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Review of Nusselt Number Correlation for Single Phase Fluid Flow through a Plate Heat Exchanger to Develop C# Code Application Software

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ABSTRACT : *Plate heat exchanger has been widely used in heating, cooling application, food industry, chemical industry, refrigeration industry and marine application. The objective of this paper is to review, the Nusselt number correlations of plat heat exchanger available to date, to understand how it is used, to evaluate the convective heat transfer coefficient, overall heat transfer coefficient and the methodology required to evaluate the correlations. In spite of their long history of commercial use there is still a lack of reliable data and generalized solutions available in the literature for calculation of heat transfer. Number of computer aided design software has been developed by the manufacturers of PHE but the information about the heat transfer coefficients is normally compressed and not a lot of data are available for research purposes about the design of these heat exchangers. The paper is useful to understand modified Wilson plot technique to evaluate the Nusselt number correlation of single phase fluid flowing through PHE.C# code application software is developed to do the calculation of experimental setup readings given by some authors and compared. Results shown by C# code application software (+/-) 5% in error. Water property data based is taken from the NIST Reference Fluid Thermodynamic and Transport Properties, RefpropV-8.*

Keywords – *Chevron Angle, Modified Wilson Plot, Nusselt Number, Plate Heat Exchanger (PHE), Single Phase Fluid*

1. INTRODUCTION

The Many efforts have been taken in the past to experimentally investigate the heat transfer characteristics of plate heat exchangers. Data have been published for different types of chevron, herringbone and wash board plate heat exchangers. The use of plate heat exchangers has increased considerably in the past two decades, and they are now accepted as standard heat transfer equipment in a broad range of application, operating both, in single and two-phase flow regimes. The plate heat exchangers are widely used in warming, heating, cooling applications, food, cosmetic and Chemistry Industry. The plate type heat exchangers are , initially developed for the pasteurized liquid food domain, which mostly requires hygienic application. But, these heat exchangers have a large application area in Chemistry and Food Sector because of being compact and having the quality to be easily cleaned. Generally, they are characterized by larger heat transfer area to volume ratio, lighter weight, design flexibility, high thermal effectiveness, hence they are suitable for energy and space saving. Their design flexibility provides an advantage in varying heat transfer area by easily adding or removing plates without disturbing the piping connections. Being compact in nature, the plate heat exchangers have better heat transfer characteristics, however, may have higher pressure drop and fouling issues especially in brazed type PHEs. Therefore, for wider engineering applications experimental data are required for both heat transfer and pressure drop characteristics of the plate heat exchangers. In spite of their long history of commercial use (since1960), there is still a lack of reliable data and generalized solutions available in the literature for calculation of heat transfer and pressure drop. Most of the research and experimental study have been taken place on plate heat exchanger for different chevron angle ranging from $20< \beta < 65$. Result shows large mutual discrepancies even when comparing specific chevron angle β, recognized as the most influencing geometric parameter for determining the heat transfer coefficient and pressure drop. Recently, plate heat exchangers are commonly used in comparison of other types of heat exchangers such as shell and tube type in heat transfer processes because of their compactness, ease of production, sensitivity, easy care after set-up and efficiency. Flow of the substances to be heated and cooled, takes place between alternating metal sheets and allowing heat transfer between the fluids. Gaskets are placed in between plates to avoid mixing of the fluids. This paper is focused on to understand the heat transfer characteristics and methodology used for development of Nusselt number correlation of chevron angle plate heat exchanger for single phase fluid flow.

2. EVALUATING PARAMETER OF PHE

2.1 Reynolds Number

In the case of the plate heat exchangers, the hydraulic diameter is very small, it is almost in order of mm, therefore, the turbulent conditions are achieved very early i.e., at a very low value of Reynolds number. Simpson [1] reported that the turbulent condition can be achieved at Reynolds numbers as low as 150. Reynolds number is a function of fluid flow rate. The mathematical formula for Reynolds number is given in (2.1). The numerator is a mass flow per unit area times a velocity i.e. a momentum flow per unit area. The denominator is a viscous stress i.e. a viscous force per unit area. The ratio represents momentum to viscous forces. If viscous forces dominate, the flow will be laminar and if momentum dominates, the flow will be turbulent.

