British Journal of Applied Science & Technology
18(3): 1-14, 2016; Article no.BJAST.28438
ISSN: 2231-0843, NLM ID: 101664541

SCIENCEDOMAIN *international*
www.sciencedomain.org

Heat Transfer Enhancement in Corrugated Plate Heat Exchanger

Maddali V. S. Murali Krishna[1], M. Basava Omkar Swamy[1], G. H. Manjunath[1], N. Venkateswara Rao[1], Battula Sreedhara Rao[2] and P. V. Krishna Murthy[3*]

[1]Department of Mechanical Engineering, Chaitanya Bharathi Institute of Technology, Gandipet, Hyderabad, Telangana State, India.
[2]Department of Chemical Engineering, Chaitanya Bharathi Institute of Technology, Gandipet, Hyderabad, Telangana State, India.
[3]Department of Mechanical Engineering, Annamacharya Institute of Technology and Sciences, Batasingaram, Hyderabad, Telangana State, India.

Authors' contributions

This work was carried out in collaboration between all authors. Author MVSMK designed the study. Author MBOS performed the statistical analysis, wrote the protocol, wrote the first draft of the manuscript and managed literature searches. Authors GHM, NVR, BSR and PVKM managed the analyses of the study and literature searches. All authors read and approved the final manuscript.

Article Information

DOI: 10.9734/BJAST/2016/28438
Editor(s):
(1) Grzegorz Golanski, Institute of Materials Engineering, Czestochowa University of Technology, Poland.
Reviewers:
(1) Adeyefa Oludele Adebayo, University of Ibadan, Nigeria.
(2) Saad A. El-Sayed, Zagazig Univeristy, Al Sharqia, Egypt.
(3) Abdullah Alessa, Al-Huson University College, Al-Balqa Applied University, Jordan.
(4) Seok-ho Rhi, Chungbuk National University, South Korea.
Complete Peer review History: http://www.sciencedomain.org/review-history/17207

Original Research Article

Received 19th July 2016
Accepted 18th November 2016
Published 12th December 2016

ABSTRACT

Background of the Problem: Heat exchanger is a device in which heat transfer occurs from hot fluid to cold fluid and vice versa; It is used for many applications particularly for process industries. Plate type heat exchanger gives better solution in terms of heat transfer, when compared with different types of heat exchangers. Design pressure is less than 20 bar, while design temperature is less than 180℃ for a plate heat exchanger. Little rep orts were available on heat transfer analysis of corrugated plate heat exchanger with variation of viscosity of fluid and corrugation angle.
Aim: Investigations were carried out to determine heat transfer rate of corrugated plate heat

**Corresponding author: E-mail: krishnamurthy_venkata@yahoo.co.in, krshnamurthy_venkata@rediffmail.com;*

exchanger with varied corrugation angles of 30°, 40° and 50° with varied test fluids of water, 10% glycerin, 20% glycerin and 30% glycerin.
Methodology: A single corrugation pattern on three plates arranged in parallel with varied chevron angles with spacing between the plates was 5 mm. Cold water/10% glycerol solution/20% glycerol solution/ and 30% glycerol solution were used as cold fluids, with the flow being counter flow. Hot water at 70℃ at constant flow rate was used for heating the test fuels.
Design Variables: Test fluids of different viscosities and corrugation angles.
Brief Results: It was observed that heat transfer coefficient was higher for a given Reynolds number for 50° corrugation angle compared to other corrugation angles of 30° and 40°. By using Regression analysis, a new correlation was developed for corrugated plate heat exchanger for the prediction of Nusselt number, which was a function of Reynolds number, Prandtl number and corrugation angle. Nusselt number predicted by correlation was correlated with experimental result, which was deviated by 8% only from actual experimental result. Also the correlation was compared with that of other researchers in order to validate the data.

Keywords: Heat exchanger applications; types; corrugation; viscosity; Newtonian fluids; heat transfer coefficient; Reynolds number and Nusselt number.

NOMENCLATURES

ρ	: Density of liquid, kg/m^3
μ	: Dynamic viscosity of fluid, N.s/m^2
ξ	: Effectiveness of heat exchanger
θ	: Corrugation angle
$\bar{y}$	: Mean value of the variables
ΔT	: Temperature difference between fluid and surface, K
A	: Area of the test section, m^2
D_h	: Hydraulic diameter= $\frac{4A}{P}$
Cp	: Specific heat, J/kg-K
K	: Thermal conductivity, W/m-K
N	: Number of variables in study
Nu	: Nusselt Number
P	: Wetted perimeter of the test section, m
Pr	: Prandtl Number
R^+	: Result
R^2	: Correlation coefficient
Re	: Reynolds number
S	: Standard deviation
V	: Velocity of fluid, m/s
W	: Width of the corrugated plate
W_R^+	: Uncertainty
X	: Spacing between corrugated plates, m
X	: Tested length of corrugated plate
X_1, X2, Xn	: Independent variables
y	: Variables

1. INTRODUCTION

The main objective of any designer is that heat exchanger should transmit more amount of heat transfer with minimum cost. The reason for wide application of plate heat exchanger in industry today is not only ease of maintenance but also the achievement of higher turbulence when compared to shell and tube heat exchanger.

Focke et al. [1], established that the inclination angle between plate corrugations and the overall flow direction is a major parameter in the thermo hydraulic performance of plate heat exchangers. Heggs et al. [2] developed an electrochemical mass transfer technique to calculate values of the local transfer coefficients within a corrugated plate heat exchanger (PHE) channels. They tested the Reynolds number range from 150–11500 for corrugation angles 30°, 45°, 60° and 90°. Also, they investigated that core of fluid flows along the troughs of the corrugation over the entire Reynolds number range. Muley et al. [3] presented steady-state heat transfer and pressure drop data for single-phase viscous fluid flows ($2 \leq Re \leq 400$) in a single-pass U-type counter flow PHE with chevron plates. They observed the effects of chevron angle β, corrugation aspect ratio γ and flow conditions on heat transfer. Giampietro Fabbri studied the heat transfer in a channel composed of a smooth and a corrugated wall under laminar flow conditions. [4]. The velocity and temperature distributions were determined with the help of a finite element model. The heat transfer performance of the corrugated wall channel was compared with that of a smooth wall duct. Michael C Georgiadis et al. [5] presented the mathematical modeling and simulation of complex PHE arrangements under milk fouling, using detailed dynamic models. A complex fouling model based on a reaction/mass transfer scheme was coupled with a general thermal dynamic model of PHEs. All the important factors affecting milk heat treatment were formally quantified.

