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Experimental investigation of heat transfer and friction factor in a corrugated plate heat exchanger

Shive Dayal Pandey, V.K. Nema

Department of Mechanical Engineering, MNNIT, Allahabad-211004 (U.P), India.

Abstract
Experiments are conducted to determine the heat transfer characteristics for fully developed flow of air and water flowing in alternate corrugated ducts with an inter-wall spacing equal to the corrugation height. The friction factor is found for air channel. The test section was formed by three identical corrugated channels having corrugation angle of 30^o with cold air flowing in the middle one and hot water equally divided in the adjacent channels. Sinusoidal wavy arcs connected with tangential flat portions make the said corrugation angle with transverse direction. The Reynolds number based on hydraulic diameter varied from 750 to 3200 for water and from 16900 to 68000 for air by changing the mass flow rates of the two fluids. The Prandtl numbers were approximately constant at 2.55 for water and 0.7 for air. The various correlations are obtained $Nu_m=0.247Re^{0.83}$ for water, $Nu_m=66.686Re^{0.18}$ and friction factor $f = 0.644 / Re^{0.18}$ for air.

Keywords: Plate type heat exchanger, Corrugated channel, Nusselt number, Friction factor, Forced convection.

1. Introduction

Heat exchangers are devices which are used to transfer heat between two flowing fluids. In the 1930s plate heat exchangers (PHEs) were introduced to meet the hygienic demands of the dairy industry. Today the PHEs are universally used in many fields; heating and ventilation, breweries, dairy, food processing, pharmaceuticals and fine chemicals, petroleum and chemical industries, power generation, offshore oil and gas production, onboard ships, pulp and paper production etc. Plate heat exchangers also find applications in water to water closed circuit cooling water systems using a potentially corrosive primary cooling water drawn from sea, river, lake, or cooling tower, to cool, non-corrosive secondary liquid flowing in a closed circuit. Design theory for PHEs is given at length in reference [1], while its application to rate Dhruva process water/sea water PHE used in a nuclear power plant at Trombay, Mumbai (India) was dealt in reference [2].
A compact heat exchanger has been arbitrarily defined as having an area density greater than 700 m^2/m^3 for units operating in gas streams and in excess of 300 m^2/m^3 when operating in liquid or two-phase stream [3]. In literature [4, 5] it is shown that the use of corrugated channel results in a more complex flow structure and improves the heat transfer by as much as two or three times compared to a conventional straight channel. In [6, 7] the authors demonstrated that the sinusoidal wavy plate arrangements and channel geometries improved the heat transfer performance by increasing the surface area and prompting the formation of vortex in the flow. The symmetric arrangement yields a superior

heat transfer performance to an asymmetric arrangement. Unfortunately, the geometric parameters are not clearly expressed. Due to the high heat transfer efficiency of the plates, a PHE is very compact when compared to a shell and tube heat exchanger with the same heat transfer capacity. Some of the research works on plate heat exchangers pertain to heat transfer and pressure drop characteristics of an absorbent salt solution in a commercial plate heat exchanger [7]. A Low cost route was developed to heat recovery in the Plate heat exchanger [8].

Some related work has been performed in the past regarding heat transfer in corrugated ducts. Beloborodov and Volgin [9] employed a corrugated duct having an inter wall spacing and corrugation angle. The experimental apparatus is not well defined, but it appears to have been a two-fluid heat exchanger, with no direct measurement of wall temperature, such that the obtained heat-transfer coefficients are averages for the device as a whole. The corrugations were formed from sheet metal and thus could not have the sharp-edged corrugation peaks. In light of these uncertainties, it is difficult to accord a great deal of generality to the correlations presented in [9]. The research performed by Goldstein [10] represents another contribution to the literature on heat transfer in corrugated flow passages. This work was largely concerned with the low Reynolds number range from 150 to 2000, within which secondary flows were identified by high-resolution local mass-transfer measurements. The corrugated channel of [10] had only two corrugation cycles, and thus the Sherwood number results are applicable only to the region of flow development near the inlet of the corrugated duct.

A situation which has certain similarities to corrugated-duct flow is flow through tube banks, commonly encountered in heat exchangers. The similarities include periodic regions of both recirculation and forward flows. Tube-bank information [11, 12] will be called upon to provide perspective for the present Nusselt-Reynolds-Prandtl correlations.

