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Research Paper

LAMINAR FORCED CONVECTION TO FLUIDS IN COILED PIPE SUBMERGED IN AGITATED VESSEL

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Laminar forced convection in curved pipe placed in agitated vessel has been studied experimentally for the flow of both Newtonian fluid (water) and power law non-Newtonian fluids (5%, 1% and 2% aqueous CMC solutions. The heat transfer has been calculated using modified Wilson method. It is found that the Nusselt number is function of Dean and Prandtl numbers. The following correlation is found for flow of Newtonian and power law non-Newtonian fluids flowing through helical coils.

$$\frac{N_{uic}}{N_{uis}} = 1 | 0.066 N_{D2}^{1/2} N_{Re}^{0.12}$$

Keywords: Agitated vessel, Newtonian, Non-newtonian power law, Wilson method

INTRODUCTION

Laminar forced convection in curved pipes has received considerable attention. From the work of Kubair and Kuloor (1966), Rajshekharan *et al.* (1966 and 1970), Akiyama and Cheng (1971) and Dravid *et al.* (1971), it is found that not much work has been done for non-Newtonian fluids in curved pipes. The existing heat transfer data in the literature for fully developed laminar forced convection in curved pipe with uniform wall temperature are rather limited and incomplete. For Newtonian fluids, perturbation method was applied by Maekawa for extremely low Dean Number. The boundary layer approximation near the wall was presented by Akiyama and Cheng (1971) for high dean number of order one. Dravid *et al.* (1971) presented the numerical results in the thermal entrance region for Dean Number less than 225 and $N_{Pr} = 5$. The improved later work of Akiyama and Cheng (1972) shows that the ratio of heat transfer coefficient in coil and in straight pipe is a function of Dean Number as well as Prandtl number.

Thus there is a possibility of obtaining a suitable correlation of the following form

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$$\frac{N_{Nuic}}{N_{Nuis}} - 1 = \varnothing_1(N_D, N_{Pr}) \quad ...(1)$$

The advantage of the above form of correlation is that for small N_D, the Equation (1) satisfies the condition that $N_{Nuic} = N_{Nuis}$ for very small N_D. N_{Nuic} may be calculated for isothermal laminar heat transfer in straight tube for the case of uniform wall temperature and the parabolic velocity distribution by following correlation as suggested by Mujawar and Raja Rao (1997).

$$N_{Nuis} = 1.75(N_{Gz})^{1/3} \quad ...(2)$$

For non-Newtonian fluids the above equation takes the form

$$N_{Nuis} = 1.75\left(\frac{3n+1}{4n}\right)(N_{Gz})^{1/3} \quad ...(3)$$

Here again it is important to note that dean number contain Reynolds number and the viscosity term appears in both Reynolds and Prandtl numbers. It is suggested that the effective viscosity at the shear stress prevailing at the wall should be used for evaluating the Reynolds and Prandtl numbers. From the above discussion it may be concluded that the correlation may be written as follows

$$\frac{N_{Nuic}}{N_{Nuis}} - 1 = C_3 N_{D2}^{b5} N_{\Pr 2}^{b6} \quad ...(4)$$

The fluid heat transfer coefficient h_{oc} were calculated from overall coefficient U_{oc} using Wilson graphical method. $\frac{1}{U_{oc}}$ versus $\frac{1}{N^{2/3}}$ plot for water gives y_c equal to 0.00012. For different combinations of aqueous CMC solutions in the agitated vessel and flowing through the coils $\frac{1}{U_{oc}}$ is plotted against $\frac{1}{N^{(3-n)/3}}$ yeieled separate straight lines for different blade diameters. The respective y_c values were found to be 0.000445, 0.000575 and 0.000905 respectively.

The outside heat transfer coefficient of the coil tube from the coil side over all heat transfer coefficient U_{oe} was calculated by following relation.

$$\frac{1}{U_{oc}} = \frac{1}{h_{oc}} + Y_c \quad ...(5)$$

where

$$Y_c = \frac{1}{h_{ic}}\left(\frac{D_c}{D_t}\right) + \frac{\Delta_x D_c}{K_m D_{tm}} + dirt\ resistance \quad ...(6)$$

and

$$h_{oc} = Y_c N^{(3-n/3)} \quad ...(7)$$

The calculated values of $\frac{1}{U_{oc}}$ and hoc for 0.5% CMC, 1% CMC, 2% CMC that of 4% CMC solution and that of water for blade diameter 7.5 cm, 12.7 cm and 18.35 cm have been used in the present work.