$$Re = G\, D_h / \mu \qquad (2.1)$$

2.2 Prandtl Number

Prandtl number is function of two important physical properties (thermal and momentum), therefore, responsible for the growth of the boundary layers and the relative thickness between them. The thermal diffusivity (α) and the momentum diffusivity (ν), the ratio of these two quantities is the well-known Prandtl number and mathematically it is given in (2.2). The Prandtl number may be seen to be a ratio of the rate that viscous forces penetrate the material to the rate that thermal energy penetrates the material.

$$(Pr) = (\mu / \rho)\, (k / Cp\, \rho) = (\mu\, Cp / k) = (\nu / \alpha) \qquad (2.2)$$

2.3 Flow Velocity

In the corrugated channel, the actual flow would most likely follow the corrugations rather than flow in vertical direction, given by Focke [2]. As a result, the actual flow velocity is much higher than the mean value in the vertical direction, which is used to calculate the Reynolds number. PHE could be treated as pure co-current or counter-current heat exchangers, in principle, if end effects are neglected. For a shell-and-tube exchanger this can hardly be the case, due to cross flows resulting from baffles. For a two-channel PHE, pure counter-current flow may be assumed and for multiple channel units, a correction factor of the LMTD is recommended sometime.

2.4 Nusselt Number

The Nusselt number may be physically described in case of plate heat exchanger as given in (2.3). Nusselt number is equal to the dimensionless temperature gradient at the surface and it essentially provides a measure of convective heat transfer. The Nusselt number may be viewed as the ratio of the conduction resistance of a material to the convection resistance of the same material. The denominator of the Nusselt number involves the thermal conductivity of the fluid at the solid-fluid convective interface.

$$N_u = (h \times L) / k_{fluid} = (h \times D_h) / k_{fluid} \qquad (2.3)$$

2.5 Nusselt Number Correlations

In single phase fluid flow heat transfer, generally N_u is represented by a pragmatic expression in the form of as given in (2.4). C_1, m, C_3 does not depend upon the nature of fluid used. The term $(\mu/\mu_w)^{C4}$ is accountable for variable viscosity effect.

$$N_u = C_1\, Re^m\, Pr^{C3}\, (\mu/\mu_w)^{C4} \qquad (2.4)$$

2.6 Overall Heat Transfer Coefficients

It is most convenient to use overall heat transfer coefficients in heat transfer calculations, as it combines all of the constituent factors into one and are based on the overall temperature drop as given in (2.5)

$$1/U = 1/h_h + t/k_{wall} + 1/h_c \qquad (2.5)$$

3. TEST FACILITY

A literature survey has been performed to examine the experimental set-ups of single phase fluid flow through a corrugated plate heat exchanger and method for developing Nusselt number (N_u). Table 1 and Table.2 includes the Nusselt number correlation developed by most of the Authors to date. Authors like T.S. Khan[3],IulianGherasim[4],MinsungKim[5],F.Akturk[6],AliHashmi[7],W.S.Kuo[8], GiovanniA.Longo[9], Jaekyoo Jang[10] had selected equipments for experimental set up as given below. They installed and used at various Laboratories by the respective authors to perform the single phase fluid flow experimentation on plate heat exchanger. Generalized schematic of the single-phase experimental loops of plate heat exchanger are shown in "Fig.1"