O'Brien and Sparrow [13] conducted the trial in a corrugated duct having an inter wall spacing employing two different but complementary apparatuses, one was for heat transfer correlations and other for pressure drops. This was for only one fluid which flowed through the corrugated duct. Lin et al. [14] in their experimental work, investigated heat exchange between two fluids air and water, the former flowed through the central corrugated duct and the latter through the outer channels having one surface corrugated and other flat. Dimensional analysis was performed and the significance of various dimensionless groups determined.

Experimental work by Liu & Tsai [15] for cross corrugated channels of a small plate heat exchanger showed that the laminar flow changes to turbulent at Re > 300, while the work by Focke et al. [16] on corrugated channels for 60^o corrugation angle mentions the critical Re to be 400. It was observed in the CFD model and laboratory tests [17, 20] that the flow was turbulent for Re in the range 600 to 1700.

In continuation of the experimental studies an experiment was conducted for finding heat transfer coefficient and friction factor in corrugated channels. Ten wavy cycles were used against two in paper by [14]; outer channels have both surface corrugated unlike in [14] where only one surface was corrugated. The setup is described below.

2. Experimental setup

The experimental set up shown schematically in the Figure 1 and its pictorial top view with upper plate and insulation removed is viewed for top in the Figure 2. The test box of length, width and height 70 cm x 11 cm x 25 cm, respectively, is made of plane 0.6 mm GI sheet. In the side walls of 11cm x 25 cm face, provision is made for holding the plates, thereby fixing the spacing between the corrugated plates which form a fluid channel. The corrugated plates are also made of same material, having corrugation angle 30^o, formed by a forming tool of 120^o included angle. Thus the projected area of the test section is (70×11) cm^2 and developed length of a corrugated plate is 82 cm. In the machining, care was taken to achieve a hydro-dynamically smooth surface finish on corrugations.

The walls are positioned in such a way that the peaks of both the top and bottom walls lie in the same plane. There are eight troughs and nine crests in a corrugated channel. The gap between two corrugated plats is 5cm. Insulation was placed in the spaces between the outer corrugated plates and horizontal plane walls of the box, to reduce heat transfer across the plates.

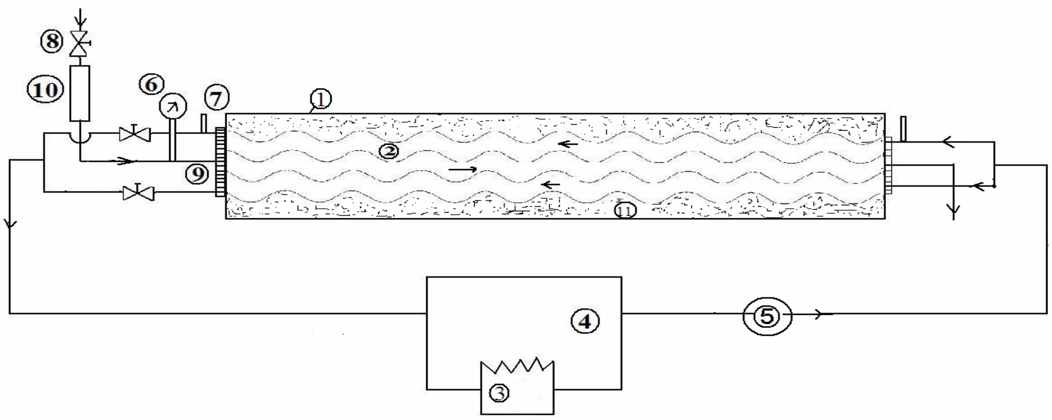

Figure 1. Schematic of the experimental setup: [1-GI sheet case, 2-Fluid channel, 3-Water heater, 4-Water container, 5-Water pump, 6-Pressure gauge, 7-Thermometer, 8-Control valve, 9-Flow straightener, 10-Rotameter, 11-Insulated chamber]

Figure 2. Corrugated channel in test section

The basic components in the experimental setup fabricated to investigate the heat transfer characteristics in the corrugated channel for different flow conditions are numbered in the schematic and their names given along with the caption. The experimental apparatus include a closed water loop, an open air loop, and a measurement system. The water loop comprises a water tank containing a heater, a pump, a flow meter, and a temperature controller. Importantly, all of the components in the water system are thermally insulated such that wall temperature of the corrugated channel can be maintained at a nearly constant temperature. The air loop consists of the test section containing the corrugated channel, a blower, an air flow meter, and a number of valves which enable the flow rate to be adjusted. Additionally, a flow straightner is installed at the entrance of the test section to maintain a uniform inlet flow.