Vlasogiannis et al. [6] tested a PHE under two-phase flow conditions by using an air/water mixture as the cold stream. The heat transfer coefficient of the air/water stream was measured as a function of air and Water superficial velocities. Wright et al. [7] showed that the operation of a two stream PHE was approximated after the plate rearrangement was made, using the existing PHE performance data. They assumed that the overall mass flow rates did not change with the new configuration and that truly counter-current flow was achieved. Deb Williams discussed the PHEs that transfer heat effectively by placing thin, corrugated metal sheets side by side and connecting them by gaskets. So, the flow of the substance to be heated and cooled takes place between alternating sheets allowing heat transfer through the metal sheets [8]. Jorge et al. [9] developed a mathematical model in algorithmic form for the steady-state simulation of gasketed PHEs with generalized configurations. The main simulation results were temperature profiles in all channels, thermal effectiveness, distribution of the overall heat transfer coefficient and pressure drops. Joerge et al. [10] proposed a screening method for effective optimization of the configuration of PHE which comprises number of plates, pass arrangements, location of inlet and outlet connections and type of channel flow and they developed a simplified form of the model.

Metwally et al. [11] considered Laminar periodically developed forced convection in sinusoidal corrugated-plate channels with uniform wall temperature and single-phase constant property flows. The flow field was found to be strongly influenced by kinematic viscosity γ and Re, and it displays two distinct regimes: a low Re or γ undisturbed laminar-flow regime and a high Re or γ swirl-flow regime. Gradeck M, et al. [12] performed experiments to study effects of hydrodynamic conditions on the enhancement of heat transfer for single phase flow. Finally they have pointed out a strong relation between the wall velocity gradient and the Nusselt number. Longo et al. [13] carried out experiments using water as a working fluid in herringbone type plate heat exchanger with chevron angle of 65° and developed Nusselt number correlation. They used modified Wilson plot technique and incorporated variable fluid property effects using the Buckingham Pi theorem. Lin JH, et al. [14] derived dimensionless correlations to characterize the heat transfer performance of the corrugated channel in a plate heat exchanger. The experimental data was substituted into these correlations to identify the flow characteristics and channel geometry parameters with the most significant influence on the heat transfer performance. Warnakulasuriya et al. [15] investigated heat transfer and pressure drop of a viscous absorbent salt solution in a commercial plate heat exchanger. Overall heat transfer coefficient and Nusselt number were reported to increase with Reynolds number while friction factor decreased. Based on the experimental data, correlations for Nusselt number and friction factor were proposed.

Fernandes et al. [16] studied laminar flows of Newtonian and power-law fluids through cross-corrugated chevron type plate heat exchangers (PHEs) in terms of the geometry of the channels. Single friction curves fRe = K for both Newtonian and non-Newtonian fluids were proposed for each by developing an adequate definition of the generalized Reynolds number, Re. Durmus Aydın et al. [17] studied the effects of surface geometries of three different type heat exchangers called as PHE-flat [Flat plate heat exchanger], PHE corrugated [Corrugated plate heat exchanger] and PHE asteriks [Asterisk plate heat exchanger] on heat transfer, friction factor and exergy loss. The experiments were carried out for laminar flow conditions with single pass in parallel and counter flow direction having Reynolds number and Prandtl number in the range of $50 \leq Re \leq 1000$ and $3 \leq Pr \leq 7$, respectively. Khan et al. [18] carried out experiment for single phase flow [water-to-water] configurations in a commercial plate heat exchanger for symmetric 30°/30°, 60°/60°, and mixed 30°/60° chevron angle plates. Based on the experimental data, a correlation to estimate Nusselt number as a function of Reynolds number, Prandtl number and chevron angle was proposed. Dovica et al. [19] investigated characteristics of the flow in chevron PHEs were investigated through visualization tests of channels with $\beta = 28^o$ and $\beta = 61°$. Mathematical model was developed with the aim of derived correlations for prediction of f and Nu for flow in channels of arbitrary geometry (β and b/l). Pandey et al. [20] conducted experiments were conducted to determine the heat transfer characteristics for fully developed flow of air and water flowing in alternate corrugated ducts. The various correlations obtained are $Nu_m=0.247Re^{0.83}$ and $Nu_m=0.409Re^{0.57}$ for water and air, respectively and $f = 2.014Re^{-0.12}$ for air channel. Jogi Nikhil et al. [21] conducted experiment in which heat transfer data was obtained for single phase flow (water-to-water) configurations in a corrugated plate heat exchanger for symmetric 45°/45°, 45°/75° chevron angle plates. Based on the experimental data, a correlation was estimated for Nusselt number as a function of Reynolds number, Prandtl number and chevron angle.

From the literature review it is observed that the work on corrugated plate heat exchanger was limited to water or air as the test fluid and with one corrugation angle only. Therefore, it is planned to undertake investigations in corrugated plate heat exchanger with different test fluids (having different viscosities) and also with different corrugation angles.

The present paper attempts to study the heat transfer rates with different viscous fluids and to study the effect of corrugation angles on the rates of heat transfer. It is also aimed to come out with a dimensionless correlation for the prediction of Nusselt number from these studies and to compare with flat plate heat exchanger. From these studies, the advantage of using corrugated plate heat exchange over flat plate heat exchange was proved experimentally and theoretically.

2. MATERIALS AND METHODS

2.1 Experimental Set-Up

Fig. 1 shows the schematic diagram of experimental setup used for investigations on corrugated plate heat exchanger. The equipment consists of a test box, test fluid collection tank, hot water tank, two motors and two rotameters in which the flow rates of the hot and cold fluids were regulated. The test box (1) is composed of three sinusoidal plates welded together. The experimental setup was fabricated using stainless steel material to prevent corrosion during conduction of the experiments. The test box consisted of corrugate plate (2) with 30°/40°/50° corrugation angles. It was fabricated using three stainless steel plates welded in a well specified manner to form two adjacent channels. Fig. 2 shows the schematic diagram of corrugated plate with corrugated angle, which was measured with respect to horizontal. The spacing provided for the top channel is 15 mm. Test Length × width × plate spacing of corrugated plate were 300 mm × 100 mm × 5 mm. Developed length of the corrugated plate was 600 mm making 6 corrugations per total length of 600 mm. Fig. 3 shows the configuration diagram of arrangement of thermocouples placed on corrugated plates. T_1 and T_{11} are the thermocouples in the hot fluid channel (upper channel of the plate) at inlet and out of fluid. Hot fluid flows in the upper channel from left to right (T_1 – T_{11}). Cold fluid flows in the bottom channel from right to left T_3–T_2. T_3 and T_2 are the thermocouples in the cold fluid channel (lower channel of the plate) at inlet and out of fluid. T_4 and T_{10} are the thermocouples on the film plate (plate which divides upper and lower channel as shown in the figure) at inlet and out of fluid respectively. These thermocouples were connected to a digital temperature indicator.

