Determination of heat transfer correlations using a plate heat exchanger

Part 1: Phat Le
Part 2: Rafael Vasquez
Part 3: Jed Durante
Part 4: Rodrick Alberto

Section B00 (M/W), Team B13, Lab 1

Abstract

Plate heat exchangers are important pieces of equipment in chemical engineering and analysis of these devices are crucial for efficient heat transfer. In this report, a plate heat exchanger with five channels were run with 60 °C water and room temperature water and then analyzed. From the analysis, the Nusselt number, Reynolds number, and Prandtl number correlations were determined. The Nusselt number correlations for C_1 and C_2 values were determined to be 0.8 ± 0.2 and 0.60 ± 0.03, respectively. In addition, the Nusselt vs. Reynolds number appeared to have an upward positive trend, but the Nusselt vs. Prandtl number appeared to have a negative downward trend. This is due to the Nusselt number primarily depending on viscosity, and the Reynolds and Prandtl number being inversely related through viscosity. These results are similar with literature data and what would be expected. Thus, the lab provides an accurate representation of literature, allowing for theoretical data to be verified. In addition, although this lab was only specific to our heat exchanger, similar standard operating procedures could be extended to different setups.

1 Introduction

The need to heat or cool fluids is an important aspect in chemical engineering. Thus, heat exchangers are widely used in the chemical engineering industry for their efficient ability to transfer heat. Heat exchangers come in various designs such as shell-and-tube, scraped surface, stacked plates, and many more. Shell-and-tube heat exchangers have multiple tubes inside a large shell, where one fluid flows through the tubes and another fluid flows through the shell transferring heat as they pass over another. Shell-and-tube heat exchangers are used in various fields such as chemical processing, air conditioning, and pharmaceuticals.[1] Scraped surface heat exchangers have a cylinder in which a fluid flows through it and another fluid flows counter-current in a tube transferring heat as they pass. Inside the cylinder are scraping blades that keep the fluid from settling on the interior surfaces. Scraped surface heat exchangers are used for highly viscous fluids such as in food processing.[2] Stacked plate heat exchangers (PHE) use corrugated plates stacked in parallel where each side of the corrugated plate are alternate channels of hot and cold fluid allowing for heat transfer. Due to their ability to efficiently transfer heat with respect to their volume, they have been used to phase out shell-and-tube heat exchangers used in petro-chemical and air conditioning.[3] The analysis of these heat exchangers has been widely studied and is crucial for the efficiency of these devices.

2 Background

A PHE has multiple design aspects that can be modified and configured in various ways to meet a specification or goal. Key designs that can have substantial effects on the efficacy and performance of a PHE include having a flat or corrugated PHE, the corrugation angle, the corrugation depth, the material used for the PHE, the fluid used, the flow rate of the fluid, and having the fluids parallel or counter flow in the PHE.

Some examples of how changing key designs effect the efficiency of the PHE include modifying a flat PHE to a corrugated PHE to increase the area of heat transfer or using a different material for the PHE with a higher thermal conductivity to improve heat transfer.[4]

Different types of corrugated PHE exist with different profiles like that of an asterisk where the plate has an impression of an asterisk or those where the impression of the corrugation is at angle to the flow of the fluid.[5] Generally, corrugated PHE are most effective at heat transfer compared to flat PHE because the increased surface area; however, the corrugation angle can also change the quantity of heat energy transferred.[4] Increasing the corrugation angle does increase the heat transfer surface area but that isn't what improves heat transfer; the increased corrugation angle causes the flow of a fluid through the PHE to become more turbulent which directly improves the heat transfer.[4]

As a result, the fluid and the flow rate play a principal role in how effectively heat is transferred in a PHE. The choice of a fluid is dependent on their physical properties such as heat capacity and viscosity.[6] Commonly used fluids in PHE include water, synthetic fluids such as ethylene glycol or different nanofluids such as multi-walled carbon nanotubes discussed in Sarafraz et al.[6,7] In terms of the flow rate of the fluid, a higher flow rate, and in turn a higher Reynolds number, increases the Nusselt number, indicating that more convective heat transfer occurs instead of conductive heat transfer.[4]

Other factors that can affect the heat transfer is fouling, where deposits accumulate onto the surface that the fluid is in contact with.[6] Foulage, the deposits that accumulates, increases the flow resistance as the operation of the PHE continues.[6] The foulage also decreases the thermal conductivity.[6] As a result, fouling is considered when calculating the Nusselt numbers and Reynolds number.

