Reynolds number

Nusselt number for fluid flowing through the corrugated surface revealed a higher value than that of smooth test section. This was due to fluid re-circulation or/and swirl flows are generated in the corrugation duct. The onset and growth of recirculation zones promote the mixing of fluid in the boundary layer thereby enhancing the convective heat transfer.

Figure 8 shows the variation of the average plate temperature with air Reynolds number for constant heat flux (600) W/m² for the V-shaped. The heat transfer rate depends on the cooling capacity rate of air. Therefore, the average plate temperature decreases as air Reynolds number increases. Figure 8 shows the variation of the local plate surface temperature distribution along the test section at constant heat flux with air velocity range was (1.1 to 1.4) m/s. It is obvious that the local plate surface temperature decreases in the flow direction. Figure 9 shows a comparison for the variation of pressure drop deterioration Δp_{Det} with Reynolds number for the test section V-shaped and trapezoidal corrugated channel. The pressure drop across the channel with corrugated surface is higher than plain duct. This can be explained by the fact that drag forces exerted on the flow field by the corrugated surface, turbulence augmentation, and rotational flow produced by the corrugated surface.

It is interesting to compare the (ΔP_{Det}) for the test surfaces as shown in figure (9) at (600) W/m^2 (1100) W/m^2 heat flux. The higher the corrugation angle the higher deterioration would be. Accordingly, at (63.4°) with trapezoidal the maximum deterioration was experienced to be (1.64) at Reynolds number of (2200) and (1.51) at Reynolds number (3338.02). The results showed that the deterioration of the pressure drop is directly proportional to the heat flux, increases as it increases and vise versa. The behavior of these curves shows that as the Reynolds number increases, the deterioration factor exhibits a gradual decrease and the opposite is true.

The deterioration percentage of pressure drop for the corrugated plate when compared with that of flat plain surfaces shown in figure (10) which ranged between (15.4%) to (23.7%) for the tests heat fluxes with V-shaped while (21.7%) to (29.4%) for the tests heat fluxes with trapezoidal corrugated channel.

5- Conclusion

The following findings can be deduced from the present work:

1. The duct with a corrugated sheet provides heat transfer coefficient enhancement percentages of the (1.036 to 1.062) and (1.042 to 1.066) for heat fluxes (600) W/m² and (1100) W/m² respectively for the V-shaped. While for trapezoidal it was ranged between the (1.052 to 1.077) and (1.058 to 1.082) for heat fluxes (600) W/m² and (1100) W/m², respectively depending on air Reynolds number.
2. The enhancement percentages for the Nusselt number of wavy plate at the same heat flux are 6%) and (8%) for heat fluxes (600) W/m² and (1100) W/m² respectively for the V-shaped. While for trapezoidal it was ranged between the (7.8%) and (9.4%) for heat fluxes (600) W/m² and (1100) W/m², respectively.
3. The deterioration percentage of pressure drop for the corrugated plate when compared with that of flat plain surfaces was ranged between (15.4%) to (23.7%) for the tests heat fluxes with V-shaped while (21.7%) to (29.4%) for the tests heat fluxes with trapezoidal corrugated channel.