Fig. 4 shows photographic view of fixing thermocouples of middle corrugated plate. Fig. 5 shows photographic view of top view of top corrugated plate, while Fig. 6 shows photographic view of experimental set up.

The flow pattern adopted here was countercurrent flow. The study was conducted using four Newtonian fluids namely water, 10% Glycerol solution (10% of glycerin + 90% of water by volume), 20% Glycerol solution and 30% Glycerol solution. Hot water was used as hot fluid. Pressure gauzes (5) of 200 mm dial gauze were provided to note down the pressure of hot and cold fluids. Rotometers (6) were provided to measure flow rate of hot fluid and cold fluid. Valves were (7) provided to regulate the flow of fluids. Two motors (8) of capacity 0.25 HP and 1500 rpm each were arranged to pump test fluids. Hot water tank (9) and test fluid tank (10) were provided for storing test fluids. The dimensions of cold fluid tank and hot fluid tank were each 300 mm ×300 mm × 600 mm. For collecting cold fluids, test fluid collection tank (11) was provided. Heater (12) was provided for hot fluid with 3000 W capacity. The test section was formed by three identical corrugated channels having corrugation angle of separately 30°/40°/and 50°/ with water/10% glycerne/ 20% glycerin/30% glycerin flowing in the bottom one and hot water on top of the section of the channel.

The flow through these channels was supported by other auxiliary components like test fluid storage tank, hot fluid storage tank. Thermodynamic properties like density, viscosity, thermal conductivity and specific heat were experimentally determined at the required temperatures to account for the changes in property with temperature. The authors obtained these values from laboratory of Indian Institute of

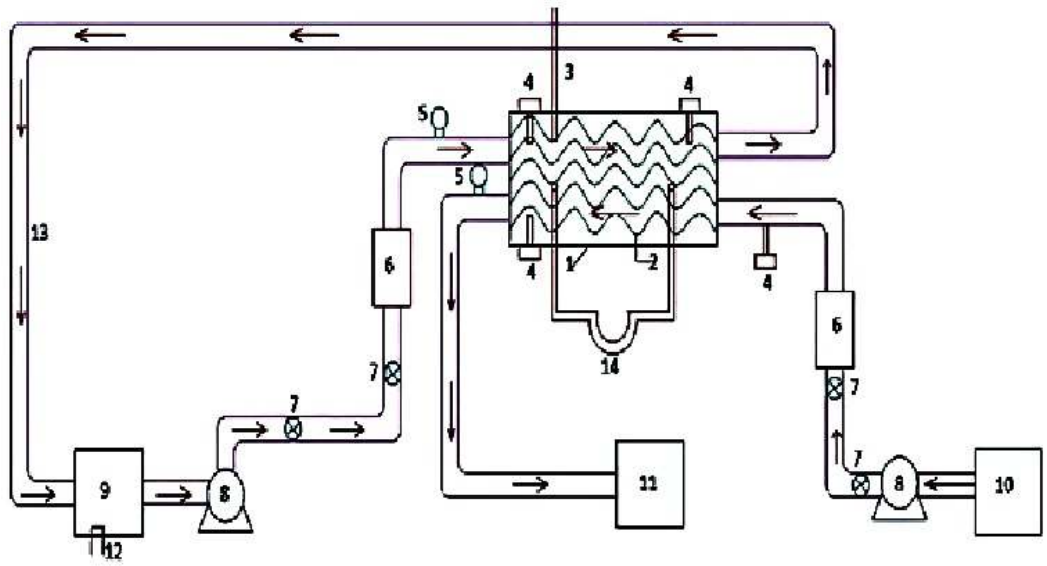

Fig.1 Schematic diagram of experimental set-up

1.Test box, 2.Corrugaed plate, 3. Thermocouples, 4. Thermocouples, 5. Pressure gauges, 6. Rotameters, 7.Ball valves, 8. Motors, 9.Hot water tank, 10. Test fluid tank, 11. Test fluid collection tank, 12. Heating coils, 13. Hot water and test fluid running pipes, and 14. Liquid manometers

Chemical Technology Hyderabad, India for accurate values instead of measuring them in institute. The physical and thermal properties of cold fluids were given in Table 1. Liquid manometers (14) were provided for noting down the pressure difference or pressure drop across the length of test section for cold fluid.

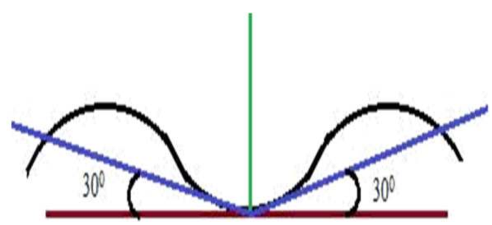

Fig. 2. Schematic diagram of corrugated plate with corrugated angle

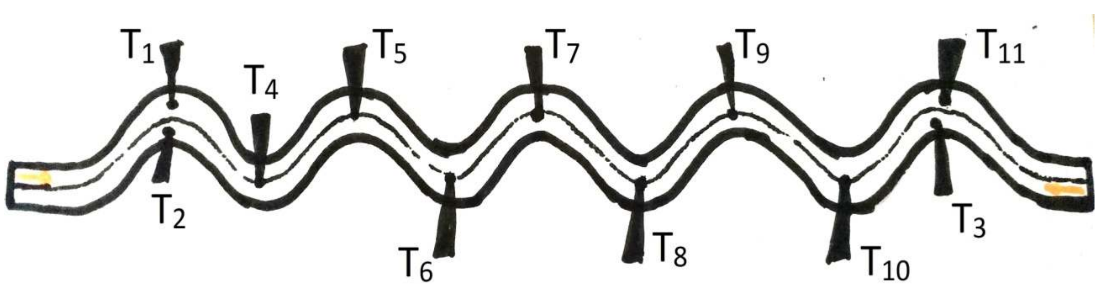

Fig. 3. Configuration diagram of arrangement of thermocouples on corrugated plates