Here in this experiment, we show how changing the flow rates of the water for the inlet and outlet of the cold and hot streams in a corrugated PHE affect the efficiency of the PHE. Different flow rates for the inlet and outlet cold and hot streams were tested and the temperature of the inlet and outlet of the cold and hot streams were recorded and analyzed using the correlations from Yang et al.[7]

3 Theory

The flow rates of the cold and hot streams were analyzed using the Nusselt number, Prandtl number and the Reynolds number. The Nusselt number,

$$Nu = C_1 Re^{C_2} Pr^{1/3} \tag{1}$$

where Re is the Reynolds number, Pr is the Prandtl number, and C_1 and C_2 are dimensional experimentally determined coefficients, is a modified version of a correlation of the Nusselt number found in Yang et al. and Pradhan et al.[7,8]

The Reynolds number is defined as

$$Re = \frac{\rho V D_h}{\mu} \tag{2}$$

where ρ is the density of the water in $\mathrm{kg\,m^{-3}}$, V is the velocity of water in $\mathrm{m\,s^{-1}}$, D_h is the hydraulic diameter of the PHE in m, and μ is the viscosity of water in $\mathrm{Pa \cdot s}$. The Reynolds number is the ratio of inertial forces to viscous forces which determines the kind of flow the fluid has: laminar or turbulent. The hydraulic diameter is defined as

$$D_h = 2b \tag{3}$$

where b is the corrugation depth m.

The Prandtl number is defined as

$$Pr = \frac{\mu C_P}{k} \tag{4}$$

where C_P is the heat capacity of water at a constant pressure in $\mathrm{J\,kg^{-1}\,K^{-1}}$, and k is the thermal conductivity of the material of the PHE in$\mathrm{W\,m^{-1}\,K^{-1}}$. The Prandtl number is a ratio of momentum diffusivity to thermal diffusivity.

The energy balance deviation (EBD) equation,

$$EBD = \frac{|Q_H - Q_C|}{Q_{\text{avg}}} \times 100\% \tag{5}$$

where Q_{H} and Q_{C} are the heat transfer for the hot fluid and the cold fluid in W, and Q_{avg} is the average value of Q_{H} and Q_{C} in W, is derived from Yang et al.[7] The purpose of the energy balance deviation equation is to determine if the data collected is consistent with heat transfer theory, such as with the heat transfer rate equation. Q_{i}, where i is an index for hot and cold, is defined as,

$$Q_i = m_i \rho C_{P,i}(T_{i,\text{in}} - T_{i,\text{out}}) \tag{6}$$

where m_{i} is the mass flow rate, in kg s^{-1}, of the hot water, $C_{\text{P,i}}$ is the heat capacity at a constant pressure of the hot water in kg s^{-1}, $T_{\text{i,in}}$ is the temperature of the inlet i water and $T_{\text{i,out}}$ is temperature of the outlet i water, both in °C.

The overall calculated heat transfer coefficient,

$$U_{\text{calc}} = \frac{1}{\frac{1}{h_H} + \frac{t}{k_w} + \frac{1}{h_C}} \tag{7}$$

where h_{H} is the heat transfer coefficient for the hot side of the PHE and h_{C} is the heat transfer coefficient for the cold side of the PHE, both in $\text{W m}^{-2}\,\text{K}^{-1}$, t is the plate thickness, in m, and k_{w} is the thermal conductivity of the PHE, in $\text{W m}^{-1}\,\text{K}^{-1}$, is compared to the overall measured heat transfer coefficient, U_{meas}, in $\text{W m}^{-2}\,\text{K}^{-1}$,

$$U_{\text{meas}} = \frac{Q}{A_{proj}\Delta T_{LMTD}} \tag{8}$$

where Q is the heat transfer rate, A_{proj} is the projected heat transfer area in m^2, and ΔT_{LMTD} is the log-mean temperature difference in °C, to determine if the data collected in the experiment is consistent with heat transfer theory, shown through the overall calculated overall heat transfer coefficient. U_{calc} uses the physical properties of water and the PHE to determine the overall calculated heat transfer coefficient whereas U_{meas} takes the data collected in the experiment to determine the overall measured heat transfer coefficient. The equations for U_{calc} and U_{meas} are from Yang et al.[7]

