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Comparative Study of Heat Transfer and Flow Friction in Plate Heat Exchangers with CarboxyMethyl Cellulose and Sodium Alginate

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Abstract: Study of Plate heat exchanger (PHE) with power law fluid is a very interesting topic now a days as there is improvement of thermal performance of heat exchanger. Power law fluids show complex flow behavior due to its rheological properties. The parameters which differ these power law fluids from Newtonian fluids are rheological properties like flow behavior index and power law index. Hence flow behavior index and power law index are the two parameters which affect the heat transfer rate. Now to enhance rate of heat transfer we should know how Nusselt number changes with change in flow behavior index and power law index. Hence the aim of present study was to observe how Nusselt number and friction factor changes with change in flow behavior index and power law index. For this purpose thermal performance of PHE have been studied with CorboxyMethyl Cellulose (CMC) and sodium alginate as a cold fluids. Comparison of Nusselt number and friction factor of CMC and sodium alginate have been done. It was observed that Nusselt number for CMC solution was more compared to sodium alginate solution with less friction factor than sodium alginate.
Keywords: Plate Heat Exchanger, CorboxyMethyl Cellulose (CMC), sodium alginate, Nusselt number, friction factor.

I. INTRODUCTION

Power law fluids have attracted wide attention of the researchers in the last few decades. And study of power law fluids with Plate Heat Exchanger is the very interesting area for food industries as generally all the food fluids behave as a shear thinning power law fluids. Heat transfer processing of viscous non-Newtonian fluids is encountered in various industrial sectors including chemicals, petrochemicals, polymers, and pharmaceuticals. Comperehensive literature reviews on nonNewtonian fluid flow and heat transfer have been published by Skelland [1], Metzner [2], Lawal and Mujumdar [3], and Etemad [4]. Yoo [5] measured heat tarnsfer of nonNewtonian turbulent flow through circular tube with constant heat flux boundary condition. Joshi and Bergles [6] and Scirocco et al. [7] did an experimental investigation related the to laminar flow of pseudoplastic solutions in a circular tube subjected to a uniform wall heat flux. The results of Scirocco et al. [7] cover the turbulent regime as well.

Literature is also available on thermal performance of PHE with power law fluid. Carla et al. (2008) presented numerically study on laminar flows of Newtonian and power-law fluids through cross-corrugated chevron type plate heat exchangers (PHEs) in terms of the geometry of the channels. Warnakulasuriya et al. (2007) [8] investigated heat transfer and pressure drop characteristics of an absorbent salt solution in a commercial plate heat exchanger serving as a solution sub-cooler in the high loop of triple-effect absorption refrigeration cycle. Established the correlation equations to predict the heat transfer and pressure drop and to analyze and optimize the operating parameters for use in the design of absorption systems. Cabral et al. (2010) [9]have studied the pressure drop of pineapple juice in a PHE with 50º chevron plates. The pressure drop of pineapple juice flowing through a PHE with 50º chevron plates was determined for various flow rates (13–119 kg/s,) and temperatures (28–81ºC). Afonso *et al.* (2003) [10] conducted an experimental investigation to obtain a correlation for the determination of convective heat transfer coefficient in a PHE using yogurt as test fluid [11]. Based on the experimental studies of Afonso *et al.* (2003)[12], Fernandes *et al.*[13] (2005,2006, 2007 &2008) has demonstrated numerical studies on PHE with yogurt as test fluid. Carezzato *et a.l* (2007) [14] has also investigated the non-Newtonian heat transfer on a PHE with a generalized configuration, using CMC as a test solution [15]. Similar numerical studies in a PHE using egg yolk as test fluid were conducted by Jorge *et al.* (2003) [16]. Renato Cabral *et al.* (2010) [17]studied the thermo-physical (density) and flow properties (rheology) of pineapple juice in a PHE over a considerable range of temperature and soluble solid content to obtain a correlation for the determination of friction factor versus Reynolds Number [18].

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A better additive which enhances the rate of heat transfer has to be selected using the rheological parameters as the criteria for selection in order to best serve the food industry.

Hence in the present work comparative study of thermal performance of PHE have been carried out between CMC and sodium alginate as a cold fluid. Nusselt number and friction factor values of CMC, sodium alginate and water have been compared to study how shear thinning behavior affect the heat transfer. Data on Nusselt number and friction factor for CMC have been used published by Muthamizi et al. (2014)[19]. Model developed by Muthamizi et al. (2014) [19] also validated by using data experimental data in present work, and observed that it gives good results.