Table 1. Physical and chemical properties of cold fluids (courtesy from IICT)

Parameter	Water	10% Glycerin	20% Glycerin	30% Glycerin
Viscosity (cp) at 35℃	0.73	0.98	1.39	2.03
Thermal Conductivity at 35℃	0.63	0.50	0.39	0.31
Specific gravity at 35℃	1.00	1.02	1.04	1.06
Specific heat at 35℃	4.18	4.03	3.83	3.65

Fig. 4. Photographic view of fixing thermocouples of middle corrugated plate

Fig. 5 Photographic view of top view of top corrugated plate

Fig. 6. Photographic view of experimental set up

2.2 Methodology

For all the experiments, hot water was made to flow through the top corrugated channels to maintain the channel surfaces at an approximately constant temperature and this water was used for heating the test fluids. For each experimental reading, the inlet and outlet temperatures of the fluid as well as the wall temperatures on the heat exchanger plate at seven different locations were noted by means of thermocouples, welded at these locations and read through the digital temperature indicator. These temperatures were used for the analysis of heat transfer. For making the heat transfer studies, the hot fluid flow rate was maintained constant. The test fluids were pumped into the bottom channel through the calibrated rotameter from 0.5 to 6 lpm. The middle plate was fitted with 7 thermocouples, along the length and breadth of the plate, to measure the wall temperatures. Four more thermocouples were inserted into the bulk fluid to measure the inlet and outlet temperatures of hot and cold fluids. These thermocouples were connected to a digital temperature indicator having an accuracy of 0.1°C. For each flow rate the inlet and outlet temperatures as well as the wall temperatures were noted from the temperature indicator, when it showed a constant value. For all the heat transfer studies the inside film heat transfer coefficient (h_i) was calculated by making an energy balance with log Redwood Viscometer No. 1 (ASTM Standard) and hydrometer respectively. The Reynolds number based on hydraulic diameter varied from 0 to 3000 for water; 0–1850 for 10% glycerin, 0–965 for 20% glycerin and 0–820 for 30% glycerin changing the mass flow rate of the two fluids.

2.3 Experimental Procedure

- The experiment was carried out for PHE having 10 mm spacing between the plates for a test fluid and for a hot fluid it was 15 mm.
- The fluids used in the experiment were water/10% glycerin/ 20% glycerin/30% glycerin / as cold fluid and hot water as hot fluid.
- For all experiments hot water at 70°C at constant flow rate was used for heating the test fluid.
- For every step, inlet and outlet temperature of a fluids and wall temperature of exchanger should be noted

down by means of seven thermocouples, which are welded on the test plate at seven different locations.

- These temperatures were used for the analysis of heat transfer. For these studies the hot fluid flow rate is maintained constant.
- The test fluids were pumped into the bottom channel through the calibrated rotameter from 1.8 to 4 lpm.
- The middle plate was fitted with 7 thermocouples, along the length and breadth of the plate, to measure the wall temperature. 4 more thermocouples are inserted into bulk fluid to measure inlet and outlet temperatures of hot and cold fluids.
- For each different flow rates, all the temperature was noted down, till the system attained steady state.
- For all the heat transfer studies the inside film heat transfer coefficient (h_1) was calculated by making an energy balance with log mean temperature difference (LMTD).
- The viscosity and specific gravity of the fluids were determined experimentally by viscometer and hydrometer respectively.

2.4 Calculations

T_1, T_{11} represented hot fluid temperature at inlet and outlet while T_3, T_2 represented cold fluid temperatures at inlet and outlet. T_4 to T_{10} represented film temperatures.

ΔT_1 = Temperature drop at the inlet = $T_{avg} - T_{c\ inlet}$

ΔT_2 = Temperature drop at exit = $T_{avg} - T_{c\ outlet}$

$$T_{avg} = \frac{T4+T5+T6+T7+T8+T9+T10}{7}$$

$$LMTD = \frac{\Delta T1-\Delta T2}{\ln\left(\frac{\Delta T1}{\Delta T2}\right)}$$

$m_{C\,=}$ mass flow gate of cold fluid

$$m_C\ (m/s) = \frac{\rho \times \mathrm{Flowrate}}{60\times100}$$

m_h (m/s) = mass flow rate of hot fluid

$$m_h\ (m/s) = \frac{\rho \times \mathrm{Flowrate}}{60\times100}$$

$$Q_C = m_C\, C_P\, \Delta T_C$$

Q_C = Amount of heat gained (KW)

C_P = specific heat (KJ/Kg K)

$\Delta T_C = T_{c\ outlet} - T_{c\ inlet}$

A = Area of head of the heat exchanger

A = l x b (m^2)

l = effective length of the plate (m)

b = breath of the plate (m)

Q_C = h A ΔT

A = cross sectional area of the plate

$$A = \frac{\pi d2}{4}\ (m^2)$$

Where $d = \frac{2bh}{b+h}$ = equitant diameter (m)

Nusselt number Nu

$$Nu = \frac{hl}{K}$$

K = thermal conductivity

Reynolds number Re

$$Re = \frac{\rho\, v\, d}{\mu}$$

V = velocity of the cold fluid (m/s)

$$V = \frac{\mathrm{discharge}}{\mathrm{area}}$$

3. RESULTS AND DISCUSSION

Fig. 7 shows variation of heat transfer coefficient with Reynolds number for water, with corrugated plate heat exchanger with corrugation angles of 30°, 40° and 50°. It is noticed from this Fig. 7, that heat transfer coefficient varied linearly with Reynolds number for different configurations of heat exchanger. Heat transfer coefficient was higher for a given Reynolds number for 50^0 corrugation angle compared to 30° and 40° corrugation angles. High velocities and swirling flows of the fluid generated with higher corrugation angles might have increased heat transfer coefficient. This trend is in agreement with the reported work in literature [7].