During experimentation, hot water was flowed through the outer two corrugated channel to maintain the channels surfaces at an approximately constant temperature and j-type thermocouples wrapped in copper tubes and inserted by drilling holes in front metal plate and junctions located in the middle plane in the channel were used to record the corresponding fluid temperature at locations along the channel. Their numbers were 9 in air and 7 in water channel at different locations along the length of the channel. Wall temperatures were measured at six places at three equidistant sections in the air channel with the j-type thermocouples inserted through back plate. Wall temperature was calculated as an arithmetic average of temperatures at two places one below vertically other, at a given axial location. The temperatures of the inlet and outlet water are measured using mercury in glass thermometers reading to 1^{o}C. The pressures of the air at the inlet and outlet of the test section are measured with the water manometer reading to 1mm.

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The experiment for determining heat transfer coefficient for water flowing in corrugated ducts utilizes the close loop system shown in the upper diagram of Figure 1, hot water first enters the system from the water tank and flows over and under the air channel. The water exits the downstream plenum chamber and empties into a weight tank (not shown), to determine the mass flow rate only. Each system component will now be discussed in greater detail, starting with the heart of the system.
Water flow through the test section was maintained by a water pump which has water heated with the help of water heater and the hot water is circulated throughout the system, from the water tank. A rotameter is used to calculate the mass flow rate of air and valves are employed at the starting for regulating the flow of water and a pump is used to circulate the hot water to the system. There are two thermometric wells, at the water inlet and water outlet to know the temperatures of water at inlet and outlet.
Air flow system consists of a compressor, rotameter and a plenum chamber. The air is taken from the compressor and then supplied in the plenum chamber through the rotameter which helps to determine the flow rate and maintain the flow rate at a fixed value. The air in the plenum chamber gets heated with the hot water and hot air is at the outlet of the test section [21].

3. Results and discussion

Heat loss to the surroundings = heat given by hot fluid (water) – heat taken by cold fluid (air).

$$H_l=m_hC_h(T_{hi}-T_{ho})-m_cC_c(T_{co}-T_{ci}) \quad (1)$$

The effectiveness of the PHE is calculated from experimental observations.

$$\epsilon=C_h(T_{hi}-T_{ho})/[C_{min}(T_{hi}-T_{ci})] \quad (2)$$

Neglecting heat transfer along the plates, the heat transfer rates for the two fluids across the elementary slant element of area dA_s is given by;

$$(m_h/2)C_{ph}[T_x-T_{x+dx}]=h_x.(T_b-T_w).dA_s, \text{for water} \quad (3)$$

and

$$m_cC_{pc}[T_{x+dx} - T_x] = 2.h_x.(T_w-T_b).dA_s, \text{ for air} \quad (4)$$

where, $T_b = [T_x + T_{x+dx}] /2$; $dA_s= W.\,dl$

Therefore, the local heat transfer coefficient can be written in non-dimensional form as:

$$Nu_x = h_x.D_h/k \quad (5)$$

Friction factor f for air air channel is found from the following relationship:

$$f= \Delta p/[(L/D_h).\ (G^2/2.\rho.g_c)] \quad (6)$$

Mass velocity, flow area, perimeter and hydraulic radius of a fluid channel are given by
$G = m/A_f$; $A_f = H.W$; $P= 2(W+H)$ and $D_h = 4.A_f/P$, respectively.