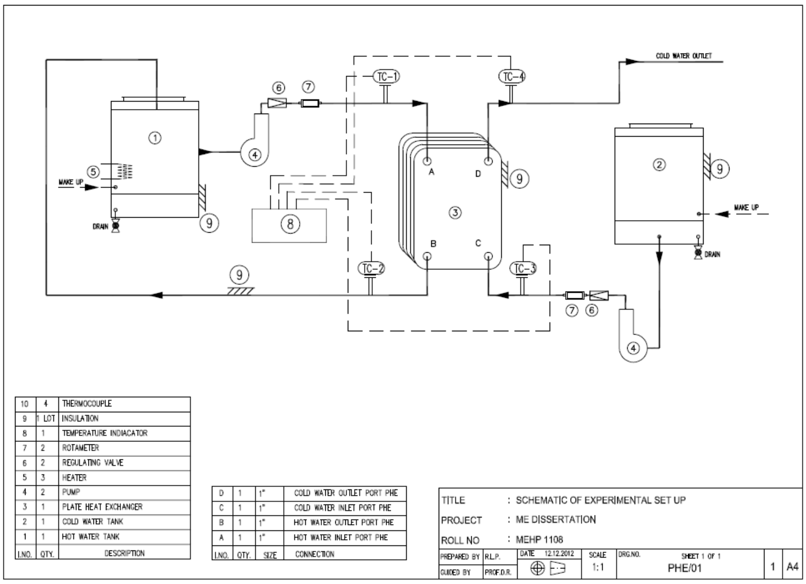


Fig.1 Schematics of Single Phase Fluid Flow Loop Experimental Setup of Plate Heat Exchanger

TABLE.1 NUSSELT NUMBER CORRLEATION WITHOUT VISCOSITY SUMMERY

Plate Heat Exchangers Nusselt Number CorrelationFor Different Chevron Angle										
Sr.No.	Author	Fluid	Correlation	ϕ	Angle(β)	R_e	Pr	C1	p	C3	C4
	Study do not include Chevron Angle and variable viscosity effect										
1	Maslov and Kovalenko[11]	Water	$Nu= C1\, R_e^{p}\, Pr^{C3}$		60	0.1<Re<15		0.63	1/3	1/3	0
					60	50<Re<20000		0.78	0.5	1/3	0
2	Edwards[12]	Water	$Nu= C1\, R_e^{p}\, Pr^{C3}$	1.18	NM	Re<10		NM	1/3	1/3	0
				1.18	NM	Re>200		NM	0.7	1/3	0
3	Talik[13]	Water	$Nu= C1Re^{p}\, Pr^{C3}$	1.22		1450<Re<11460	2.5<Pr<5	0.248	0.7	0.4	0
				1.22		10<Re<720	70<Pr<450	0.2	0.75	0.4	0
4	Okada[14]	Water	$Nu= C1\, R_e^{p}\, Pr^{C3}$	1.147	30	700<Re<20000		0.1528	0.66	0.4	0
				1.294	45	700<Re<20000		0.2414	0.64	0.4	0
				1.412	60	700<Re<20000		0.3174	0.65	0.4	0
				1.412	75	700<Re<20000		0.4632	0.62	0.4	0
5	Thonon[15]	Water	$Nu= C1\, R_e^{p} Pr^{C3}$		30	50<=Re<=15000		0.2946	0.7	1/3	0
					45	50<=Re<=15000		0.2998	0.645	1/3	0
					60	50<=Re<=15000		0.2267	0.631	1/3	0
6	Focke[2]	Water	$Nu= C1\, R_e^{p} Pr^{C3}$	1.464							
					45	45<Re<300		1.67	0.44	0.5	0
						300<Re<2000		0.405	0.7	0.5	0
						2000< Re <20,000		0.84	0.6	0.5	0
					60	120<Re<1000		0.77	0.54	0.5	0
						1000<Re<42000		0.44	0.64	0.5	0
					30	120<Re<150		1.89	0.46	0.5	0
						150<Re<600		0.57	0.7	0.5	0
						600<=Re<=16000		1.112	0.6	0.5	0
7	Emerson[16]	Water	$Nu= C1\, R_e^{p}\, Pr^{C3}$		NM	10 <Re <25		0.76	0.48	1/3	0
					NM	40 <Re <1000		0..52	0.61	1/3	0
8	Rosenblad and Kullendroff[17]	Water	$Nu= C1\, R_e^{p} Pr^{C3}$	1.21	60	60< Re <2415		0.289	0.697	1/3	0
9	Roetzel[18]	Water	$Nu= C1\, R_e^{p} Pr^{C3}$		70	400<Re<2000		0.371	0.703	1/3	0
10	Rene[19]	Water	$Nu= C1Re^{p}\, Pr^{C3}$		60	1<Re<5.5		0.59	1/3	1/3	0
					60	5.5<Re< 4500		0.352	0.639	1/3	0
11	Longo and Gasparella[09]	Water	$Nu= C1\, R_e^{p} Pr^{C3}$		65	200<Re< 1200	5<Pr<10	0.277	0.766	0.333	0
12	Aydm Drumus[20]	Water Hot Side	$Nu= C1Re^{p}\, Pr^{C3}$								
						500<Re<1000	3<Pr<7	0.0488	0.864	1/3	0
						500<Re<1000	3<Pr<7	0.0443	0.8709	1/3	0