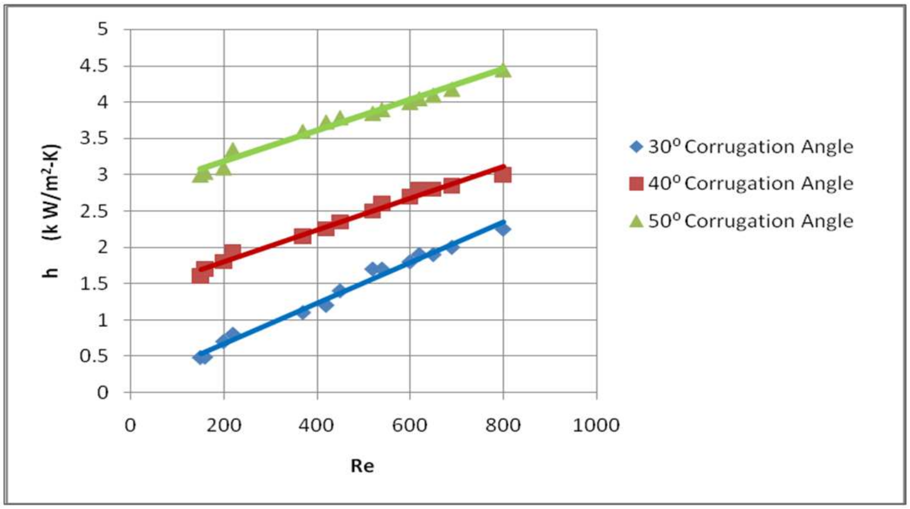

Fig. 7. Variation of heat transfer coefficient h, (k W/m^2-K)) with Reynolds number for water with corrugated plate heat exchanger with 30°, 40° and 5 0° corrugation angles

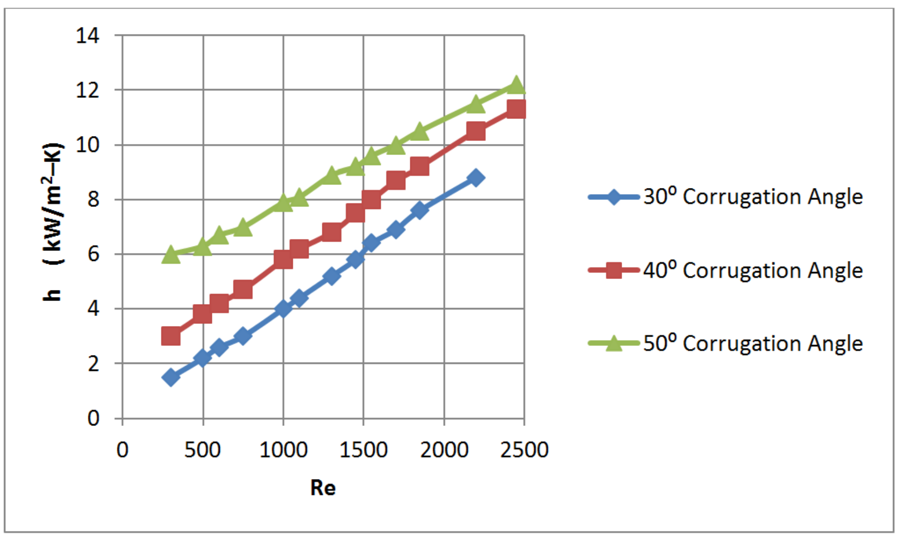

Fig. 8. Variton of heat transfer coefficient (h, kW/m^2–K) with Reynolds number (Re) for 30% Glycerol with corrugated plate heat exchanger with 30°, 40° and 50° corrugation angles

Fig. 8 above shows variation of heat transfer coefficient with Reynolds number with 30% glycerol (typical diagram) with corrugated plate heat exchanger with 30°, 40° and 50° corrugation angles. Since 30% glycerol has higher viscosity than 10% glycerol and 20% glycerol, 30% glycerol was taken as cold fluid in order to study the effect of viscosity on heat transfer coefficient. It is observed from this figure that heat transfer coefficient was higher for a given Reynolds number for 50° corrugation angle compared to 30° and 40° corrugation angles like in case of water. Though, the fluid is laminar high swirling flows and velocity of the fluid might have increased heat transfer rate for higher corrugation angles.

Fig. 9 shows variation of Nusselt number with Reynolds number for different test solutions like water, 10% glycerol, 20% glycerol and 30% glycerol for corrugated plate heat exchanger with 50^{o} corrugation angle. From this Figure, it is observed that Nusselt number increased with Reynolds number for all the fluids considered. It can also be observed from Fig. 9, that Nusselt number was higher for 30% Glycerol solution compared to 20% glycerol, 10% Glycerol and water at a given Reynolds number. This was also evident from theoretical considerations for a given Reynolds number, Nusselt number increases with Prandtl Number. Prandtl number for water varies from 1.7 – 13.7, while it varies from 2000 –100,000 for glycerol (100% glycerin solution). This was because of higher dynamic viscosity (1499 cp), moderate specific heat (2.41 kJ/kg-K) and lower thermal conductivity (0.285 W/m-K) with 100% glycerol solution at room temperature against water at lower dynamic viscosity (0.894 cp), higher specific heat (4.18 kJ/kg-K) and higher thermal conductivity (0.609 W/m-K). Therefore Prandtl number was higher with 30% glycerol solution compared to 20% glycerol, 10% Glycerol and water. The similar trends can be observed for other corrugation angles. The increase in Nusselt number will be more significant at higher corrugation angles (say 50° to lower corrugation angles). Heat transfer rate increased with 30% glycerol solution when compared with other test solutions with heat exchange with corrugation angle of 50°. As the corrugation angle decreases, the geometry reduces to flat plate heat exchanger.

3.1 Dimensional Analysis

Heat Transfer coefficient is function of fluid properties like density, dynamic viscosity, heat capacity and thermal conductivity and also on the parameters like hydraulic diameter, velocity, spacing, corrugation angle etc.

$$h = \varphi\left(\rho, \mu, C_{P,} k, D_{h,} V, x, \theta, \Delta T\right)$$

$$D_h = \frac{4A}{P} = \frac{2Wx}{W+x}$$

By dimensional analysis, it is found that Nu is function of Re, Pr, cosθ and ‘x*’Where x*= x/D.

For a given spacing, Nu is a function of Re, Pr and cosθ only. $Nu = \varphi(Re, Pr, cos\theta)$.

