Performance Investigation of Commercially Available Corrugated Plate Heat Exchanger (PHE) using Nanomaterial in Base Fluid

R. N. Lipare[1] S. M. Shaikh[2]
[1]M.Tech. Student [2]Assistant Professor
[1,2]Department of Mechanical Engineering
[1,2]Dr. J J Magdum College of Engineering, Jaysingpur Maharashtra, India

Abstract— The plate heat exchanger concept, with flow-through channels formed by corrugated plates and the heat transfer taking place through the thin plates, is an extremely efficient heat exchange technique. The objective of this paper is investigating performance of Plate Heat Exchanger using Al2O3 nanomaterial in water and finding out overall heat transfer coefficient of a commercially available gasketed plate heat exchanger using modified Wilson plot technique. Experimentation was done on a commercially available PHE. 1% Al2O3 Nanofluid and Water was used as a working fluid for both hot and cold sides respectively. Maintaining the hot and cold inlet temperature constant, mass flow rate of water at the inlet of hot and cold sides were changed. Temperature at the outlet of hot and cold sides were measured each time and tabulated. Reynolds numbers was maintained between 500<Re<5000. Finally, a new C# code was developed in order to perform the calculations and to find out Overall heat transfer coefficient correlation. The modified Wilson plot technique in this paper will facilitate in determining Overall heat transfer coefficent correlation for any type of plate heat exchanger with any fluid as medium, without human interference and thereby eliminating human error.

Key words: PHE, Nano-Particle Suspensions

I. INTRODUCTION

In the last two decades, use of plate heat exchanger is increased because of high efficiency and compactness. The plate heat exchanger concept, with flow-through channels formed by corrugated plates and the heat transfer taking place through the thin plates, is an extremely efficient heat exchange technique. The other nano-fluid technology has emerged as a new technique in recent years. Nano-fluid as a next generation fluid that may revolutionize heat transfer by adding tiny particles to a conventional fluid, up to 40% of the fluids capability to transfer heat can be improved. That is, the dispersion solution (i.e., the nano-fluid), which is produced by dispersing nano-particles into fluids, is known to significantly enhance the poor thermal conductivity of the water. The basic phenomenon of nano-fluid is that suspensions that contain solid particles have effective thermal conductivity by their mixing effects. The nano-particles used in nano-fluids commonly have a small average size, below diameter. Therefore, nano-fluid technology has succeeded in the enhancement of thermal conductivity without the aforementioned problems.

II. SELECTION OF NANOMATERIAL AND PREPARATION OF NANOFLUID

Alumina nano-fluids are used in that research because they are widely used in that area of research due to requirements such as stability, homogeneity, and continuous suspension without any outstanding chemical change of the base fluid . The size has a normal distribution in a range from 10 nm to 100 nm (47 nm avg. diameter is given from the manufacturer).

Preparation of nano-particle suspensions in the water as base fluid is the first step of applying nano-fluids in heat transfer enhancement. In the present study, Al2O3 nano-particles were dispersed in distillate water as a base fluid. After measuring the equivalent volume to the required mass of nano-particle powder alumina nano-fluids with certain mass concentrations for the experiments are prepared by controlling the amounts of the particles. Nano-fluids of 1% , volume fraction of particles were prepared. The nano-fluids were treated by ultrasonication without using any dispersant or stabilizer to prevent any possible changes of chemical properties of the nano-fluid due to presence of additions. The prepared nanofluids samples were subjected to ultrasonication for about 4 h. No precipitation / settlement of nano-particles was observed after 24 h of settling the suspension. In order to ensure a stable, uniform, continuous suspension, the dispersion solutions are vibrated in an ultrasonic bath for about 8 h just before performing the tests. The presence of sedimentation leads to non-homogenous nano-particle dispersions and possible fouling.