TABLE.2 NUSSELT NUMEBER CORRLEATION WITH VISCOSITY SUMMERY

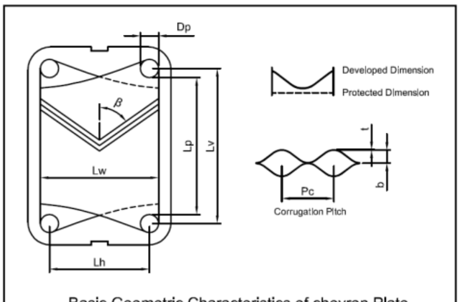
Plate Heat Exchangers Nusselt Number CorrelationFor Different Chevron Angle											
Sr.No.	Author	Fluid	Correlation	ϕ	Angle(β)	R_e	Pr	C1	p	C3	C4
	Study do not include Chevron Angle But Include variable viscosity effect										
1	Kumar[21]	Water	$Nu= C1\ R_e^{\ p}\ Pr^{C3}\ (\mu/\mu_w)^{C4}$		30	Re>10		0.348	0.663	0.33	0.17
					45	10<Re<100		0.4	0.598	0.33	0.17
					60	20<Re< 400		0.306	0.529	0.33	0.17
2	Warnakulasuriya and Worek[22]	Salt Solution	$Nu= C1\ R_e^{\ p}\ Pr^{C3}\ (\mu/\mu_w)^{C4}$			250<Re<1100	82<Pr<174	0.292	0.705	0.35	0.14
3	F. Akturk ,G. Gulben[6]	Water	$Nu= C1Re^p\ Pr^{\ C3}\ (\mu/\mu_w)^{C4}$		30	450<Re<5250		0.3259	0.6125	1/3	0.14
4	M.S.Khan[3]	Water	$Nu= C1Re^p\ Pr^{\ C3}\ (\mu/\mu_w)^{C4}$								
				1.117	60	500<Re<2500	3.5 <Pr<6	0.1449	0.8414	0.35	0.14
				1.117	30	500<Re<2500	3.5 <Pr<6	0.1368	0.7424	0.35	0.14
				1.117	30/60	500<Re<2500	3.5 <Pr<6	0.1437	0.781	0.35	0.14
5	Ali Hashmi, Fraaz Tahir[7]	Water	$Nu= C1Re^p\ Pr^{\ C3}\ (\mu/\mu_w)^{C4}$	1.12	45	500<Re<4500	5.6<Pr<8.0	0.0566	0.881	0.33	0.14
6	Amar Jokar[23]	Water	$Nu= C1Re^p\ Pr^{\ C3}\ (\mu/\mu_w)^{C4}$		60/60	1700<Re<6500		0.134	0.712	0.33	0.14
					60/27	1400<Re<5600		0.214	0.698	0.33	0.14
					27/27	950<Re<4500		0.24	0.724	0.33	0.14

3.1 Chevron Plate Heat Exchanger

The major equipment used by most of the authors to carry out experimentation was the plate heat exchanger. The geometrical features of commercial plate heat exchanger used by T.S. Khan [3], Amir Jokar [23], Iulian Gherasim [4] and Ali Hashmi [7] are given in Table.3. Fig.2 has a typical specific plate geometry dimension. Plates consist of fixed chevron angle profiles. Each plate within an exchanger is formed with a corrugated angle configuration. Two plates form a channel through which a fluid flows, and the inclination angles of two neighboring plates are oriented in opposite directions.