The dimensionless parameter groups Π1 to Π5 listed in following Table 2 were obtained by the procedure as explained below. The dimensionless correlation of heat exchanger performance within the channel was explained. As spacing was fixed (5 mm), the above equation, reduced into $\pi 1 = a\, \pi_2^{\ b} \pi_3^{\ c} \pi_4^{\ d}$

By Regression Analysis using statistical software like matlab, one can find the coefficients and all the exponents to various dimensionless groups by giving input that is, experimental value of Nu, Re, Pr and cosθ.

The obtained correlation is $Nu = 0.46(Re^{0.61} Pr^{0.1} cos\theta^{-2.45})$ with R^2 value of 0.93.

Effectiveness of this experimental corrugated PHE was found as 82%. Reynolds number, Prandtl number and Nusselt number were established by experiment and correlated with theoretical result determined by above equation.

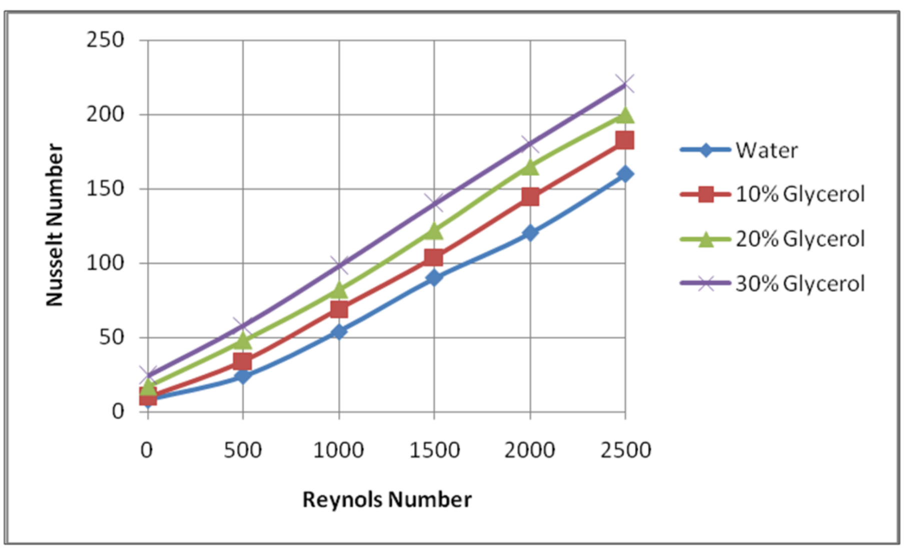

Fig. 9. Variation of Nusselt number with Reynolds number with heat exchanger with 50% corrugation angle with test solutions

However, the correlation obtained by other researcher was $Nu=0.247Re^{0.83}$ [20]. The variation might be with operating conditions adopted by the other researcher. In their experiment, water was allowed to pass through middle corrugated chamber, while air was allowed to pass through top and bottom corrugated chambers. However, the authors sent water or glycerin through bottom corrugated chamber and hot water on top channel. Mass flow rate of air carried out in the experiment by the other researchers was reduced by an half. The corrugation angle was not taken into consideration by other researchers, as it was fixed in their experiment [20]. However, variation of corrugation angle was provided by the authors and hence in their correlation, angle theta was accommodated. Thermal conductivity term was there in LHS equation of the correlation proposed by the authors in form of Nusselt number. Same term occupied in RHS of the equation in form of Prandtl number developed by the authors. But in case of equation developed by other researchers [20], thermal conductivity term was not there in right hand side of the equation. There was deviation of 8% obtained in the value of heat transfer coefficient between the data of authors and other researchers [20].

Through Brien and Sparrow correlation for water was used by authors and other researchers in evaluating dimension correlation, however, the experimental conditions were different [22].

3.2 Goodness of Fit for Various Curves

The calculated in terms of the following statistical parameters namely, Standard deviations (s) and Square of the correlation coefficient (R^2), defined by

$$s = \sqrt{\frac{\sum (y_i - \bar{y})^2}{(n-1)}}, \text{ and}$$

$$r^2 = 1 - \frac{\sum (y_i - y_{ip})^2}{\sum (y_i - \bar{y})^2} = 1 - \frac{norm(d,2)^2}{(n-1)s^2}$$

Values of s and r^2 for Nusselt number Nu_m are 0.3386 and 0.9997, respectively as observed by other researchers [20]. The authors obtained 0.3256 as the value of standard deviation, while they obtained different values of R^2 for different situations as explained below. The accepted range of R^2 should be greater than 0.90, R^2=1 was an ideal case, where the expected and predicted values were exactly same by the authors. From, this relation, it was established that Nusselt number was proportional to Reynolds number, Prandtl number and inversely proportional to cosine angle of corrugated plate. Fig. 10 shows the variation of Nusselt number (experimental) and with Nusselt number (predicted) with corrugation angles of 30°, 40° and 50° for different test solutions. The proposed correlation was in good agreement with the experimental values and this correlation can be used for corrugated plate heat exchanger for the prediction of Nusselt numbers for the range of Reynolds numbers 110 to 2400.

From the values of R^2 and standard deviation were for different for different systems [standard deviation=0.3256; $R^2$for water is 0.91' for 10% Glycerol solution, standard deviation =0.3345; R^2 is 0.933; for 20% Glycerol solution, standard deviation =0.3432 R^2 is 0.938; for 30% Glycerol solution R^2 is 0.944, while standard deviation it is 0.3476] it was concluded that the value of R^2 form all fluids together have good value and was found to be 0.93 and this was in the acceptable range. So from this analysis one can conclude that the given correlation was in good agreement.

It is observed from the above discussion that the correlation proposed for the corrugated plate heat exchanger predicted Nusselt number with an R^2 value of 0.93. It is also observed that the proposed correlation can be used effectively for the prediction of heat transfer rates for the similar types of corrugate plate heat exchanger for a given corrugation angle. As the corrugation angle decreased, it reduced to a flat plate heat exchanger (when the angle is zero), and when the corrugation angle increased it tends to a square channel plate heat exchanger. The proposed correlation predicted higher heat transfer rates for higher values of corrugation angles and predicted lower values of heat transfer for lower corrugation angles and which is also true theoretically. However, heat transfer rate increased up to certain corrugation angle and later it may decrease with an increase of corrugation angle due to the separation of flow. Similar trends were expressed by other researchers [1].