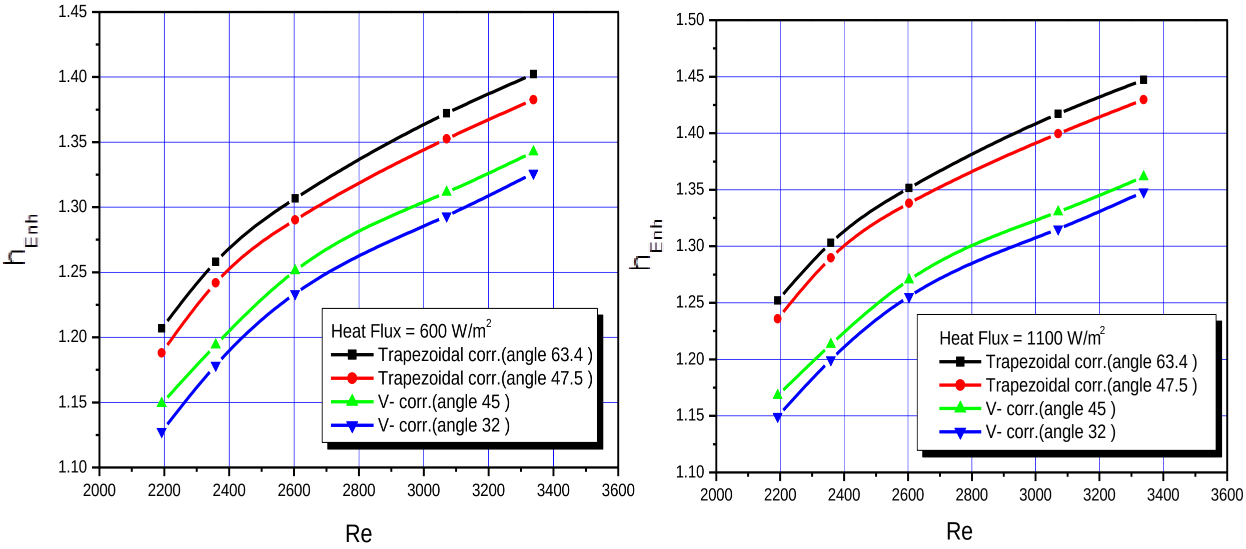


Fig.4 The variation of enhancement heat transfer coefficient factor, $h_{Enh,m}$ with Reynolds number for both test heat fluxes.

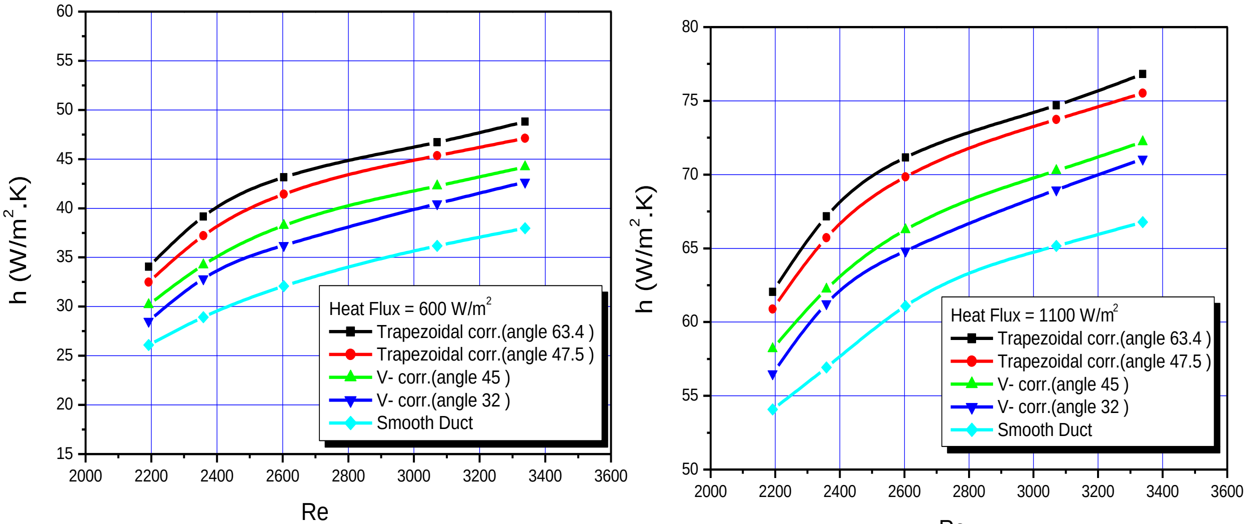


Fig.5 The variation of heat transfer coefficient, h with Reynolds number for both test heat fluxes.