III. EXPERIMENTAL SET UP

A literature survey has been performed to examine the experimental set-ups of single phase fluid flow through a corrugated plate heat exchanger and Authors like T.S. Khan[3], Iulian Gherasim[4], Minsung Kim[5], F.Akturk[6], Ali Hashmi[7], Giovanni A.Longo [9], Jaekyoo Jang [10], A Kabeel [8] had selected equipments for experimental set up as given below.

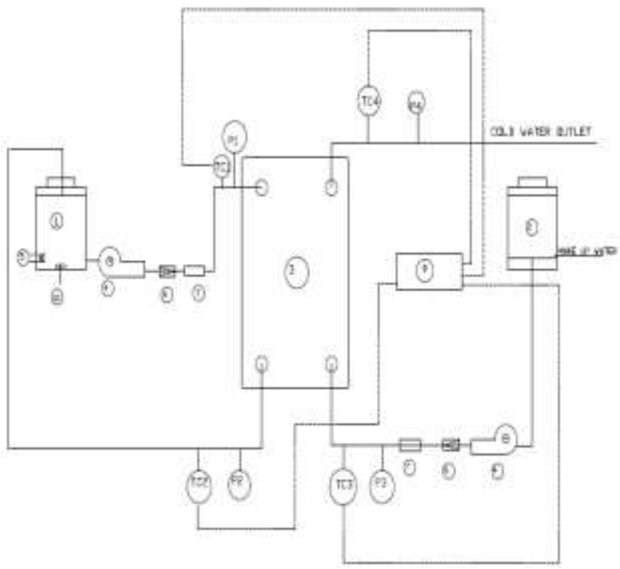


Fig. 1: Experimental Setup

IV. EVALUATING PARAMETER OF PHE WITH 1%Al_2O_3 NANOFLUID

A. *Reynolds Number*

In the case of the plate heat exchangers, the hydraulic diameter is very small, it is almost in the order of mm, therefore the turbulent conditions are achieved very early i.e., at a very low value of Reynolds number.

$$Re = G\ Dh / \mu \qquad (4.1)$$

B. *Overall Heat Transfer Coefficient*

It is most convenient to use overall heat transfer coefficients in heat transfer calculations as these combine all of the constituent factors into one, and are based on the overall temperature drop.

$$1/U = 1/hh + t/kwall + 1/hc \qquad (4.2)$$

C. *Nusselt Number*

The Nusselt number may be physically described in case of plate heat exchanger as

$$Nu = (h \times L) / kfluid = (h \times Dh) / kfluid \qquad (4.3)$$

Nusselt number is equal to the dimensionless temperature gradient at the surface, and it essentially provides a measure of convective heat transfer.

$$Nu = C1\ Re^{p}\ Pr^{C3}\ (\mu/\mu w)^{C4} \qquad (4.4)$$

The generalized polynomial equation for Thermal conductivity of 1% Al_2O_3 nanofluid from[1] is as follows :

$$K = -1.241+0.1365*T-0.003108*T^2 +0.1311 * 10^{-4}*T^3+ 0.3303*10^{-6}*T^4 -0.2542*10^{-8}*T^5-0.1723*10^{-11}*T^6 \qquad (4.5)$$

The generalized polynomial equation for viscosity of 1% Al_2O_3 nanofluid from[1] is as follows:

$$\mu=1.67-0.04373*T+0.1835*10^{-4} *T^2 + 0.2076 *10^{-4}*T^3-0.42*10^{-6}*T^4+0.34*10^{-8} * T^5- 1.007* 10^{-10}* T^6 \qquad (4.6)$$

The generalized polynomial equation for Density of 1% Al_2O_3 nanofluid from[1] is as follows

$$\rho =1.051 - 0.004699*T + 0.2573*10^{-3}*T^2 -0.7934*10^{-5}*T^3+0.1322*10^{-6} *T^4- 0.1106*10^{-8} *T^5+ 0.3638*10^{-11}*T^6 \qquad (4.7)$$