Fig. 11 shows comparison of Nusselt number and Reynolds number for 30% Glycerol solution (typical). The In this graph the trend line is drawn with R^2 values greater than 0.986. It can be inferred from graph, that the corrugated plate heat exchanger predicted higher values of Nusselt number compared to flat plate heat

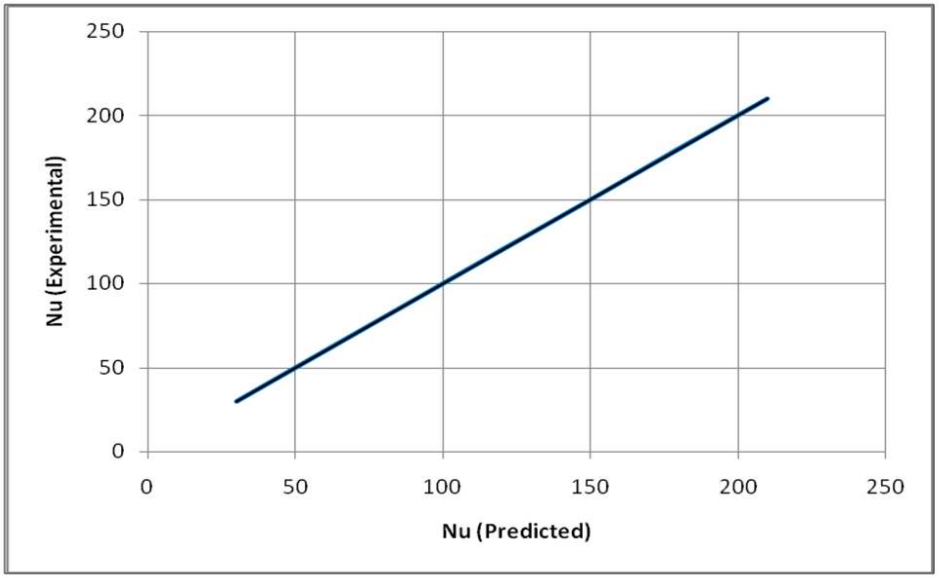

Fig . 10. Variation of Nusselt number (experimental) with Nusselt number (predicted) for all systems with corrugation angles of 30°, 40° and 50° for different test solutions

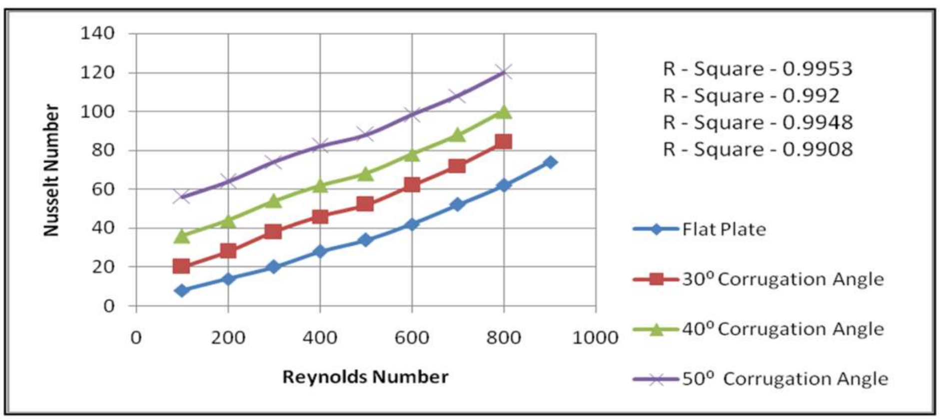

Fig. 11. Comparison of Nusselt number and Reynolds number for 30% Glycerol solution (typical)

exchanger Increase of swirling flows and velocity might have increased heat transfer with corrugated plate heat exchanger in comparison with flat plate heat exchanger. Hence, higher heat transfer rates can be achieved for corrugated plate heat exchanger compared to flat plate heat exchanger.

3.3 Uncertainty of Measurements

Estimate of experimental errors on the local Nusselt number (Nu_x) and effectiveness (ϵ) taking into account the uncertainties (σ) associated with each of the independent variables.

The uncertainty W_R^+, arising in calculating a result R^+ due to several independent variables, is given as follows;

$$W_{R^+} = \left(\left(\frac{\delta R^+}{\delta X_1} w_1 \right)^2 + \left(\frac{\delta R^+}{\delta X_2} w_2 \right)^2 + \cdots .. + \left(\frac{\delta R^+}{\delta X_n} w_n \right)^2 \right)^{1/2}$$

where the result R^+ is a given function of the independent variables X_1, X_2, ... X_n and w_1, w_2, ... w_n are uncertainties in the independent variables [20]. Uncertainty calculations showed maximum

Table 2. Dimensionless groups

S.No	Groups	Definition	Effect	Range
1	Π1	$\frac{hD}{k}$	Nusselt number	---
2	Π2	$\frac{DV\rho}{\mu}$	Reynolds number	11.3–2385
3	Π3	$\frac{\mu Cp}{k}$	Prandtl number	3.4–15.4
4	Π4	cosθ	Geometry	0.643–0.866
5	Π5	$\frac{x}{Dh}$	Location	0.5247

value of 2.8, 5.3 and 4.0 in results for Reynolds number, Nusselt number, Prandtl and number respectively by the other researchers [20]. The authors observed the uncertainties in Reynolds number, Nusselt number and Prandtl number were 3.0, 5.8 and 4.5. The individual contributions to the uncertainties of the non-dimensional parameters, for each of the measured physical properties are summarized in Table 3.

Table 3. Typical uncertainties of relevant variable

S. No	Variable	Uncertainity
1	Hot fluid inlet temperature	$\pm 0.1\ C$
2	Hot fluid outlet temperature	$\pm 0.1\ C$
3	Cold fluid inlet temperature	$\pm 0.1\ C$
4	Cold fluid outlet temperature	$\pm 0.1\ C$
5	Inner wall temperature	$\pm 0.1\ C$
6	Outer wall temperature	$\pm 0.1\ C$
7	Ambient temperature	$\pm 0.1\ C$
8	Hydraulic diameter	$\pm 1\ mm$
9	Water mass flow rate	$\pm 5\ ml$
10	Pressure difference	$\pm 1\ mbar$
11	Redwood viscometer time	1 second
12	Hydrometer	0.01 gm/cc
13	Rotometer	0.01 lpm

4. CONCLUSIONS

1. Both heat transfer coefficient and Nusselt number increased with increase in corrugation angle for a particular fluid.
2. With an increase of corrugation angle by 30° to 40° heat transfer rate increased by 15%, for 40° to 50° corrugation angle there is a 30% increase in heat transfer rates and for 30° to 50° corrugation angle the increase is as high as 50% for all fluids.
3. Nusselt number and heat transfer coefficient were higher for high viscous fluids at a given Reynolds number and for a given corrugation angle.
4. As the corrugation angle increased, there was significant improvement in the Nusselt number or heat transfer rates for high viscous fluids compared to low viscous fluids.
5. Therefore a low viscous fluid at higher corrugation angle will result in the same rates of heat transfer compared to high viscous fluids for lower corrugation angles. This aspect can be taken into consideration in choosing the type of corrugated plate heat exchanger in design.
6. The proposed correlation was in good agreement with the experimental data and this correlation can be used for corrugated plate heat exchanger for the prediction of heat transfer rates for the range of Reynolds numbers carried out in the investigations.
7. Corrugated PHEs resulted in higher heat transfer rates compared to flat plate heat exchanger.