TABLE.3 PLATE GEOMETRY FEATURES

Sr. No.	Particulars	Notation
1	Plate Width	L_w (mm)
2	Vertical Distance Between Centers of Ports	L_v (mm)
3	Horizontal Distance Between Centers of Ports	L_h (mm)
4	Channel Spacing	b (mm)
5	Effective Area	A_x (m^2)
6	Plate Thickness	t (mm)
7	Surface Enlargement Factor	Φ
8	Chevron Angle	B
9	Port Diameter	D_p(mm)
10	Corrugation Pitch	P_c(mm)



Basic Geometric Characteristics of chevron Plate

Fig.2 Basic Geometric Characteristics of Chevron plate

3.2 Fluid Flow Measurement

To measure accurately, water flow rates of hot water and cold water, Author’s Amir Jokar [20], used turbine flow meter. Whereas Variable frequency drive was installed on pump by T.S. Khan [3].

3.3 Temperature Readings Devices

Temperature was measured accurately at the inlet and outlet ports of the plate heat exchanger by using RTD PT-100 by T.S. Khan[3], Ali Hashmi [7] and 4-wire RTD as well as T-type thermocouple by Amir Jokar[23], Jaekyoo Jang [10], F. Akturk[6], Iulian Gherasim[4],with an accuracy of (+/- 0.5 & 0.65 %).

3.4 Water Pump. In order to supply hot water and cold water in plate heat exchanger loop, the variable speed water pump was selected by T.S. Khan [3] and Water Pump was selected by Amir Jokar [23] for their experiment.

4. SINGLE PHASE ANALYSIS

The experimental result analysis used by Jose Fernandez [24], W.S Kuo [8], R.U.Yang [25], Ali Hashmi [7], F. Akturk [6], Minsung Kim [5], Iulian Gherasim [4], Jaekyoo Jang [10], T.S. Khan [3], and GiovanniA.Longo [9] is briefly stated in this section. The so-called 'Modified Wilson-Plot' technique is commonly accepted as the preferred method for interpreting heat transfer performance data for Liquid-Liquid and refrigerant-to-air heat exchangers. As the first step in analyzing the collected data, properties of water for each test point were calculated at bulk temperatures averaged between the inlet and outlet ports on each side. Each property, including density, specific heat, conductivity, and viscosity, was evaluated and correlated, using third order logarithmic regressions. These properties were then used to find the flow characteristics in the channels for each plate. The Reynolds number of the flow within the channels was calculated by (2.1), where the hydraulic diameter was defined as two times the average plate spacing $D_h=2b$, and the mass flux was calculated based on the minimum free flow area (A_0) between the plates, as described by Shah and Wanniarachchi[26].

$$G = \rho V^* / A_0 \tag{4.1}$$

Due to the complicated geometries that gasketed plate heat exchangers contain, this minimum free flow area is difficult to estimate and has not been universally standardized, however for the sake of simplicity; many studies have considered this free flow area as the average plate spacing, b, multiplied by the width of the plate.

$$A_0 = b \text{ X } w \tag{4.2}$$

It is noteworthy that the minimum free flow area between the two neighboring plates, which depends on the corrugation angle, is much less than the area given by (4.2). A more thorough description of the minimum free flow area is given in a previously published paper by the authors, Hayes and Jokar [23], while (4.3) was used for data reduction in this study. An energy balance was applied in order to obtain heat transfer rates on both hot and cold sides.

$$Q^* = m^* \, Cp \, \Delta T \tag{4.3}$$

Using the log- mean temperature differene,

$$\Delta T_{lm} = [(T_h \text{ in} - T_c \text{ out}) - (T_h \text{ out} - T_c \text{ in})] / \ln [(T_h \text{ in} - T_c \text{ out}) - (T_h \text{ out} - T_c \text{ in})] \tag{4.4}$$

The overall heat transfer coefficient in the heat exchanger was calculated by (4.5)

$$U= Q^*/ (A_x \, \Delta T_{lm}) \tag{4.5}$$

where A_x is the effective heat transfer surface area, which was calculated by the projected heat transfer area multiplied by the enlargement factor. A common analysis method of single-phase heat transfer in gasketed plate heat exchangers is the modified Wilson plot technique. Due to the possibility of large property variations, the heat transfer correlations format was chosen similar to, De-witt [27] and Amir Jokar [23].