4 Methods

The experiment was run in a steady state process where room temperature water is pumped from a cold tank and passed through six stacked PHE with five channels at atmospheric pressure in process with 60 °C water pumped from a separate hot reservoir. The now heated water is sent to a secondary tank and not pulled from until the original tank water was all used up. Flow rates varied between 3 - 8 gallons per minute for both the hot and cold water. Thermocouples were attached at the inlet and outlet entrances and exits where it was recorded, with each trial run until steady state, temperatures appearing constant with time.

5 Results and Discussion

The trials conducted for this experiment yielded three major results: the relationship between U_{meas} and U_{calc}, the coefficients C_1 and C_2 for Eq. (1), and the relationships between the various dimensionless numbers with each other. Before any of these are addressed, however, an analysis of the EBD was performed to ensure that the heat lost from the hot fluid was equivalent to the heat gained by the cold fluid. Fig. 1 demonstrates this relationship. As seen here, all the measurements display an EBD of less than 12%. This reveals that for the most part, the energy from the hot fluid was equivalent to the energy gained by the cold fluid. Such information is relevant since it matches up with the heat transfer concept of heat going in a system equating to the heat going out a system. In comparison to the EBD calculation done by Yang et al., theirs was not as plausible since they obtained negative results.[7] Their data implies that the temperatures going out the hot stream and in the cold stream are greater than the temperatures going in the hot stream and out the cold stream, respectively. This is impossible considering

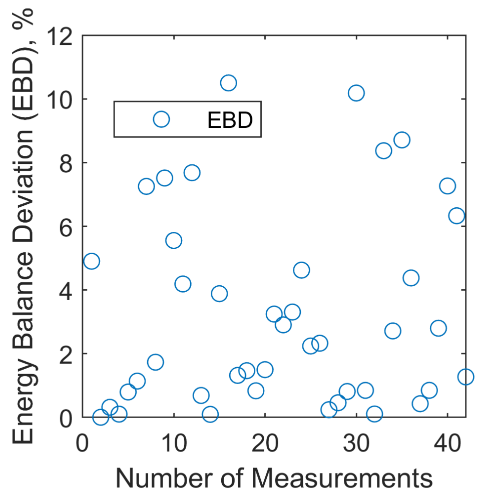

Figure 1: The figure above displays the EBD of each experimental trial, shown as blue dots. As can be seen here, the EBD of each trial never reached greater than 12%. This demonstrates that the energy lost by the hot stream was equal to the energy gained by the cold stream.

that the hot water has to be cooled down and the cold water has to be warmed up since it's a counterflow heat exchanger. Consequently, the EBD calculations done with the experimental data is much more sensible. After ensuring that the EBD lined up with heat transfer theory, U_{meas} and U_{calc} were obtained using Eq. (8) and Eq. (7), respectively.

Based off the data acquired from the experimental trials, the values for U_{meas} and U_{calc} were shown to be very close to one another. Fig. 2 displays the close relationship between the two values. The line in red does not serve as a calibration curve or line of best fit. Rather, it serves as the $y = x$ line to demonstrate that for the most part, U_{meas} and U_{calc} are equivalent. Furthermore, the relationship between U_{meas} and U_{calc} matches with what was reported by Yang et al. They display a similar figure in their paper showing the relationship between the measured and calculated overall heat transfer coefficients.[7] Thus, the results obtained from this experiment align with those that have been reported in previous literature. Using the

