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HEAT TRANSFER IN NON-CORRUGATED PLATE HEAT EXCHANGER USING SERVOTHERM OIL

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ABSTRACT

The heat transfer coefficient for SERVOTHERM^{TM} Medium fluid was determined in three plate heat exchangers of different lengths. The effect of temperature and the length to width (L/W) ratio was investigated. It was found that conductive heat transfer dominates heat transfer. Correlations for Nusselt number based on the viscosity correction for individual heat exchangers and for all the heat exchangers based on L/W have been proposed. The model predictions show a good match with the experimental values.

KEYWORDS: Plate Heat Exchanger, Non-Corrugated, Heat Transfer Coefficient, Viscous Fluids

Servotherm oils are used as heat transfer medium in industrial applications. They possess high heat capacity, high viscosity index, excellent thermal stability and low vapour pressures. These properties make them potential candidates for heating services between 150 and 500 ^{0}C. These oils have a high resistance to thermal cracking and hence maintain their heat transfer efficiency under repeated heating and cooling cycles. Plate heat exchangers (PHE) are being extensively used in industries and in some applications have stated replacing the conventional shell and tube heat exchangers. This is evident from the review article by Abu-Khader (2012) wherein the potential of compact heat exchangers for different applications has been elaborated. Extensive research has been done to study the effect of geometry, corrugation pattern and the gap between subsequent plates on the performance of plate heat exchanger. CFD studies have been done to explore the flow profiles within the exchanger and their effect on the heat transfer coefficient (Freund and Kabelac; 2010, Rios-Eribe et al.; 2016). Majority of the work reported in the literature is based on corrugated plates with the prime objective to induce turbulence and enhance heat transfer. Studies by Dovic et al. (2009) using the conventional Chevron plates reveal that the corrugation pattern contributes a lot in the heat transfer. Modeling and simulation studies based on this have been done by several researchers (Gut et al.; 2004, Selvam et al.; 2007). These studies were done under laminar and turbulent flow conditions. Doo et al. (2010) conducted experiments with riblet-mounted dimple geometry. The studies suggest that the use of such modified surfaces not only enhances heat transfer but also makes the heat exchanger more compact. Generalized correlations highlighting the influence of Prandtl number on heat transfer have been proposed by Arsenyeva et al. (2014). Very few researchers have used non-corrugated plates for heat transfer studies. Moreover, the work done using viscous fluids using these is sparse. The performance of plate heat exchangers using water as a working fluid was evaluated by Kumbhare and Dawande (2013) in laminar and turbulent flow regimes. Modeling has been done by Pinson et al.; 2007 by assuming plate heat exchanger configuration as a porous media. This approach has been compared with several other models. Studies using viscous absorbent salt solution have been done by Warnakulasuriya and Worek (2008) in the Alfa-Laval plate heat exchanger. Correlations were proposed to predict the heat transfer and pressure drop. Ho et al. (2017) investigated the thermal performance of power-law fluids. Significant improvement in the heat transfer efficiency has been reported. Similar studies have been done by Quintero and Vera (2017) using viscous fluids in a fully developed laminar regime with large Peclet numbers. Analytical expressions were proposed for the interfacial and bulk temperatures, overall heat transfer coefficient and Nusselt number. The present study focuses on the heat transfer studies using Servotherm^{TM} as the working fluid. Three plate heat exchangers with different length and approximately the same heat transfer area were used. This was done by varying the number of plates. The data was generated at three different temperatures under laminar flow conditions.

EXPERIMENTAL AND MODELING

Servotherm^{TM} was obtained from Indian Oil Corporation Limited, Vadodara. This was chosen as a working fluid based on its availability, application as a heat transfer fluid and reasonably high viscosity (~ 30 cp). The details of the experimental set-up are reported elsewhere (Joshi et al.; 2016). Three plate heat exchangers having 0.128 m, 0.21 m and 0.30 m as length respectively were used for the experimentation. The width of the plates for all these exchangers was 0.044 m. Gasket of suitable thickness was used to maintain a gap of 0.001 m between the subsequent plates. Based on these dimensions the heat transfer

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areas calculated for the three heat exchangers were 0.0337 m^2, 0.0369 m^2 and 0.0264 m^2 respectively. On the either ends of the plate heat exchanger, pressure plates were used to make the system leak proof. The experimental data was generated at 50 °C, 60 °C and 80 °C. Steady state conditions with respect to temperature and flow rate were ascertained before taking the readings. The flow rates were adjusted using a dimmerstat. In order to minimize the contribution of the outside film heat transfer coefficient (h_o), the flow rate of cooling water was deliberately kept high. The heat transfer coefficient was in the range of 8000 to 10000 kCal/hr-m^2-°C. The rate of heat transfer (Q) and the overall heat transfer coefficient (U_i) were calculated using Eq. 1 and Eq. 2.