II. MATERIALS AND METHODS

A. Experimental Setup and Procedure

A six-channel corrugated type PHE with different flow rates and concentrations of working fluids sodium alginate was used in the present study. The apparatus shown in Fig.1consisted of a hot water storage tank, a cold fluid storage tank , immersed type of a pair of electrical heaters, a couple of liquid Rota meters, resistance temperature detectors, a manometer, two monoblock pumps and separate collection tank for cold fluids and hot water was recycled for reuse. A 25 liter capacity stainless steel tank for hot water storage was thermally well insulated to avoid heat loss to the atmosphere. Immersing type of electrical heaters of 4Kw capacity was fixed inside the hot water tank to raise the water temperature. A thermostat temperature controller with a range of 0°C to 110°C was connected with electrical heaters to set the temperature of hot water at a desired value.

Double pole on/off switch was connected with the 4Kw capacity electrical heaters. A monoblock type pump of 0.25 hp capacity was connected to the hot water storage tank to pump the hot water from the hot water storage tank to the PHE and a flow control valve in the same line was meant for regulating the flow. The cold fluid was stored in a separate stainless steel tank of equal capacity and well connected with another monoblock type pump of similar capacity used to pump cold fluid from cold fluid storage tank to PHE.A return flow line was provided to convey the cold fluid discharged at the outlet back into the collecting tank.

Two liquid Rotameters with an accuracy of ± 2% and measurement range of 0-10 LPM were well connected separately with the hot and the cold fluid lines to measure the fluid flow rate. These liquid flow meters were calibrated within their flow range.

Four Resistance Temperature Detectors (RTDs) of model type: PT 100 was mainly used to measure the inlet and outlet temperature of each fluid with an accuracy of ± 0.1°c. Out of the total four, two RTD's were separately placed at the inlet ports of both the fluids to measure the inlet fluid temperature and remaining two RTD's were separately placed at the outlet ports of both the fluids to measure the outlet fluid temperature. Digital temperature indicators with channel selectors connected with RTD's displayed the output results of RTDs. These RTDs were calibrated within their temperature range via the corresponding calibration procedure. Table 1 shows the specifications of PHE.

Sodium alginate pure (food grade) was provide by by LOBA Chemie, Merck & Himedia. For a constant hot fluid mass flow rate and concentration, experimental data were collected for different flow rates of sodium alginate solution (0.016-0.099 kg/s). Likewise, all runs were carried out for different concentrations (0.1-0.6%w/w) and different hot fluid mass flow rates (0.016-0.099 kg/s) as detailed in the experimental plan shown in Fig. 2.

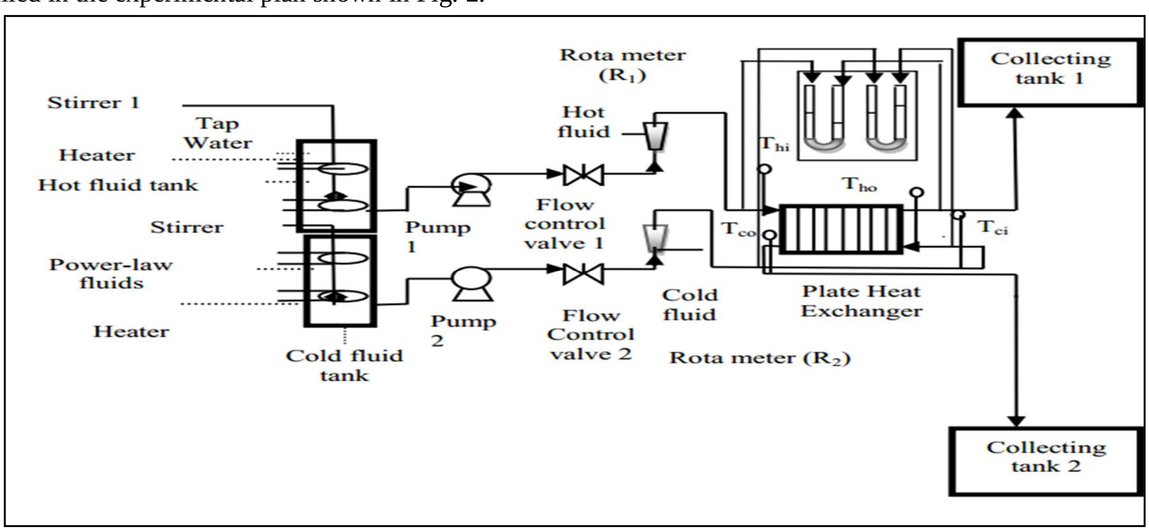

Fig 1. Experimental setup of PHE

Table 1. Specifications of PHE

Parameter	Value (m)
Plate thickness	0.0008
Plate width	0.125
Plate length	0.425
Port diameter	0.32
Channel spacing	0.004