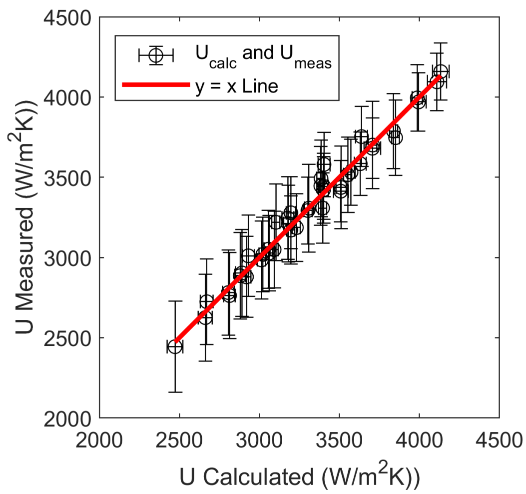

Figure 2: The figure above displays U_{meas} vs. U_{calc}. The red line is the $y = x$ line, which reinforces the idea that U_{meas} and U_{calc} are generally similar to each other. This trend is previously highlighted in other works, such as in Yang et al.[7]

values of U_{calc}, C_1 and C_2 were able to be calculated next.

The next major result obtained from the experimental trials was the coefficients C_1 and C_2 for Eq. (1). The coefficients were found to be $C_1 = 0.8 \pm 0.2$ and $C_2 = 0.60 \pm 0.03$. In comparison to coefficients found in others' results, these coefficients are a small departure from what has been presented in previous literature. The coefficient values from other works were used as an initial guess in order to solve for the coefficient values that pertain to this specific experiment. Pradhan et al. provides various coefficients for the correlation similar to the one presented by Yang et al.[8] This deviation in coefficient values is attributed to the fact that these coefficients are for this specific scenario and experimental setup only. For example, Yang et al.'s setup included a chiller to keep the water at a much more colder temperature.[7] For this experiment, only room temperature water was kept in a tank without any external chilling equipment, so the temperatures for the cold inlets and outlets were not as cold compared to Yang et al.'s. Furthermore, the various coefficients provided in Pradhan et al. changed based on the chevron angle, which is the angle in which the grooves of the plates are arranged.[8] As a result, the values for C_1 and C_2 were slightly different than what is normally reported since they only apply to this specific scenario. Once C_1 and C_2 were found, they were used to calculate

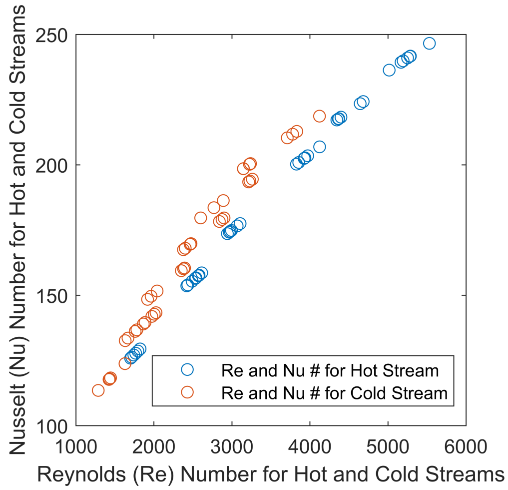

Figure 3: The figure above displays Nusselt vs. Reynolds for both the hot and cold streams. In both cases, there's a positive slope. The purpose of this plot is to show the relationship between Nusselt and Reynolds. Note: the error bars have been omitted for sake of clarity since they are too large.

the Nusselt number (Nu), one of several dimensionless numbers that will be explored shortly.

Finding C_1 and C_2 allowed for the calculation of the Nusselt number, which was then compared to two other dimensionless numbers: the Reynolds number (Re) and the Prandtl number (Pr). Fig. 3 shows the relationship between Nusselt and Reynolds for both the hot and cold streams. As the hot flow rate is kept constant and cold flow rate increases, the Reynolds number decreases and thus the Nusselt number decreases. The opposite could be said for the cold side. As cold flow rate is kept constant and hot flow rate increases, the Reynolds number increases and thus the Nusselt number increases. Thus, there's a positive slope for both instances. They exhibit this relationship since density doesn't increase at a faster rate than the rate in which viscosity decreases. At high viscosities and lower Reynolds numbers, more conductive heat transfer occurs which is why the Nusselt number is smaller. The increase in viscous forces decreases the Reynolds number, which in turn decreases the Nusselt number. Furthermore, the different sets of data depends on the velocity of the stream V as seen in the Reynolds equation, Eq. (2). Thus, as the velocity increases, it can be seen that Reynolds increases as well; consequently, Nusselt increases also.