It may be noted that H and W are same for the two fluid channels in the experimental setup. The Reynolds number, based on hydraulic radius is given by:
$Re = G.D_h/\mu$

Focke correlations for corrugated plates with plate corrugation angle $\beta = 30^o$ are:
$j_{cp} = 0.2334\ Re^{-0.297}$ (for $1000<Re<5\times10^4$)
$j_{cp} = 0.80\ Re^{-0.477}$ (for $100<Re<1000$)
$f_{cp} = 0.287\ Re^{-0.147}$ (for $3000<Re<10^4$)

Taking into account the variation of viscosity with temperature, Nusselt number and friction factor needed for determining heat transfer coefficient h and hydraulic frictional pressure drop for fluids are calculated from the equations given below:

$$Nu_m = j_{cp}.Re.Pr^{1/3}(\mu_w/\mu)^{-0.2} \quad (7)$$

$$f = f_{cp}.(\mu_w/\mu)^{0.58} \quad (8)$$

The Dittus-Bolter equation for a fully developed turbulent flow in a smooth circular tube with constant wall temperature condition, $0.7\leq Pr\leq 160$, Re> 10000 and L/D> 10 is given by [4]:
$Nu_m = 0.023\ Re^{0.8}\ Pr^n$, n=0.3 for fluid cooling and 0.4 for fluid heating. It may be noted that in case of channel flows, the limitation of Reynolds number would be quite lower as already discussed.
The Brien and Sparrow [13] correlation for water is given by $Nu=0.409Re^{0.614}\ Pr^{0.34}$ in the range from 1500 to 25000 and Prandtl number from 4 to 8 for flow in a triangular wavy channel with corrugation angle 30°. The cooper and Usher [18] gave the correlation $Nu=0.4Re^{0.64}\ Pr^{0.4}$ for small size chevron plates.
Experiments were conducted with Reynolds number varying between 16900 to 68000 for air and 750 to 3200 for water by varying the flow rates of the fluids. Nusselt number and friction factor were then determined.
The observations for the typical combination of air- and water-flow rates for counter flow are given in Table 1. Additional observations are the average wall temperatures, from right to left, at three locations which are found as 64.75°C, 63.69°C, 59.98°C. The heat loss to the surroundings is found as 8.6×10^{-3} W for the typical case.

Table 1. Observations for a typical combination of air- and water-flow rates for counter flow

Air Channel (air flow rate = 0.1128 kg/s) Manometric level difference(Δz_i-Δz_o)= 1.5 cm of H_2O			Water Channel (water flow rate per channel = 0.1220 kg/s)		
Distance from left end(x) mm	Local Temperature (T_x) °C	Bulk Temperature (T_b) °C	Distance from right end(x) (mm)	Local Temperature (T_x) °C	Bulk Temperature (T_b) °C
0	44.000		0	71.000	
40	44.950	44.475	70	70.549	70.774
140	49.775	47.362	170	70.352	70.451
180	51.625	50.700	270	70.301	70.326
280	52.625	52.125	360	70.235	70.268
320	54.450	53.537	410	69.784	70.009
420	54.750	54.600	510	69.745	69.764
460	55.975	55.362	610	69.509	69.627
560	60.375	58.175	700	69.000	69.254
660	60.625	60.500			
700	61.000	60.812			

The variation of local heat transfer coefficient along channel length (x/D_h) is shown in Figure 3 for water and Figure 4 for air. For both the fluids, it is seen that h_x attains a local peak at the individual crests of the channel surface. This result is very different from that observed in a straight duct. Further, the maximum value occurs at a crest in the vicinity of $x/D_h=8$, for all Reynolds numbers involved, where the irreversibility is expected to be the least as the temperature difference between donor and recipient is least. This result is very different from that observed in the straight duct, in which case the h_x converges to a constant value within the fully developed region.