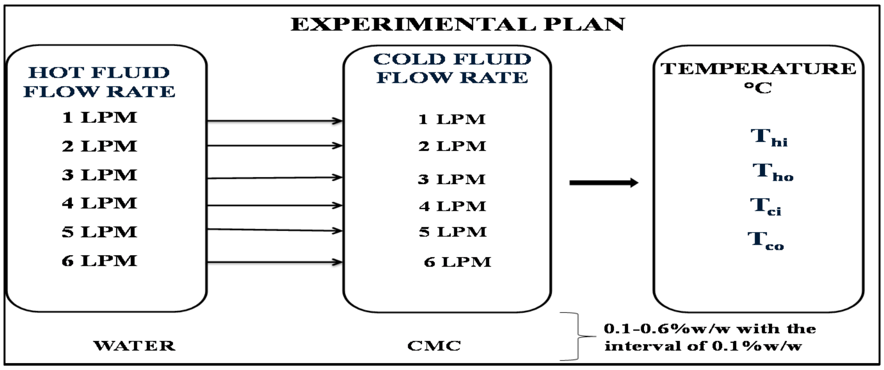

Fig 2. Experimental plan

B. Data Processing

The heat load, Q, of a PHE, can be represented by Eqs. 1(a, b and c).

$$Q_h = m_h Cp_h \left(T_{h,i} - T_{h,o}\right) \quad 1\ (a)$$

$$Q_c = m_c Cp_c \left(T_{c,o} - T_{c,i}\right) \quad 1\ (b)$$

$$Q = A_p U_{exp} \Delta T_{lm} F_T \quad 1\ (c)$$

$$\Delta T_{lm} = \frac{(T_{h,i}-T_{c,o})-(T_{h,o}-T_{c,i})}{\ln\frac{(T_{h,i}-T_{c,o})}{(T_{h,o}-T_{c,i})}} \quad 1\ (d)$$

where, m mass flow rate and Cp is specific heat of fluids, A is the effective PHE area calculated by multiplying area of one plate (length x width) by no. of plates for effective heat transfer (5), U_{exp} is the overall heat transfer coefficient, ΔT_{lm} is the logarithmic mean temperature difference and F_T is the correction factor [18]. All quantities were known except heat transfer rate ($Q=Q_h=Q_C$), which is calculated by using Eq. 1 (a&b). Thermo physical properties of sodium alginate solution thermal conductivity, density and specific heat were taken from the first phase study. The heat transfer rate calculated from above equation was then used to calculate the overall heat transfer coefficient U_{exp}, where ΔT_{lm} was calculated by Eq. 1(d). The correction factor F_T is a function of the exchanger configuration (number of transfer units (NTU, defined in Eq. (2) and heat capacity ratio (c^*, defined in Eq. (3)). For pure countercurrent flow ideal case, F_T =1. For all other types of flow distribution, $0<F_T<1$

$$NTU = \left(\frac{(N_c-1)A_p U_{exp}}{\min(m_h Cp_h, m_c Cp_c)}\right) \quad 2$$

$$C^* = \left(\frac{\max(m_h Cp_h, m_c Cp_c)}{\min(m_h Cp_h, m_c Cp_c)}\right) \quad 3$$

For the most usual configurations, researchers have utilized PHE simulation models to generate charts and tables in the form $F_T = F_T$ (NTU, c*) or ε = ε (NTU, c*), where ε is the thermal effectiveness of the exchanger [19]. Harika et al. has developed an effectiveness chart for 9 channels corrugated counter flow heat exchanger. [20]

$$\varepsilon = \frac{Q}{\min\left(m_h Cp_h, mCp_c\left(T_{h,i} - T_{c,i}\right)\right)} \quad 4\text{ (a)}$$

$$F_T = \begin{Bmatrix} \frac{1}{NTU(1-C^*)} \ln\left(\frac{1-C^*\varepsilon}{1-\varepsilon}\right) \text{ if } c^* < 1 \\ \frac{\varepsilon}{NTU-(1-\varepsilon)} \text{ if } c^* > 1 \end{Bmatrix} \quad 4\text{ (b)}$$