$$Q = MC_p \Delta T \tag{1}$$

$$U_i = \frac{Q}{A \Delta T_{lm}} \tag{2}$$

The inside heat transfer coefficient (h_i) was calculated from U_i and h_o. Two approaches were used to propose correlations for Nusselt number. The first one based on the viscosity correction factor was used for individual heat exchangers (Eq. 3). The second approach included the ratio of length to width (L/W) of the plate for the heat exchangers under study (Eq. 4). The latter was consolidated for all the heat exchangers.

$$Nu = aRe^b Pr^c \left(\frac{\mu}{\mu_w}\right)^d \tag{3}$$

$$Nu = aRe^b Pr^c \left(\frac{\mu}{\mu_w}\right)^d \left(\frac{L}{W}\right)^e \tag{4}$$

Regression analysis using the SOLVER feature of MS EXCEL was used to determine the model parameters by minimizing the objective function (Eq. 5).

$$OF = \sqrt{\sum \frac{\left(h_i^{pred} - h_i^{\exp}\right) / h_i^{pred}}{n}} \times 100 \tag{5}$$

RESULTS AND DISCUSSION

The effect of temperature and the length of plate heat exchanger on the heat transfer coefficient were studied using three different exchangers. Since the present work was focused on viscous fluid, the change in the viscosity due to temperature cannot be ignored. It plays a significant role in heat transfer and is also responsible for altering the flow patterns. All the experiments conducted were well within the laminar regime. The effect of temperature on the heat transfer coefficient for the three exchangers under mention is shown in Figures 1 to 3. It is evident from the figures that the heat transfer coefficient decreases with temperature. This trend is different from that observed conventionally where temperature favours heat transfer. However, the latter holds in the case when convective heat transfer is predominant. In the present case owing to the high viscosity and fully developed laminar regime, conduction dominates over convection. This becomes more relevant given the fact that the gap between the subsequent plates is 1 mm. Since conduction controls the heat transfer, thermal conductivity of liquid comes into play. It is known that the thermal conductivity of liquids is directly proportional to the density. The increase in temperature lowers the density thus decreasing the thermal conductivity which ultimately results in decreasing the heat transfer. Figures 1 to 3 show that the profiles for heat transfer coefficient vs. Reynolds' number at 50 °C and 80 °C in are wide apart. The parameters of Eq. 3 for all the three plate heat exchangers obtained by minimizing the objective function (Eq. 5) are reported in Table 1. The power of Reynolds number is in the range of that observed in the case of heat transfer in laminar regime. The negative values of the power for viscosity correction are the result of the inverse relationship of thermal conductivity with temperature. Moreover it is also observed that at a given temperature as the length of the exchanger increases the heat transfer coefficient also increases. This could be because of the increased contribution of convective heat transfer owing to the scope for the formation of eddies with increase in the length. The expression for Nusselt number taking into account the contribution of length to width ratio (L/W) is given in Eq. 6. The constants of this equation were obtained by regression analysis of all the data points for the three plate heat exchangers.

Table 1: Parameters of Nusselt equation for different Plate Heat Exchangers

PHE Length (m)	a	b	c	d
0.128	0.520	0.270	0.337	-0.1
0.210	0.523	0.210	0.400	-0.1
0.300	0.396	0.293	0.400	-0.1

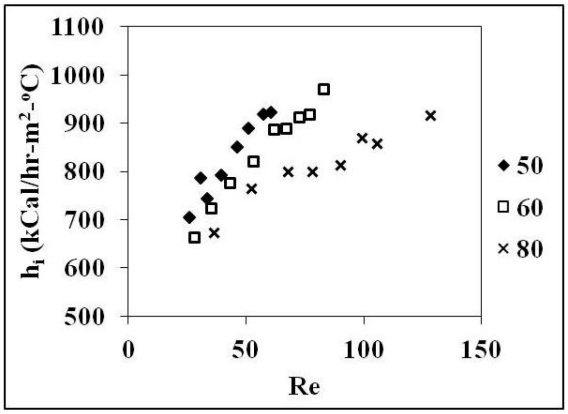

Figure 1: Effect of Temperature on the Heat Transfer Coefficient for PHE with 0.128 m length