4.1 Future Scope of Studies

Studies on pressure drop can be taken up with corrugated plate heat exchanger.

HIGHLIGHTS

- Increase in corrugation angle affects heat exchanger performance.
- Increase in viscosity of the test fluid affects the heat transfer rate.
- The proposed relationship among variables affecting heat exchanger performance is in good agreement with experimental data.

- Corrugated plate heat exchange is proved to be efficient when compared with flat plate heat exchanger.

ACKNOWLEDGEMENTS

Authors thank authorities of Chaitanya Bharathi Institute of Technology (CBIT), Hyderabad and for providing facilities for carrying out research work. Authors thank authorities of Indian Institute of Chemical Technology, Hyderabad for rendering help in evaluating thermal and physical properties of test fluids used in the experiment. Financial assistance provided by R & D of CBIT Hyderabad is greatly acknowledged.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

REFERENCES

1. Focke WW. Zachariades J, Olivier I. The effect of the corrugation inclination angle on the thermo hydraulic performance of plate heat exchangers. Int. J. Heat Mass Transfer. 1985;28(8):1469–1479.
2. Heggs PJ, Sandham P, Hallam RA, Walton C. Local transfer coefficient in corrugated plate heat exchanger channels. Trans I Chem E. 1997;75:Part A.
3. Muley A, Manglik RM. Experimental study of turbulent flow heat transfer and pressure drop in a plate heat exchanger with chevron plates. J. Heat Transfer 1β. 1999;1(1):110-117.
4. Giampietro Fabbri. Heat transfer optimization in corrugated wall channels. International Journal of Heat and Mass Transfer. 2000;43(23):4299–4310.
5. Michael C Georgiadis, Sandro Macchietto. Dynamic modeling and simulation of plate heat exchangers under milk fouling. Chemical Engineering Science. 2000; 55(9):1605-1619.
6. Vlasogiannis P, Karagiannis G, Argyropoulos P, Bontozoglou V. Air–water two-phase flow and heat transfer in a PHE. International Journal of Multiphase Flow. 2002;28(5):757–772.
7. Wright AD, Heggs PJ. Rating calculation for PHE effectiveness and pressure drop using existing performance data. Trans I Chem E. 2002;80:Part A.
8. Deb Williams. Plate heat exchangers. State of the Art Engineering Report, MME. 2002;499:1-9.
9. Jorge AW Gut, Jose M Pinto. Selecting optimal configurations for multisection plate heat exchangers in pasteurization processes. Ind. Eng. Chem. Res. 2003;4β(β4):6112–6124.
10. Jorge AW Gut, Jose M Pinto. Configuration design for plate heat exchangers and pasteurizers: Mathematical modeling and optimization methods. Em: 6th Int. Conference on Foundations of Computer-Aided Process Design. 2004;357-360.
11. Metwally HM, Manglik RM. Enhanced heat transfer due to curvature-induce lateral vortices in laminar flows in sinusoidal corrugated-plate channels. International Journal of Heat and Mass Transfer. 2004;47:2283–2292.
12. Gradeck M, Hoareau B, Lebouche M. Local analysis of heat transfer inside corrugated channel. International Journal of Heat and Mass Transfer. 2005;48:1909–1915.
13. Longo GA, Gasparella A. Refrigerant R134a vaporization heat transfer and pressure drop inside a small brazed plate heat exchanger. International Journal of Refrigeration. 2007;30:821–830.
14. Lin JH, Huang CY, Su CC. Dimensional analysis for the heat transfer characteristics in the corrugated channels of plate heat exchangers. International Communications in Heat and Mass Transfer. 2007;34:304–312.
15. Warnakulasuriya FSK, Worek WM. Heat transfer and pressure drop properties of high viscous solutions in plate heat exchangers. International Journal of Heat and Mass Transfer. 2008;51:52–67.
16. Carla S, Fernandes Ricardo P, Dias Joao M, Nobregad Joao, Maiad M. Friction factors of power-law fluids in chevron-type plate heat exchangers. Journal of Food Engineering. 2008;89(4):441–447.
17. Durmus Aydın, Benli Huseyin, Kurtbas Irfan, Gul Hasan. Investigation of heat transfer and pressure drop in plate heat exchangers having different surface profiles. International Journal of Heat and Mass Transfer. 2009;52:451–1457.
18. Dovica D, Palmb B, Svaica S. Generalized correlations for predicting heat transfer and pressure drop in plate heat exchanger

channels of arbitrary geometry. International Journal of Heat and Mass Transfer. 2009;52(19–20):4553–4563.
19. Khan TS, Khan MS, Chyu Ming-C, Ayub ZH. Experimental investigation of single phase convective heat transfer coefficient in a corrugated plate heat exchanger for multiple plate configurations. Applied Thermal Engineering. 2010;30:1058–1065.
20. Pandey SD, Varma VK. Investigations of the performance of an experimental plate heat exchanger in single phase flow. International Journal of Energy Engineering. 2011;1(1):19-24.
21. Jogi Nikhil G, Lawankar Shailender M. Heat transfer analysis of corrugated plate heat exchanger of different plate geometry: A review. International Journal of Emerging Technology and Advanced Engineering. 2012;2(10):110-115.
22. Brien JEO, Sparrow EM. Corrugated-duct heat transfer, pressure drop, and flow visualization. ASME Journal of Heat Transfer. 1982;104:410-416.

Peer-review history:
The peer review history for this paper can be accessed here:
http://sciencedomain.org/review-history/17207