After the determination of U_{exp}, Eq. 5 was used to calculate cold side heat transfer coefficient. In which the convective coefficient of the water side h_{hot} of the PHE was obtained using Eq. 5, and same flow analysis as described by Vlasogiannis et al. (2002) [21]. Introducing pure water as the cold fluid and hot fluid as well, (h_h=h_c) the exchanger was first tested in a single-phase operation. To extract an accurate correlation for the hot fluid side heat transfer coefficient of the available PHE, a series of measurements were analyzed by the variant of the modified Wilson plot technique.

$$\frac{1}{U_{exp}} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{\Delta x}{k_{ss}} \quad 5$$

Calculated h_h was again substituted in Eq. 5 to predict cold side heat transfer coefficient (h_c) where Δx is the plate thickness and k_{ss} is the thermal conductivity of the plate material (Stainless steel) [22].

Nusselt number is then calculated from Eq. 6

$$Nu = \frac{h * D_h}{k} \quad 6$$

The pressure drop was calculated using Eq. (7) with the measured difference in the level of manometer fluid.

$$\Delta p = \rho g h \quad 7$$

Reynolds analogy is based on similarities between heat transfer and fluid friction (which causes the pressure drop). The simple analogy is valid only for fluids with Prandtl number equal to one. The Prandtl number expresses the relative magnitude of diffusion of momentum and heat in the fluid and thus the heat and momentum are transported at the same rate. This is not applicable to plate heat exchangers as the flow is generally turbulent with random transportation of heat and momentum. Hence the modified Reynolds analogy Colburn factor (j), which gives an approximate rationalization over a wide range of Prandlt numbers as given in Eq. (8) has been used for our calculation.

$$j = StPr^{2/3} = \frac{Nu}{RePr^{1/3}} = \frac{f}{2} \quad 8$$

The surface performance is assumed to be describable by Colburn factor (j) and friction factor (f) as a function of Reynolds number. The pressure drop of a fluid through a surface in terms of the friction factor is

$$\Delta p = \frac{1}{2}\rho u^2 \frac{4L}{d_m} f, \text{where } \{f = f(Re)\} \quad 9$$

Inserting friction factor f from Eq. (9) into Eq. (10) gives

$$j = \frac{\Delta p d_h}{4\rho u^2 L} = \frac{Nu}{RePr^{1/3}} \quad 10$$

Pressure drop, Δp were calculated using Eq. (7), then these values were used to calculate the friction factor values from equation (9) and the calculated friction factor values have been used for further analysis.

C. Nusselt number correlation used

Muthamizhi et al. (2014) [19] has developed the Nusselt number correlation for a counter flow six-channel corrugated type PHE with CMC as a working fluid. Dimensional analysis has been used to develop the Nusselt number correlation which is the mathematical technique of deriving relations between physical quantities by identifying their dimensions. Developed correlation also take care the rheological properties of CMC which affect the heat transfer.

The following expression of heat transfer coefficient in term of Nusselt number was reported by Muthamizhi et al. (2014)

$$\mathbf{Nu} = \mathbf{f}\left(\left(\frac{U}{Dh^{\frac{n}{(n-2)}}\rho^{1/(n-2)}}\right)\left(\frac{cp}{Dh^{\frac{2(1-n)}{(n-2)}}\rho^{\frac{(1-n)}{(n-2)}} K}\right)\left(\frac{\Delta T\beta g}{Dh^{\frac{(n+2)}{(n-2}}\rho^{\frac{2}{n-2}}}\right)\right) \quad 11$$

The constants and powers of the parameters involved in the correlation developed were estimated using least square method. An excel tool function has been used to determine the effect of each variable involved in the Eq. 11 of Nusselt number.