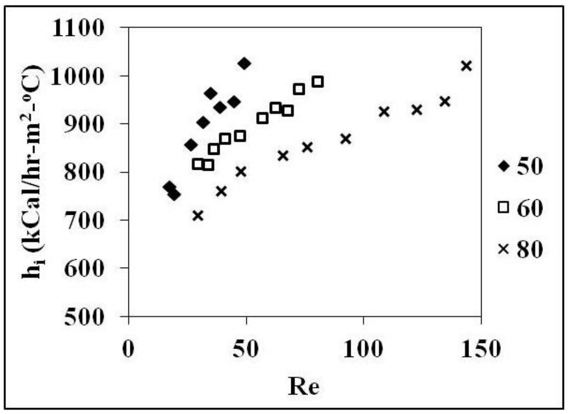

Figure 2: Effect of Temperature on the Heat Transfer Coefficient for PHE with 0.210 m length

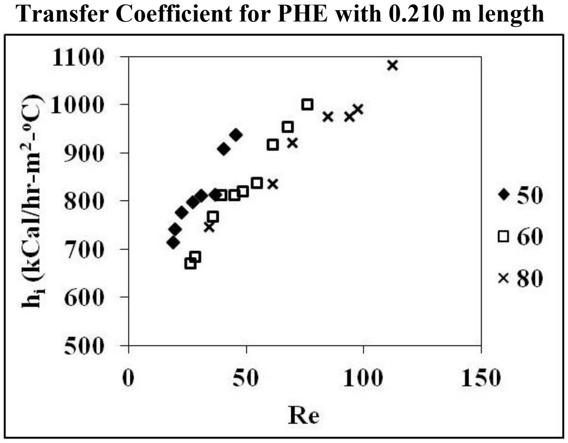

Figure 3: Effect of Temperature on the Heat Transfer Coefficient for PHE with 0.300 m length

$$Nu = 0.18 Re^{0.25} Pr^{0.4} \left(\frac{\mu}{\mu_w} \right)^{-0.099} \left(\frac{L}{W} \right)^{0.16} \quad (6)$$

Based on Eq. 6 it can be opined that the heat transfer coefficient is proportional to approximately 1/6th power of the length. The increase in the heat transfer with length is evident from Figures 1 to 3. Relative to Figure 1 the profile at 80 ^{0}C in Figure 3 is closer to those at 50 ^{0}C and 60 ^{0}C. The parity plot comparing the predicted values of heat transfer coefficient based on Eq. 2 with the experimental values is shown in Figure 4.

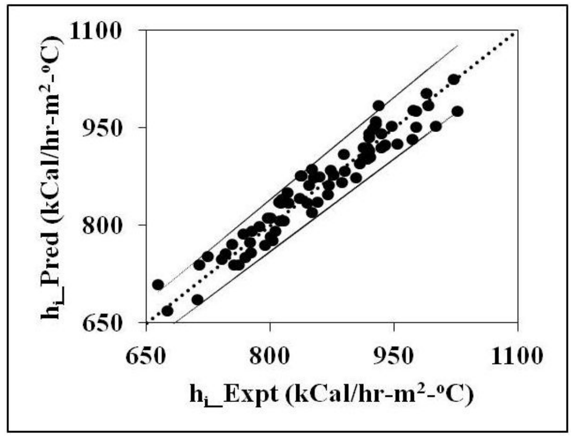

Figure 4: Parity plot for predicted and experimental values of heat transfer coefficient

CONCLUSIONS

The effect of temperature and the length of plate heat exchanger on the heat transfer coefficient were studied. Experiments were performed at 50 ^{0}C, 60 ^{0}C and 80 ^{0}C and three different lengths of plate heat exchanger. The decrease in the heat transfer coefficient with increase in the temperature was due to the conductive heat transfer dominating heat transfer. The experimental and the predicted values of heat transfer coefficient compare well with a maximum deviation of ± 5%.

Nomenclature

a	Model parameter, dimensionless
A	Constant in Eq. (5), m^3/hr-mol$^{1/3}$
b	Model parameter, dimensionless
B	Gap between two plates, m
c	Model parameter, dimensionless
C_p	Heat capacity, kCal/kg-^{0}C
d	Model parameter, dimensionless
h	Film heat transfer coefficient, kCal/hr-m^2-^{0}C
K	Thermal conductivity, kCal/hr-m-^{0}C
L	Length of plate, m
M	Molecular weight, kg/mol
n	Number of data points, dimensionless
Nu	Nusselt number, dimensionless
Pr	Prandtl number, dimensionless
Re	Reynolds number, dimensionless
U	Overall heat transfer coefficient, kCal/hr-m^2-^{0}C

Greek Letters

ρ Density, kg/m^3
μ viscosity, kg/m-s

Subscripts

i Inside
o Outside
w Wall

Superscripts

pred Predicted
exp Experimental

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