The generalized polynomial equation for Volumetric Specific Heat of 1% Al_2O_3 nanofluid from [1] is as follows

$$C_{P}=-1.092+0.3467T-0.01194*T^2+ 0.1831*10^{(-3)}*T^3-0.1595*10^{(-5)}*T^4 +0.1012*10^{(-7)}*T^5 - 0.3858*10^{(-10)}*T^6 \qquad (4.8)$$

V. ANALYSIS OF CORRUGATED PHE USING MODIFIED WILSON PLOT TECHNIQUE

The experimental result analysis used by Ali Hashmi [7], F. Akturk [6], Minsung Kim [5], Iulian Gherasim [4], Jaekyoo Jang [10], T.S. Khan [3], and Giovanni A. 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. It is based on the separation of the overall thermal resistance into the inside convective thermal resistance and the remaining thermal resistances participating in the heat transfer process. As the first step in analyzing the collected data, properties of 1% Al_2O_3 nanofluid for each test point were calculated at bulk temperatures averaged between the inlet and outlet ports on each side.

The properties specific heat, conductivity, and viscosity were used to find the flow characteristics in the channels for each plate. The Reynolds number of the flow within the channels was calculated by, 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 [12].

$$G = \rho\ V^* / A0 \qquad (5.1)$$

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.

$$A_0 = b \times w \qquad (5.2)$$

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 (5.2). A more thorough description of the minimum free flow area is given in a previously published paper by the authors, Hayes and Jokar [11], while (5.3) 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.

$$Q^* = m^*\ Cp\ \Delta T \qquad (5.3)$$

Using the log- mean temperature differene,

$$\Delta T_{lm} = [(T_h in - T_c out) - (T_h out - T_c in)] / \ln [(T_h in - T_c out) - (T_h out - T_c in)] \qquad (5.4)$$

The overall heat transfer coefficient in the heat exchanger was calculated by

$$U= Q^*/ (A_x\ \Delta T_{lm}) \qquad (5.5)$$

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 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, Amir Jokar [11].

$$Nu = C\ Re^{p}\ Pr^{1/3}(\mu/\mu_w)^{0.14} \qquad (4.5)$$

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 [13], 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

$$(1/U)-(t/k)_{wall} = \{(1/C_c(k_c/D_h)Re^p_c Pr_c^{1/3}(\mu/\mu_w)_c^{0.14})\} + \{(1/C_h(k_h/D_h)Re^p_h Pr_h^{1/3}(\mu/\mu_w)_h^{0.14})\} \quad (5.8)$$

Multiplying both sides of (5.8) by

$$(k_c/D_h)Re^p_c Pr_c^{1/3}(\mu/\mu_w)_c^{0.14}) \quad (5.9)$$

It gives to form of ,

$Y_1 = m X_1 + b$

Where

$$Y_1 = \{(1/U) - (t/k)_{wall}\} \times \{(k_c/D_h)Re^p_c Pr_c^{1/3}(\mu/\mu_w)_c^{0.14})\} \quad (5.10)$$

$$X_1 = \{[(k_c/D_h)Re^p_c Pr_c^{1/3}(\mu/\mu_w)_c^{0.14})] / [(k_h/D_h)Re^p_h Pr_h^{1/3}(\mu/\mu_w)_h^{0.14}]\} \quad (5.11)$$

$$\text{Slope: } m = (1/Ch) \quad (5.12)$$

$$\text{Intercept: } b = (1/Cc) \quad (5.13)$$

A. Logarithmic Modification

Given below is the logarithmic modification of (5.8)

It will become

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

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

Taking log on both sides

$\ln y2 = -\ln C_c - p \ln Re_c$

$Y2 = \ln y2$

$X2 = \ln Re_c$

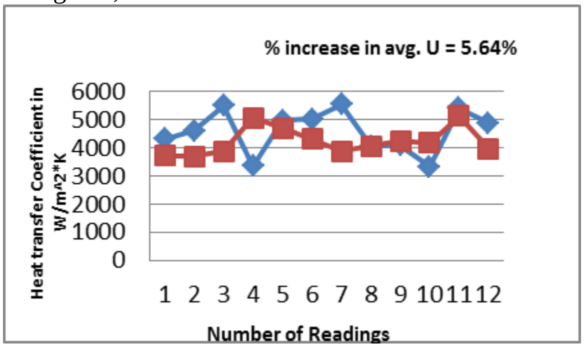
Slope: (p), Intercept: ($-\ln C_c$)