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LINEST(least squares ,method) was used to calculate the statistics for a line to calculate straight line that best fits the data and then returns an array that describes the line. Equation (11) was fitted to the set of 36 experimental data points obtained using counter flow six-channel corrugated type PHE with CMC as a cold fluid. After finding the coefficients and constant equation (11) reduces to the following form:

$$\mathbf{Nu_x} = \mathbf{0.415834}\left(\left(\frac{U}{D_h^{\frac{n}{(n-2)}}\rho^{1/(n-2)}}\right)^{0.651139}\left(\frac{cp}{D_h^{\frac{2(1-n)}{(n-2)}}\rho^{\frac{(1-n)}{(n-2)}}K}\right)^{2.267862}\left(\frac{\Delta T\beta g}{D_h^{\frac{(n+2)}{(n-2)}}\rho^{\frac{2}{(n-2)}}}\right)^{-0.15291}\right) \tag{12}$$

This developed model has been validated by using experimental data in present study and found that it gives good results.

III. RESULTS AND DISCUSSION

A. Comparison of Nusselt Number of Cmc, Sodium Alginate Solution and Water

The calculated Nusselts numbers for sodium alginate and CMC reported by Muthanizhi et al. (2014) are plotted against cold side Reynolds number for 0.1, 0.3 and 0.5%w/w concentration for hot fluid mass flow rates of 0.016 kg/s Fig.3. The Nusslet number increases with both cold sides Reynolds number as well as with concentration for both fluids. The reason is higher Reynolds number results in turbulence leading to increase in Nusselt number. Properties like density, specific heat and viscosity increases with concentration except thermal conductivity of the Sodium alginate solution which reduces with concentration. The Nusselt number is proportional to the raised power of Reynolds and Prandtlt number and this in turn depends on physical properties and hence Nusselt number also increases with increase in concentration. Reynolds number varies from 7.25 to 64.20 and the resulting Nusselt number varies from 6.74 to 75.07.

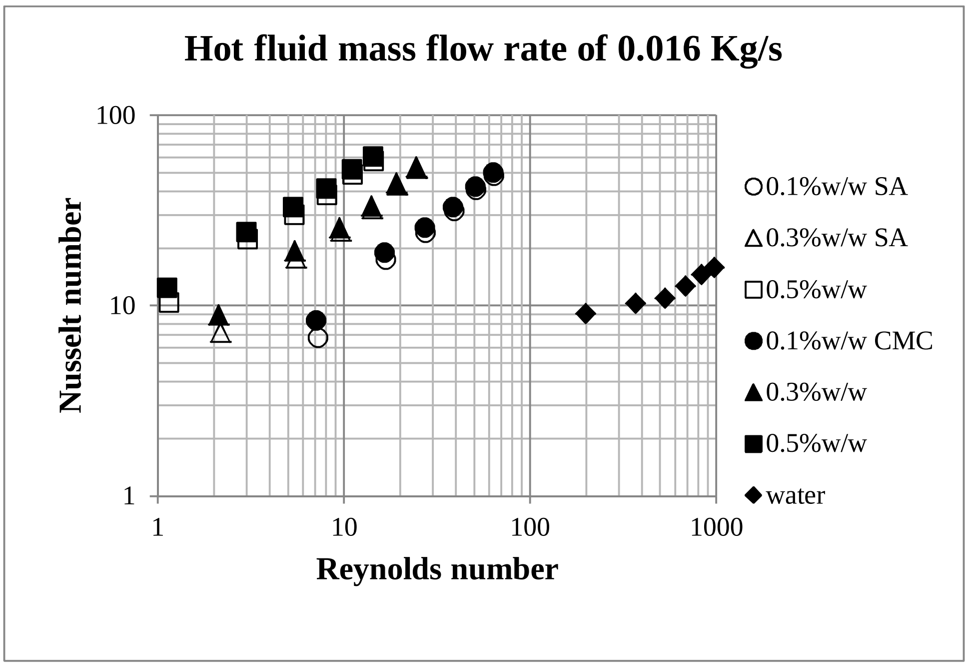

Fig. 3 Comparison of Nusselt number of CMC, sodium alginate and water.

Figures 3 show that heat transfer is enhanced due to presence of sodium alginate and CMC. The Nusselts number values calculated using pure water as working fluid in PHE was also incorporated in these figure for comparison. It is observed that there was considerable increase in Nusselt number i.e heat transfer even at low Reynolds number due to additives like sodium alginate and CMC. Addition of sodium alginate and CMC enhances the thermo physical properties of water which results in increase in heat transfer. Also comparisons have been done between Nusselt number of sodium alginate and CMC. From graph it is observed that Nusselt number for CMC is more than sodium alginate. Heat transfer coefficient depends on Reynolds number and power law index. It increases with increase in Reynolds number and decrease in power law index. As CMC is having low values of power law index than sodium alginate which results in higher Nusselt number than sodium alginate. Similar trend was found for all studied mass flow rates of hot fluid.