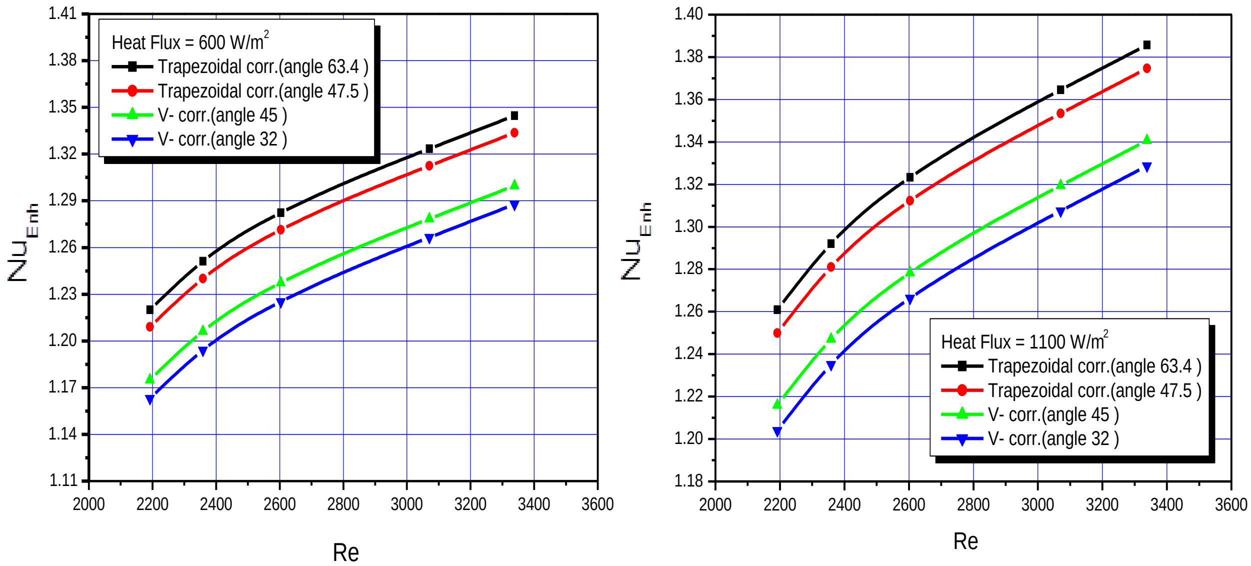


Fig.6 A comparison of the variation of enhancement of $Nu_{Enh,m}$ with Reynolds number at both of the test heat fluxes.

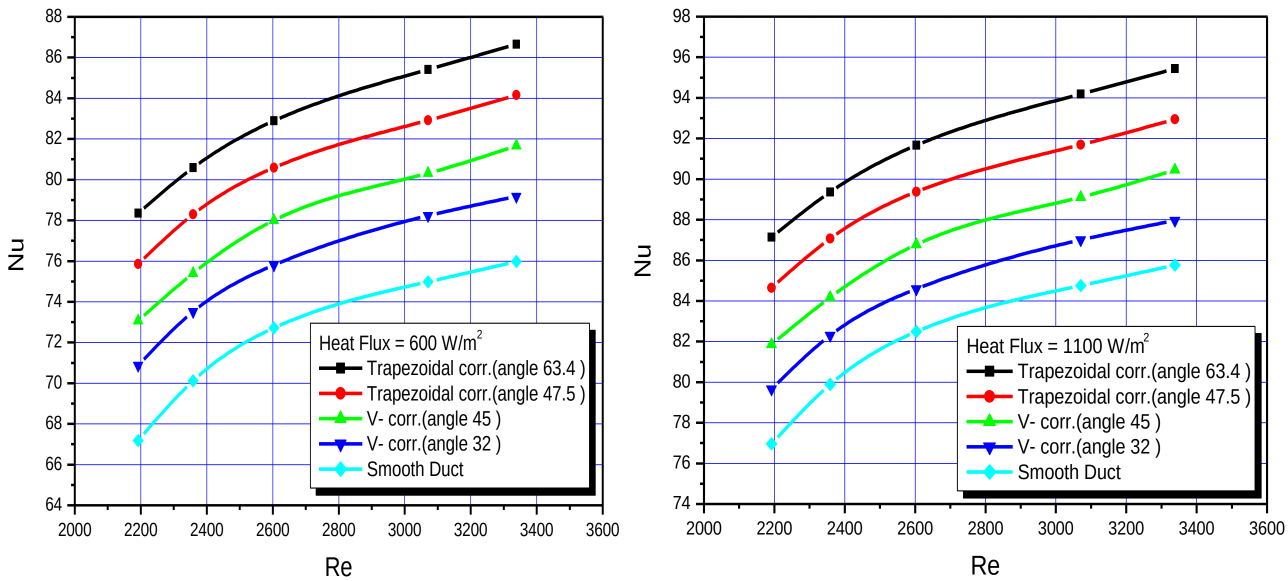


Fig.7 A comparison of the variation of enhancement of the mean Nusselt number with Reynolds number at both of the test heat fluxes.