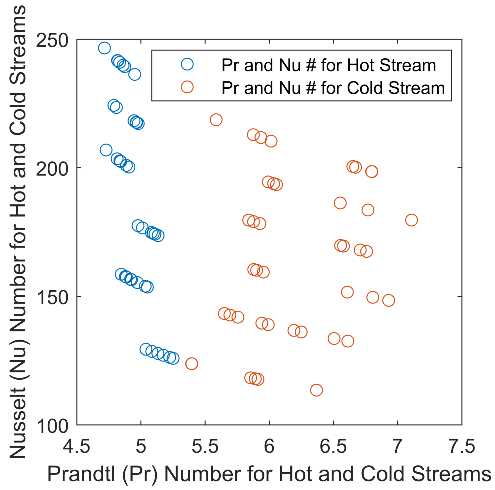

Figure 4: The figure above displays Nusselt vs. Prandtl for both the hot and cold streams. In this instance, there's a negative trend for both cases presented in the figure. Similar to Fig. 3, the purpose of the plot is to compare the two dimensionless numbers featured, which are Nusselt and Prandtl in this case. Just like in Fig. 3, note that the error bars have been omitted for sake of clarity since they are too large.

Fig. 4 appears different than Fig. 3. As the hot stream flow rate is kept constant and cold stream flow rate increases, the Prandtl number increases and thus the Nusselt number decreases. The opposite could be said for the cold side. As cold flow rate is kept constant and hot flow rate increases, the Prandtl number decreases and thus the Nusselt number increases. Additionally, since Nusselt is the ratio between convective to conductive heat transfer, increasing the velocity of the stream V increases the convective heat transfer. As the velocity increases, the Nusselt number increases as well. However, the Prandtl number increases since more cooling is done and therefore the viscosity of the water at low temperatures would be higher. Furthermore, Fig. 4 has a negative slope is due to Prandtl increasing at a slower rate than Reynolds since Prandtl is raised to the $1/3$ while Reynolds is raised to the 0.60. Therefore, Nusselt primarily depends more on Reynolds rather than Nusselt. As demonstrated by these two figures, the dimensionless numbers exhibit certain relationships with each other.

6 Conclusions

This experiment conducted using a PHE with five channels shows strong similarities between theoretical and experimental values and relationships. For instance, it was found that when plotting U_{meas} vs. U_{calc} there appears to be a positive upward trend, this indicates that the theoretical and experimental values are similar and match up. Although the C values, $C_1 = 0.8 \pm 0.2$ and $C_2 = 0.60 \pm 0.03$, were slightly different from literature values, the difference is considered negligible. In addition, when plotting the Nusselt vs. Reynolds number there appears to be a similar positive upward trend which is expected since the Nusselt number is a function of the Reynolds number. However, when the Nusselt vs. Prandtl number were plotted there appears to be a negative downward trend despite Nusselt also being a function of the Prandtl number. This is due to the fact that viscosity is playing an important factor in determining the Nusselt number. The Reynolds and Prandtl number are inversely related in terms of viscosity, and since Reynolds is raised to 0.60 it has larger changes in magnitude than Prandtl because Prandtl is only raised to 0.33. The disparities between theoretical and experimental are attributed to not having the exact equipment as literature. Thus, to replicate literature exactly there would need to be identical equipment being used. The work done in this lab is specific for the setup presented, but similar work for other types of setups could be achieved and verified following similar standard operating procedures.

Bibliography

(1) Bougriou, C.; Baadache, K. Shell-and-double concentric-tube heat exchangers. *Heat and Mass Transfer* **2010**, *46*, 315–322.

(2) Fitt, A.; Lee, M.; Please, C. Analysis of heat flow and "channelling" in a scraped-surface heat exchanger. *Journal of Engineering Mathematics* **2007**, *57*, 407–422.

(3) Bigoin, G.; Tochon, P.; Grillot, J. Numerical investigation of plate heat exchanger surfaces. *6th International Conference on Advanced Computational Methods in Heat Transfer* **2000**, *3*, 507–516.

(4) Kahn, T. et al. Experimental Investigation of Single Phase Convective Heat Transfer Coefficient in a Corrugated Plate Heat Exchanger for Multiple Plate Configurations. *Applied Thermal Engineering* **2009**, *30*, 1058–1065.