B. Comparison Of Fricton Factor Of CMC, Sodium Alginate Solution And Water

The friction factor as a function of cold side fluid Reynolds number is plotted for different sodium alginate concentrations (0.1, 0.3 and 0.5% w/w) and at 0.016 kg/s flow rate of hot fluid as shown in the Fig. 4. In the same graph the friction factor values reported by muthamizhi et al. (2014) for CMC solution and friction factor for water have been incorporated for comparison. It can be observed from these figures that there is a drop in friction factor with increase in cold side Reynolds number and friction factor increases with increase in concentration for both sodium alginate and CMC solutions. The inverse proportionality correlation between friction factor and velocity results in decrease of friction factor with increase in Reynolds number i.e. with increase in velocity of cold side fluid. The friction factor is proportional to pressure drop (Eq. 9) as described in chapter 3, which in turn depends on density and viscosity of the fluid.

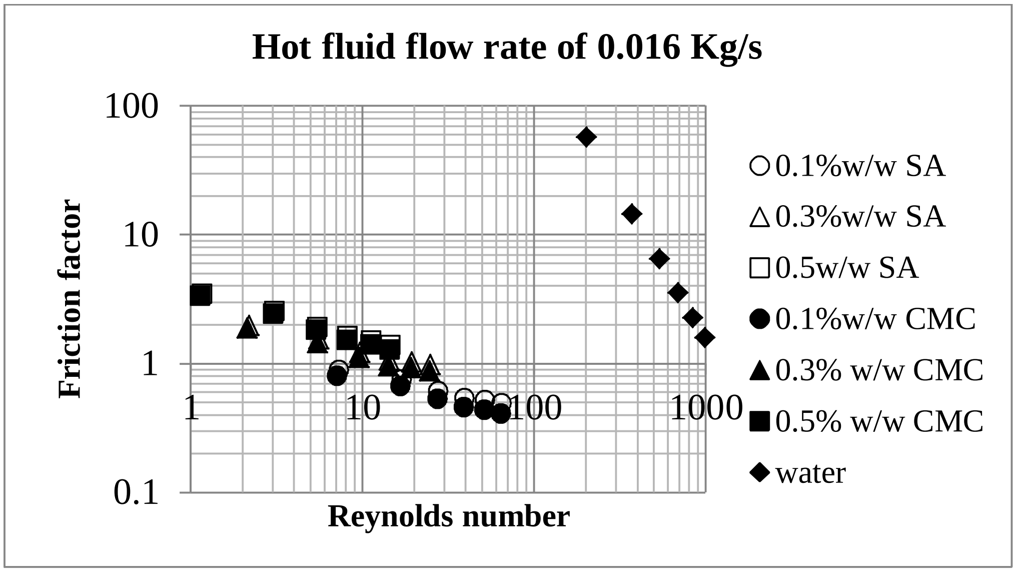

Fig. 4 Comparison of friction factor of CMC, sodium alginate and water.

As pressure drop increases with increase in viscosity which is a result of increase in concentration. Hence friction factor increase with increase in concentration. The friction factor of Sodium alginate and CMC solutions for 0.1, 0.3 and 0.5%w/w concentration have been compared with friction factor of water through plots between friction factor and Reynolds number (Fig 5). Friction factor in PHE with sodium alginate and CMC solutions is observed to be lower than pure water, which shows good agreement with the results reported by Fernandes et al.(2008) who have reported that friction factor of sheer thinning fluids are lower than those from Newtonian fluids. It is observed that friction factor of sodium alginate is more than CMC. Friction factor depends on power law index that is shear thinning behavior as drag coefficients which leads to skin friction decreases with decrease in power law index. This shows good agreement with the results reported by Khan et al. (2006) [10] who has concluded that pseudoplastic fluids offers less skin friction than Newtonian fluid. Similar trend was found for all studied mass flow rates of hot fluid.

3.3 Validation of model

Model developed by Muthamizi et al 2014. Have been validated by using experimental data in present study for PHE with sodium alginate solution as a cold fluid. The Root Mean Square (RMS) deviation between the experimental and estimated Nusselt Number was found to be 16.87. Whereas the RMS deviation of 14.61 was reported for CMC solution. Hence the developed model for CMC solution can be used with little more error for sodium alginate solution and even for any power law fluid, instead of spending time on experimental work, and without wasting money.