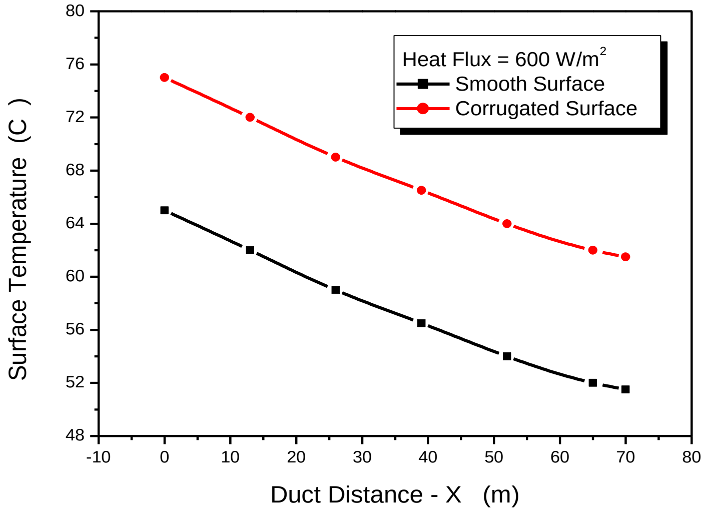


Figure (8): A comparison of the local surface temperature variation with the test section length in the flow direction for both of the V-shaped corrugated and smooth surface at the same heat flux (U=1.1 m/s)

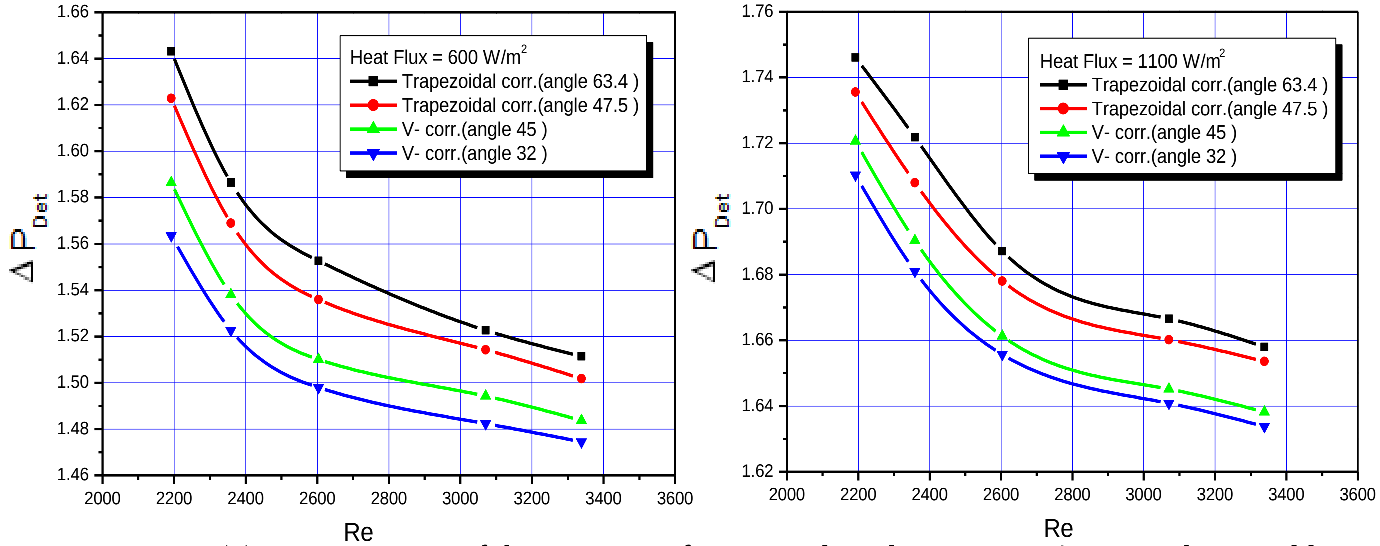


Figure (9): A comparison of the variation of pressure drop deterioration Δp_{Det} with Reynolds number of the test sections at different heat fluxes.