B. Iterative Procedure

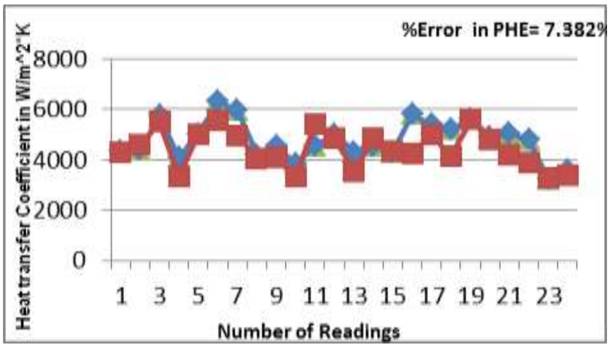
Since the viscosity ratio groups and the Reynolds number exponents undergo a mathematical relaxation method with the fluid flow rates and temperatures, successive linear regressions can be performed to execute the nonlinear regression that these equations require. These two linear regressions consist of evaluating X1 (5.11) -and Y1 (5.10).The X1 and Y1 regression starts with an initial p value as well as a guess for the Ch 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 X1 and Y1 regression, Cc and Ch coefficients are found. This Ch coefficient is then used in a mathematical relaxation method to converge the viscosity ratio in the X2 and Y2 linear regression, producing values of p and Cc. 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 Ch values and the Cc values from the X1-Y1 and X2-Y2 linear regressions reach a predetermined allowable error.

VI. MODIFICATION OF C# CODE APPLICATION SOFTWARE

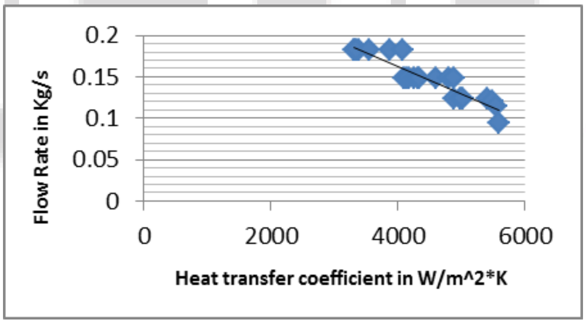
By reviewing available research paper, C #code application software is modified to develop a Nusselt number correlation and in turn U correlation as given in (2.4) and using single phase analysis, considering viscosity effect for a specific plate geometry and chevron angle. By referring readings of different chevron angle configuration of plate heat exchanger given by authors Amir Jokar [11], Ali Hashmi [7], i.e. Th in , Th out, Tc in, Tc out, mh, mc at various inlet/outlet temperature and mass flow rate of hot and cold water to evaluate C1, m of Nusselt number correlation. Overall heat transfer coefficient arrived experimentally as well as developed correlations arrived by C# code application software are compared. C# code application software calculation comparison work is shown in Fig. 6.1, 6.2 and 6.3.



6.1 Comparison U for 1% Alumina nanofluid and U for water



6.2 Comparison of U correlation and U experimental for 1% Alumina nanofluid



6.3 Flow Rate VS Heat Transfer Coefficient for 1% Alumina nanofluid

VII. CONCLUSION

Developed C# code application software using Modified Wilson Plot technique gives heat transfer characteristics behavior of a commercial plate heat exchanger for 1% Al_2O_3 nanofluid. 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. It is found that, calculations done in C# code application with the help of 1% Al_2O_3 nanofluid properties from Harkirat and Dr. D Gangacharyulu [1] , calculations matches with (+/-) 7.5 % 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 $20^{o} < \beta < 65^{o}$ using 1% Al_2O_3 Nanofluid with water as base fluid. 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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