IV. CONCLUSION

In this work comparison has been done between two power law fluids to select better additive. Is was found that Nusselt number increases with decrease in power law index, it shows heat augmentation is more for fluids which shows more shear thinning behaviour. Another benefit of these shear thinning power law fluids is they provide less friction than Newtonian fluids, also friction decreases with more shear thinning behaviour. Shear thinning behaviour increases with decreases in power law index. So for better heat transfer we can chose the power law fluid with law power law index.

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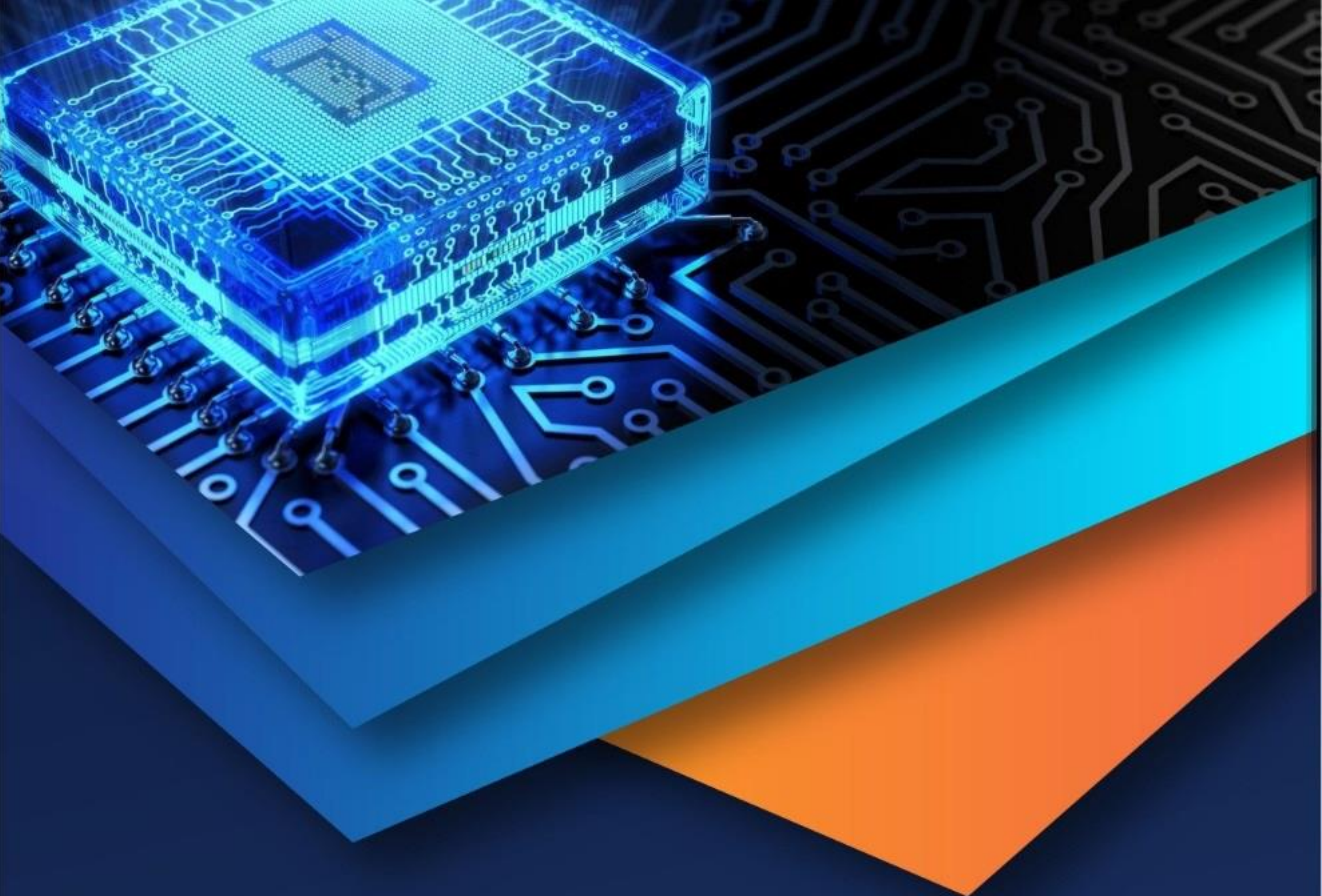

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