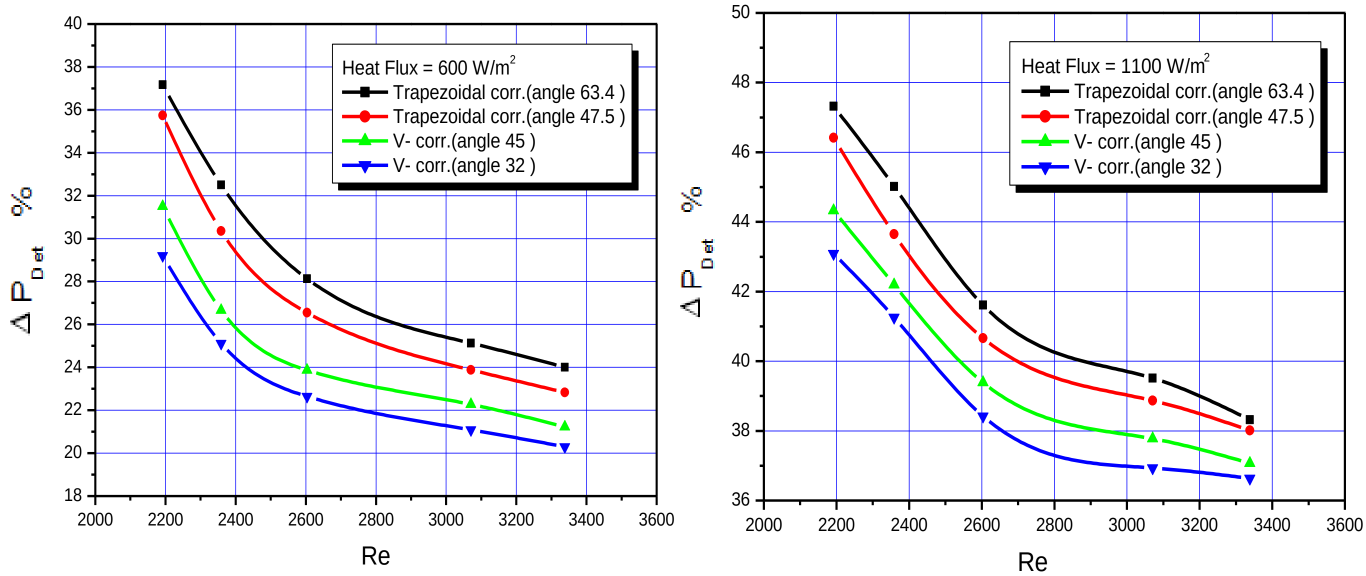


Figure (10): A comparison of the variation of percentages pressure drop deterioration $\Delta p_{Det}\%$ with Reynolds number of the test sections at different heat fluxes.