$$N_u = C \, R^pe \, Pr^{\,1/3} (\mu/\mu_w)^{0.14} \tag{4.6}$$

The heat transfer coefficients for the hot and cold sides of the gasketd plate heat exchanger are thus result into (4.7) and (4.8)

$$h_c = (k_c/D_h) C_c \, Re^p{}_c \, Pr_c^{\,1/3} (\mu/\mu_w)_c^{0.14} \tag{4.7}$$

$$h_h = (k_h/D_h) C_h \, Re_h \, Pr_h^{\,1/3} (\mu/\mu_w)_h^{0.14} \tag{4.8}$$

Plate geometries in gasketed plate heat exchangers are so complex and varied among different manufacturers, the flow regimes cannot be assumed like the in-tube flow. However, due to the similarity in geometries and configurations on the cold and hot sides of a gasketed plate heat exchanger, the flow regimes and the Reynolds number exponents on both sides can be assumed identical at any given Reynolds number. This flow assumption is safe even for very different fluids flowing in the exchanger because the fluid properties are taken into account with Prandtl number and viscosity ratio portions of the mathematical relations. Prandtl number exponent and viscosity ratio exponents, which account for the different fluid properties of water, can also be assumed constant at 1/3 and 0.14, respectively. The original Wilson plot technique requires data to be recorded at constant flow rates and constant average bulk fluid temperatures on both hot and cold sides, which is not easily accomplished. However, Modified Wilson plot technique, derived by Briggs and Young [28], allows data to be taken at varying flow rates and varying bulk fluid temperatures on hot and cold sides. The overall heat transfer equation based on this method is obtained through the following thermal resistance equation

$$(1/U)-(t/k)_{wall} =\{ (1/ C_c(k_c/D_h) R_e{}^p{}_c Pr_{rc}{}^{1/3}(\mu/\mu_w)_c{}^{0.14})\} + \{(1/C_h(k_h/D_h) R_e{}^p{}_h Pr_h{}^{1/3}(\mu/\mu_w)_h{}^{0.14})\} \quad (4.9)$$

Multiplying both sides of (4.9) by

$$(k_c/D_h) Re^p{}_c Pr_c{}^{1/3}(\mu/\mu_w)c^{0.14}) \quad (4.10)$$

It gives to,

$$\{(1/U) - (t/k)_{wall} \}x\{(k_c/D_h) Re^p{}_c Pr_c{}^{1/3}(\mu/\mu_w)_c{}^{0.14})\} =[1/C_c]+$$
$$\{[(k_c/D_h) Re^p{}_c Pr_c{}^{1/3}(\mu/\mu_w)_c{}^{0.14})] / [C_h(k_h/D_h) Re^p{}_h Pr_h{}^{1/3}(\mu/\mu_w)_h{}^{0.14}]\} \quad (4.11)$$

This is in the form of ,

$$Y_1 = m X_1 + b \quad (4.12)$$

Where

$$Y_1 =\{(1/U) -(t/k)_{wall} \} x \{(k_c/D_h) Re^p{}_c Pr_c{}^{1/3}(\mu/\mu_w)_c{}^{0.14})\} \quad (4.13)$$

$$X_1 = \{[(k_c/D_h) Re^p{}_c Pr_c{}^{1/3}(\mu/\mu_w)_c{}^{0.14})] / [(k_h/D_h)Re^p{}_h Pr_h{}^{1/3}(\mu/\mu_w)_h{}^{0.14}]\} \quad (4.14)$$

Slope: $m= (1/C_h)$ (4.15)

Intercept: $b= (1/C_c)$ (4.16)

4.1 Logarithmic Modification

Given below is the logarithmic modification of (4.9)

$$(1/U)-(t/k)_{wall} =\{(1/C_c(k_c/D_h)Re^p{}_cPr_c{}^{1/3}(\mu/\mu_w)_c{}^{0.14})\}+\{(1/C_h(k_h/D_h)Re^p{}_h Pr_h{}^{1/3}(\mu/\mu_w)_h{}^{0.14})\}$$