(5) Durmuş, A. et al. Investigation of heat transfer and pressure drop in plate heat exchangers having different surface profiles. *Int. J. Heat Mass Transfer* **2007**, *52*, 1451–1457.

(6) Sarafraz, M.; Hormozi, F. Heat transfer, pressure drop and fouling studies of multi-walled carbon nanotube nano-fluids inside a plate heat exchanger. *Exp. Therm. Fluid Sci.* **2015**, *72*, 1–11.

(7) Yang, J.; Jacobi, A.; Liu, W. Heat transfer correlations for single-phase flow in plate heat exchangers based on experimental data. *Applied Thermal Engineering* **2016**, *113*, 1547–1557.

(8) Pradhan, R.; Ravikumar, D.; Pradhan, D. Review of Nusselt Number Correlation for Single Phase Fluid Flow through a Plate Heat Exchanger to Develop C Code Application Software. *IOSR Journal of Mechanical and Civil Engineering* **2013**, *4*, 1–8.

Appendix

1. Shell-and-double concentric-tube heat exchangers
   - Author(s): Bougriou, C.; Baadache, K.
   - Year published: 2010
   - Journal name: Heat and Mass Transfer
   - 1-3 major accomplishments of this paper:
     (a) Analysis of shell-and-double concentric-tube heat exchangers was done to determine performance of the device.
     (b) Applications of shell-and-double concentric-tube heat exchangers in industries such as chemical processing, air conditioning, and pharmaceuticals were discussed.
2. Analysis of heat flow and "channelling" in a scraped-surface heat exchanger
   - Author(s): Fitt, A.D.; Lee, M.E.M.; Please, C.P.
   - Year published: 2007
   - Journal name: Journal of Engineering Mathematics
   - 1-3 major accomplishments of this paper:
     (a) Analysis of heat flow and fluid dynamics for scraped-surface heat exchanger was done to determine how different variables affect the heat exchanger's efficiency.
     (b) Applications of scraped-surface heat exchangers in industries such as the food industry was discussed.
3. Numerical investigation of plate heat exchanger surfaces
   - Author(s): Bigoin, G.; Tochon, P.; Grillot, J.M.

- Year published: 2000
- Journal name: 6th International Conference on Advanced Computational Methods in Heat Transfer
- 1-3 major accomplishments of this paper:
  (a) Plate heat exchangers were examined to determine how different variables, such as geometry and number of plates, affect the plate heat exchanger's operations.
  (b) Applications of plate heat exchangers in industries was discussed, such as in the air conditioning, petro-chemical, and food industries.

4. Experimental investigation of single phase convective heat transfer coefficient in a corrugated plate heat exchanger for multiple plate configurations
   - Author(s): Khan, T.S.; Khan, M.S.; Ming-C, Chyu; Ayub,Z.H
   - Year published: 2009
   - Journal name: Applied Thermal Engineering
   - 1-3 major accomplishments of this paper:
     (a) In the paper, correlations of the Nusselt number were derived from experimental data with small error.
     (b) An increase in the corrugation angle lead to an increase to the heat transfer coefficient.

5. Heat transfer, pressure drop and fouling studies of multi-walled carbon nanotube nanofluids inside a plate heat exchanger
   - Author(s): Sarafraz, M.M.; Hormozi, F.
   - Year published: 2015
   - Journal name: Experimental Thermal and Fluid Science

- 1-3 major accomplishments of this paper:
  (a) Water containing multi-walled carbon nanotubes improved the thermal conductivity coefficient to 68% relative to water.
  (b) The convective heat transfer coefficient increased with the use of a nano-fluid.

6. Investigation of heat transfer and pressure drop in plate heat exchangers having different surface profiles

   - Author(s): Durmuş, Aydin; Benli, Hüseyin; Kurtbaş, Ífran; Gül, Hasan
   - Year published: 2007
   - Journal name: International Journal of Heat and Mass Transfer
   - 1-3 major accomplishments of this paper:
     (a) A corrugated PHE increases heat transfer compared to flat PHE.
     (b) The Nusselt number increases with a corrugated PHE compared to the asterisk and flat type PHE.