3.1 Average Nusselt number

The mean heat transfer coefficient for air varied from 162.73 W/m^2K to 204.18 W/m^2K and hence Nu_m varied from 388 to 500. Corresponding values were 633.42 W/m^2K, 2029.17 W/m^2K, 65 and 210 for water. The variation of average Nusselt number, Nu_m with Reynolds number is given in Figures 3 and 4 for the two fluids. The experimental data has been compared with those calculated using the Dittus-Bolter equation for constant wall temperature for smooth tubes, and Brien & sparrow [13], Cooper and Usher [18] and Focke [1] correlations for corrugated plates. While comparing with Focke [1] correlations it is assumed that the correlations for Nu_m and f which are valid up to Re= 50000 and 10000, respectively hold good for values in the present case. Comparison of experimental and theoretical mean Nusselt numbers with Reynolds number for water is shown in Figure 5. The experimental results have the same trend as the calculated values for air from using the Dittus-Bolter equation for constant wall temperature and Brien & sparrow, Cooper and Usher and Focke correlations for corrugated plates equation. However, the experimental Nusselt number is highest at a given Reynolds number. It means that none of the existing correlations is able to predict real Nusselt number for the case under consideration. It is worthwhile to find a correlation for the case under consideration for Nusselt number with Reynolds number for water. The correlation is $Nu=0.247Re^{0.83}$. Such a variation is drawn for air in Figure 6 for air where the correlation found is $Nu=66.686Re^{0.18}$.

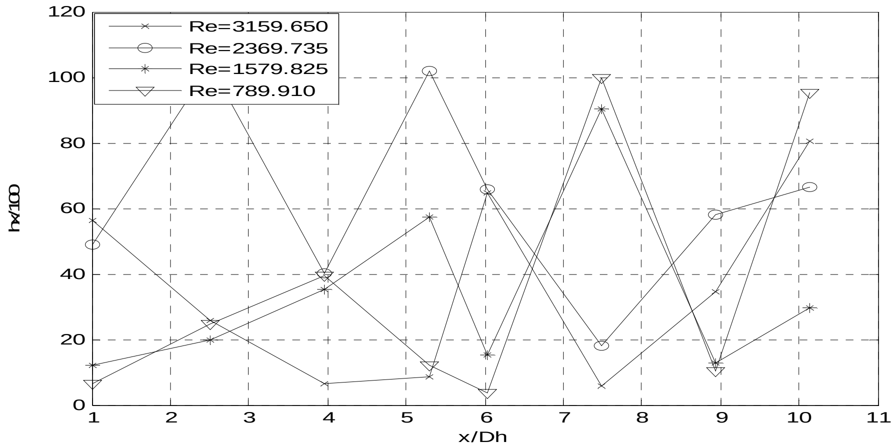

Figure 3. Variation of h_x along channel length (For water)

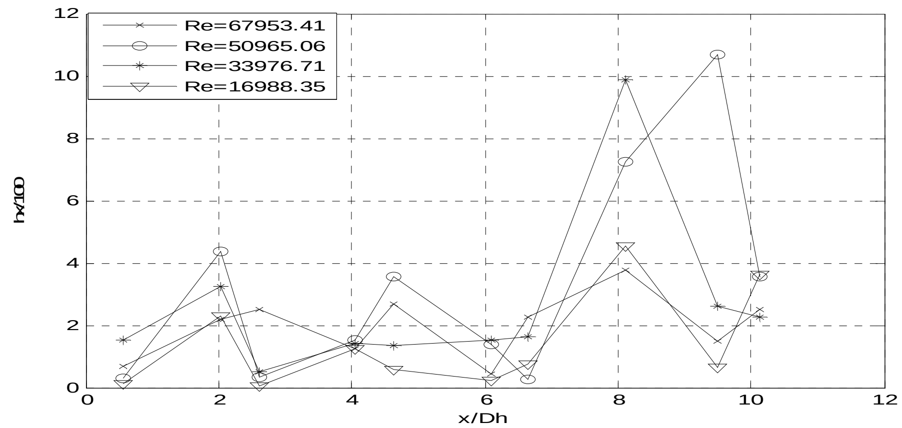

Figure 4. Variation of h_x along channel length (For Air)