It will become

$$(1/U)-(t/k)_{wall}-\{(1/C_h(k_h/D_h)Re^p{}_h Pr_h{}^{1/3}(\mu/\mu_w)_h{}^{0.14})\}=\{(1/C_c(k_c/D_h) Re^p{}_c Pr_c{}^{1/3}(\mu/\mu_w)_c{}^{0.14})\}$$

$$(1/U) - (t/k)_{wall} -(1/C_h(k_h/D_h)Re^p{}_h Pr_h{}^{1/3}(\mu/\mu_w)_h{}^{0.14})x Pr_c{}^{1/3}(\mu/\mu_w)_c0.14x(k_c/D_h)=1/(C_cRe^p{}_c) \quad (4.17)$$

Taking log on both sides,(4.17) becomes

$$\ln y2 = - \ln Cc - p \ln Re c \quad (4.18)$$

$$Y2 = \ln y2 \quad (4.19)$$

$$X_2 = \ln Re_c \quad (4.20)$$

Slope: (p), Intercept: $(- \ln C_c)$

4.2 Iterative Procedure

Since the viscosity ratio groups and the Reynolds number exponents undergo a mathematical relaxation method with the fluid flow rates and temperatures, successive linear regressions can be performed to execute the nonlinear regression that these equations require. These two linear regressions consist of evaluating X_1 (4.14) and Y_1 (4.13).The X_1 and Y_1 regression starts with an initial p value as well as a guess for the C_h value. These values have an impact on the wall temperature calculations; therefore, the viscosity ratio must be adjusted in both linear regression processes. From the X_1 and Y_1 regression, C_c and C_h coefficients are found. This C_h coefficient is then used in a mathematical relaxation method to converge the viscosity ratio in the X_2 and Y_2 linear regression, producing values of p and C_c. The new p is used in the next iteration of regressions (which has new viscosity ratios to be relaxed). Calculations continue following this procedure until the difference between the successive p and C_h values and the C_c values from the X_1-Y_1 and X_2-Y_2 linear regressions reach a predetermined allowable error.

5. MODIFICATION OF C# CODE APPLICATION SOFTWARE

By reviewing available research paper, C #code application software is modified to develop a Nusselt number correlation as given in (2.4) and using single phase analysis, considering viscosity effect for a specific plate geometry and chevron angle. By referring readings of different chevron angle configuration of plate heat exchanger given by authors Amir Jokar [23], Ali Hashmi [7], i.e. $T_{h\ in}$, $T_{h\ out}$, $T_{c\ in}$, $T_{c\ out}$, m_h, m_c at various inlet/outlet temperature and mass flow rate of hot and cold water to evaluate C_1, m of Nusselt number correlation. Overall heat transfer coefficient arrived experimentally as well as developed correlations of authors, compared with correlations arrived by C# code application software. C# code application software calculation comparison work is shown in Fig.3 and Fig.4.

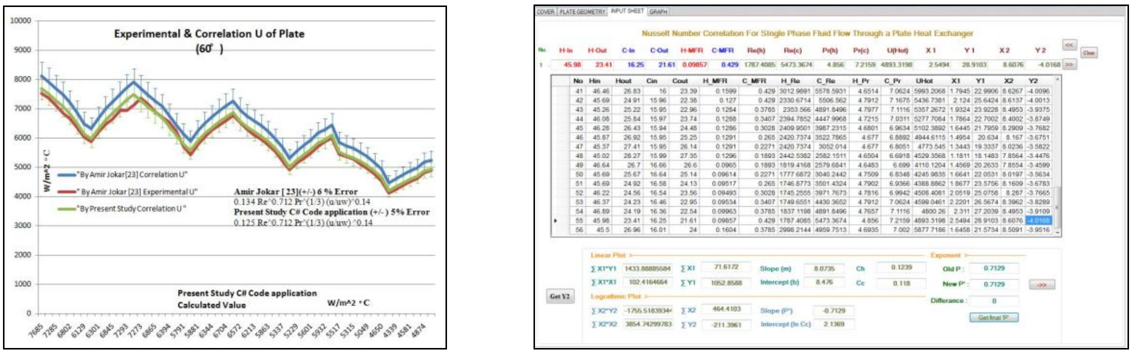


Fig.3 Comparison of Authors Experimental ‘U’ & Correlations ‘U’with C# code application software arrived Correlations‘U’of plate chevron angle β=60/60 □ and screen shot of C# code application software