References

[1] Abed, W.M., and Ahmed, M.A.,*" Numerical Study of Laminar Forced Convection Heat Transfer and Fluid Flow Characteristics in A Corrugated Channel"*, Journal of Engineering and Development, Vol.14, No. 3, September, pp. 70-85, 2010.
[2] Promvonge, P., and Eiamsa, S., *"Heat Transfer Augmentation in A Circular Duct Using V-nozzle Turbulator Inserts and Snail Entry"*, in Experimental Thermal and Fluid Science 32, pp. 332–340, 2007.
[3] Naphon,P**.,** *" Effect of wavy plate geometry configurations on the temperature and flow distributions"*, International Communications in Heat and Mass Transfer 36, pp.942-946, 2009.
[4] Fabbri, G.,*"Heat transfer optimization in corrugated wall channels"*, in International Journal of Heat and Mass Transfer, 43, pp. 4299- 4310, 2000.
[5] M. M. Awad & Y. S. Muzychka,*" Models for Pressure Drop and Heat Transfer in Air Cooled Compact Wavy Fin Heat Exchanger "*, Journal of Enhanced Heat Transfer, Vol. 18, No.3, pp. 191-207, 2011.
[6] H.M.S.Bahaidarah, *"Fluid Flow and Heat Transfer Characteristic in Sharp Edge Wavy Channels with Horizontal Pitch "*, Emirates Journal for Engineering Research Emirates Journal for Engineering Research, Vol.14, No. 1, pp. 53-63, 2009.
[7] Maryam Mirzaei et al., **"Large Eddy Simulation of the Flow and Heat Transfer in a Half-Corrugated Channel with Various Wave Amplitudes"**, Int. J Heat Mass Transfer, Vol.76, pp. 432-446, 2014.
[8] Mohammad Mehdi Rashidiet,et al., **" Effects of Phase Shift and Wavy Amplitude on the Laminar Forced Convection Heat Transfer Enhancement in Corrugated Channels Using Copper-Water Nano-Fluid"**, Journal of Computational and Theoretical Nanoscience, JANUARY 2016.
[9] Abdulsayid, A.G., *"Modeling of Fluid Flow in 2D Triangular, Sinusoidal, and Square Corrugated Channels"*, World Academy of Science, Engineering and Technology, 71, pp.34-52, 2012.
[10] Frank M., *" Fluid Mechanics"*, Fourth Edition, NASA/Color-Pic Inc., 2003.
[11] Zhang, L. *" Turbulent Three – Dimensional Air Flow and Heat Transfer in a Cross – Corrugated Triangle Duct"* , Journal of Heat Transfer, October, Vol. 127, pp.1151-1158, 2005.
[12] Holman, J.P., *"Heat Transfer"* ,Tenth Edition, Published by McGraw-Hill, a business unit of The McGraw-Hill Companies, Inc., 2010.
[13] Mohammad Zakir Hossain and A.K. M. Sadrul Islam, **"Numerical Investigation of Unsteady Flow and Heat Transfer in Wavy Channels"**, 15th Australasian Fluid Mechanics Conference , The University of Sydney, Sydney, Australia, 13-17 December 2004.
[14] Shive Dayal Pandey, V.K. Nema, **"Experimental investigation of heat transfer and friction factor in a corrugated plate heat exchanger"**, International Journal of Energy and Environment, Vol.2, *Issue 2, pp.287-296*, 2011.

Nomenclature:

A	heat transfer surface area of test duct, (m^2)
Cp	Specific heat, (J/ kg.C)
Cd	Discharge coefficient
Dh	Hydraulic diameter, (m)
do	Orifice diameter, (m)
Δho	Head of orifice , (m)
H	Heat transfer coefficient, (W/m^2 K)
I	Current, (A)
K	Thermal conductivity, (W/m K)
L	Duct length, (m)
$\dot{m}$	Mass flow rate , (kg/s)
Nu	Nusselt number, Dimensionless
p	Perimeter, (m)
Δp	pressure drop, (Pa)
Pr	Prandtl number (Dimensionless)
Qair	Air cooling load, (W)
Re	Reynolds number (Dimensionless)
So	Cross sectional area of orifice , (m^2)
T	Temperature, (℃)
u	Air velocity, (m/s)
V	Voltage, (V)
$\dot{V}$	Air volumetric flow rate (m^3/s)

Greek symbols

Δ	Difference
μ	Air viscosity, (Pa.s)
Ρ	Air density, (kg/m^3)
β	ratio of d_o/d_{pipe}

Subscripts

a	air
ab	Air bulk
ai	Air inlet
ao	Air outlet
av	Average
conv	Convection
Enh,m	Mean enhancement
Enh,h	Heat transfer coefficient enhancement
Enh,Nu	Nusselt enhancement
m	Mean
o	Orifice
w	wall