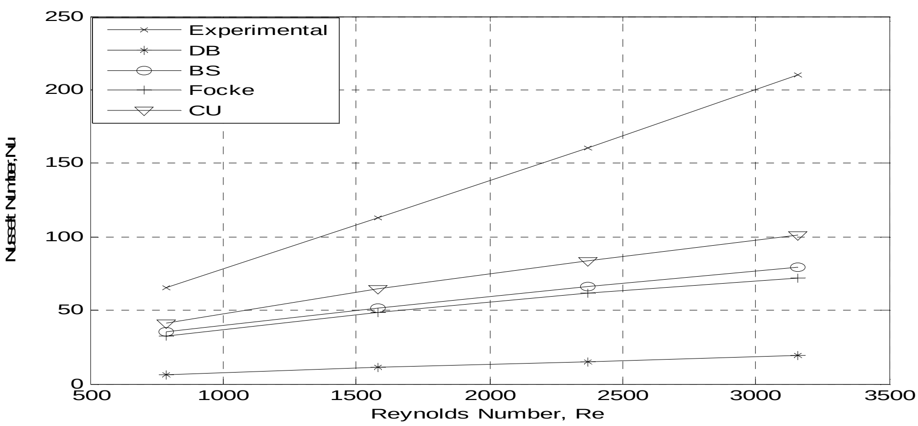

Figure 5. Comparison of experimental and theoretical Nusselt numbers (For Water)

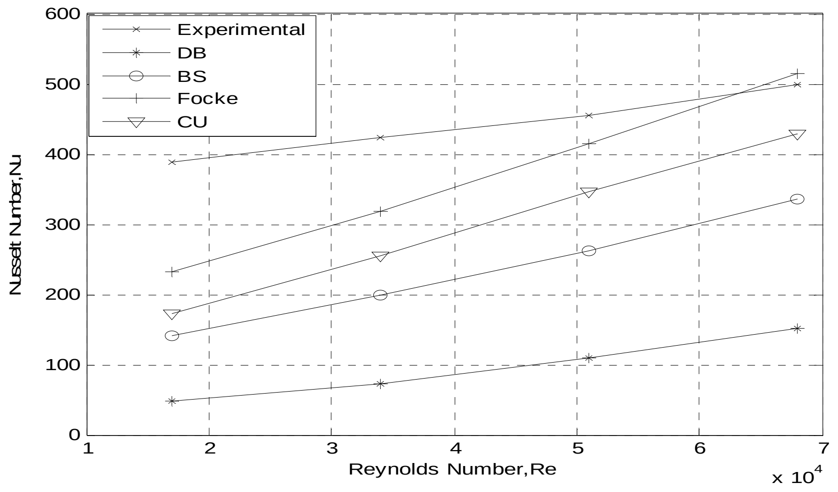

Figure 6. Comparison of experimental and theoretical Nusselt numbers (For Air)

3.2 Friction factor

Experimental friction factor was compared with the values obtained using Focke & Kumar's correlations and the comparison is shown in Figure 7. Theoretical friction factors using Kumar's empirical correlation [19] are also included in this figure.

Figure 7 shows that the friction factor (f) decreases with increase in Reynolds number because it is inversely proportional to square of velocity and hence of Reynolds number. The values of f for the experiment are closer to Focke correlations and much lower than Kumar correlation. The friction factor varied from 0.0803 to 0.1048 for air. The correlation for the case under consideration for friction factor, using experimental values, is found as $f = 0.644 / Re^{0.18}$.

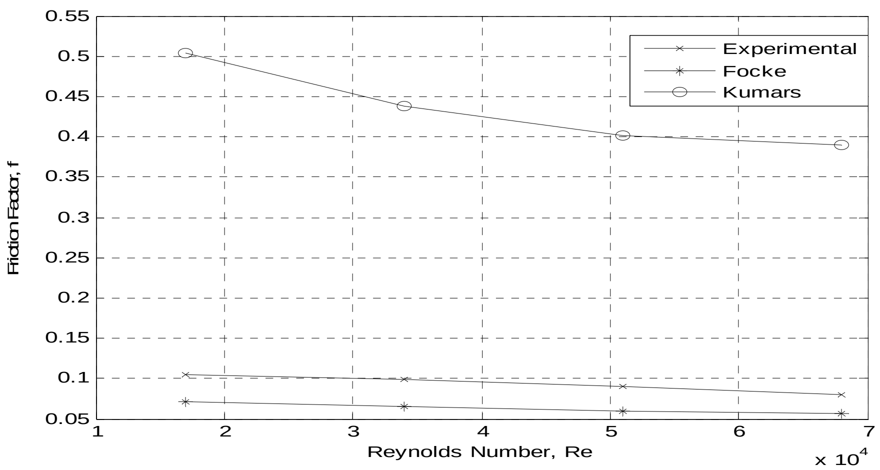

Figure 7. Comparison of experimental and theoretical friction factor (For Air)

3.3 Effectiveness

Effectiveness of this experimental corrugated PHE was found as 82%.