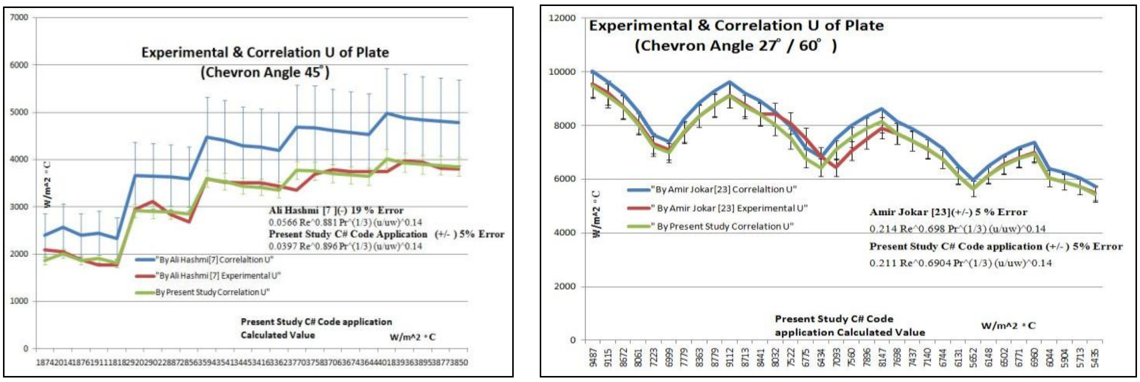


Fig.4 Comparison of Authors Experimental ‘U’ & Correlation ‘U’ with C# code application software arrived Correlations ‘U’ of plate chevron angle β=45 □ and β=27/60 □

6. CONCLUSION

Developed C# code application software gives heat transfer characteristics behavior of a commercial plate heat exchanger with different geometrical parameters under varying flow conditions. It provides convective heat transfer coefficient for the hot and cold fluid resulting into overall heat transfer coefficients. To use this C# code application software one needs to be, experimental readings at various temperature and flow rate of hot/cold water. Earlier research paper readings were used in application software and checked with the authors result. It is found that, calculations done in C# code application with the help of water property data base at various temperatures, calculations matches with (+/-) 5 % in error. Based on the result, we can use C# code application for developing a simplified Nusselt number correlation incorporating effects of Reynolds number, Prandtl number, viscosity variation, for different type of plate geometric configuration and specific chevron angle ranging from 20 □ <β< 65 □ using water as a fluid medium. This will add data base of Nusselt number correlation for different plate geometry and chevron angle. However this study does not include the chevron angle effect on Nusselt number correlation.

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NOMENCLATURE

A_x :effective surface area,(m^2)
A_o : minimum free flow area between two plates(m^2)
b: plate spacing in PHE(m)
C: constant
C_p :specific heat at constant pressure(Kj/kg-K)
D_h :hydraulic diameter(m)
G :mass flux(Kg/m2-s,
h :convective heat transfer coefficient (W/m2-K)
k : thermal conductivity of fluid (W/m-K)
K_{fluid}: thermal conductivity of the fluid (W/m-K)
k_{wall} :thermal conductivity of the plate material separating the fluids (W/m°C)
L :characteristic length in general (m)
LMTD: Log Mean Temperature Difference
m* : mass flow rate, kg/s
NTU :Number of transfer units
N_u: Nusselt number
P_r: Prandtl number
Q*: heat transfer rate, W (Btu/hr)
R_e: Reynolds number
t : thickness of plate(m)
T: temperature (°C)
ΔT : temperature difference(°C)
U :overall heat transfer coefficients (W/m^2 □ C)
V*: volumetric flow rate, m3/s (gpm)
w: width of plate, m
X_1: Wilson plot parameter in x direction
Y_1: Wilson plot parameter in y direction
β: chevron Angle (°)
P: density, kg/m3
p: Reynolds number exponent
□: enlargement factor
μ : viscosity of fluid at avg temperature, kg/m-s
μ_w : viscosity of fluid at wall temperature, kg/m-s

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