4. Conclusion

The test section was formed by three identical corrugated channels having corrugation angle of 30^o with cold air flowing in the middle one and hot water equally divided in the adjacent channels. Sinusoidal wavy arcs connected with tangential flat portions make the corrugation angle with transverse direction. The Reynolds number based on hydraulic diameter varied from 750 to 3200 for water and from 16900 to 68000 for air the inlet Prandtl numbers for both the fluid were approximately constant. Experimental investigations are done by changing the mass flow rate of air and water. The pressure drop and heat transfer characteristics in the corrugated channel of a plate heat exchanger were found. Dimensionless correlations for Nu_m and friction factor have been developed for both air and water, for volume flow rates 100 lps, 75 lps, 50 lps, 25 lps of air and mass flow rates 0.244 kg/s, 0.1533 kg/s, 0.1022 kg/s, 0.0511 kg/s of water, based on experimental results. It is found that Nu_x depends on Re and x/D_h. Friction factor was found to depend on Re. The magnitudes of the Nu_m and friction factors are high when compared to conventional pipe flows. The mean heat transfer coefficient varied from 162.73 W/m^2K to 204.18 W/m^2K and hence Nu_m varied from 388 to500 for air. Corresponding values were 633.42 W/m^2K, 2029.17 W/m^2K, 65 and 210 for water. Correlations found for Nusselt numbers are $Nu_m= 0.247Re^{0.83}$ for water and $Nu_m= 66.686Re^{0.18}$ for air. Similarly the friction factor varied from 0.0803 to 0.1048 for air and the correlation for friction factor is found as $f = 0.644 / Re^{0.18}$. Effectiveness of this experimental corrugated PHE is found as 82%.

Nomenclature

A	Area, m^2	p	Pressure, Pa
C_p	Specific heat at constant pressure, kJ $kg^{-1}K^{-1}$	p	Wetted perimeter, m
D_h	Hydraulic diameter, m	Pr	Prandtl number
f	Friction factor	Q	Heat transfer rate, kW
H	Height of flow channel, m	Re	Reynolds number
h	heat transfer coefficient, W m^{-2} K^{-1}	T	Temperature, K
j	Colburn factor	W	Channel width, m
k	Thermal conductivity, W m^{-1} K^{-1}	z	Manometric height, m
L	Projected length of the channel, m	μ	Viscosity of air, N s m^{-2}
l	Developed length of the channel, m	β	Corrugation angle, deg.
m	Mass flow rate, kg s^{-1}	ρ	Density, kg m^{-3}
Nu	Nusselt number	ϵ	Effectiveness

Suffixes

b	Bulk	m	Mean value
c	cold fluid	o	Outlet
cp	Constant property	s	Surface
h	Hot fluid	w	Wall
i	inlet	x	Location along the channel from inlet of the fluid

Prefix

d,δ Elemental

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Shive Dayal Pandey is a research scholar in the department of Mechanical Engineering at MNNIT, Allahabad, Uttar Pradesh (India) since 2008. He has completed M.Tech (thermal engineering) degree with honours from department of Mechanical Engineering, IT, BHU Varanasi, (U.P), India,. He has published five papers in International and National conferences and journals. He has good knowledge in thermal engineering area of plate heat exchangers. Mr. Pandey is the member of a numbers of professional societies and organizations like IAENG, IEEE, Institution of engineers (India) etc.
E-mail address: pandeymnnit@gmail.com; rme0809@mnnit.ac.in

V.K.Nema is a Professor of Mechanical Engineering at MNNIT, Allahabad (U.P), India with his service for more than 30 years. He did his B.E and M.E. both from Govt. Engineering College Jabalpur (M.P) India and Ph.D degree from IIT, Kanpur (U.P), India. He has published a number of papers in International and National conferences and journals and is a reviewer of different journals. Dr. Nema is a life member of professional societies, namely ISTE and Institution of engineers (India).
E-mail address: vknema@mnnit.ac.in