The liquid film coefficient inside the coiled tube, i.e., h_{ic} was calculated by the relation

$$\frac{1}{h_{ic}} = \left(\frac{1}{U_{oc}} - \frac{1}{h_{oc}}\right)\frac{D_t}{D_o} \quad ...(8)$$

For a set of observation the fluid flow rate in the coil was varied at a constant rotational speed of the agitator and for that set of readings h_{oc} was maintained constant. The value of h_{oc} was obtained from the Wilson plot according to the relation

$$\frac{1}{h_{oc}} = \frac{1}{U_{oc}} - Y_c \quad ...(9)$$

EXPERIMENTAL PROCEDURE

The experimental setup consisted of a flat bottomed cylindrical test vessel of 45.25 cm inner diameter and 60 cm. heights made from 1/8 inch thick copper sheet. The vessel was jacketed by providing annular space surrounding it with another cylinder of 44 cm height and made from 1/8 inch thick G.I sheet. The jacket and vessel assembly was adequately insulated from outside. A helical coil having 34 cm mean coil diameter and made from 1.898 cm internal diameter and 2.25 cm outer diameter copper tubing was placed in the centre of the test vessel.

A rectangular tank of about 300 litre capacity filled with heaters was used to heat the water to a pre-determined temperature which in turn was circulated through the jacket (annular space around the test vessel) with the help of a centrifugal pump.

The fluid in the test vessel was agitated by marine type agitator fitted in the centre of the coil. The agitator shaft was driven at a known speed by a 2HP electric motor through a reduction gear assembly. Provision was made for replacement of impeller of desired shape and size.

The water in the hot water tank was heated and the temperature of fluid was brought to the desired level. The temperature was controlled to a pre determined value with the help of temperature controller. The water circulation was then started in the coil as well as in the jacket and adjusted by means of regulating values and bypasses. The agitator was then started at a fixed rpm.

At steady state condition the inlet and outlet water temperatures in the jacket, mass rate of water flow through jacket and that of fluid in the coil, rpm of the agitator and temperatures of fluid in the agitated vessel and that of water in the storage tank were noted. The test fluid side wall temperatures of the test vessel were noted at different locations and at different heights from its bottom with the help of copper constantan thermocouples.

The readings were duplicated to ensure the steady state and to eliminate any error in measurement. Similar measurements were made by varying the flow rate in the coil and then by varying the rotational speed of the agitator. The above procedure was repeated for all the fluids.

The heat transfer characteristics of five fluids, viz., water and four aqueous CMC solutions of concentration 0.5, 1, 2 and 4% by weight have been investigated. The rheological properties were determined with the help of capillary tube viscometer. All the CMC solutions were found to be psuedoplastic in nature obeying power law relation.

The flow behavior indices were found to be 0.937, 0.851, 0.793, and 0.698 for 0.5%, 1%, 2% and 4% aqueous CMC solutions respectively. Thermal conductivity of CMC solutions were determined by comparative concentric cylinders and were found to be equal to that of water. Specific heats of the solutions, as measured by calorimetric method, were found to be nearly equal to that of water.

RESULTS AND DISCUSSION

In order to obtain exponent b_5 and b_6 and constant C_3 in Equation (4) the proposed form of correlation for laminar forced convection to non-Newtonian fluids in helical coils 0.5, 1 and

2% CMC solutions were investigated in a helical coil of 34 cm diameter made from a tube of diameter 1.898 cm. Before conducting the heat transfer runs, pressure drop measurements were made to verify the fitness of the coil. Coil flow is compared with capillary shear flow in Figure 1. Friction factor f_c is shown in Figure 2 against Reynolds number N_{Re_2} defined by the following equation.

$$N_{Re_2} = \frac{D_t U_p}{\nu_2} \qquad \text{...(10)}$$

Figure 1: Flow Diagram for 0.5%, 1% and 2% CMCA Through HT Test Coil

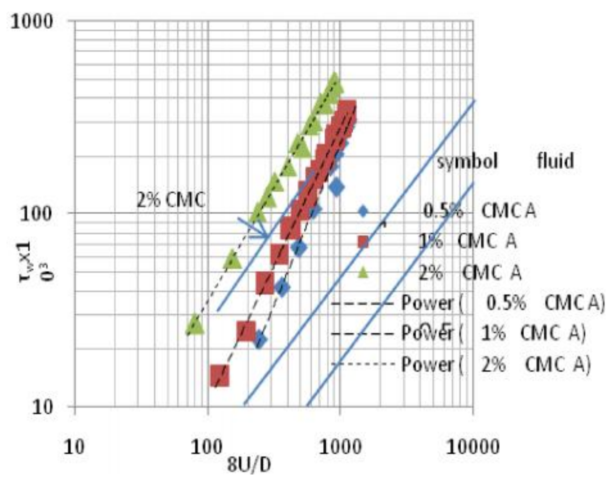


Figure 2: Variation of Friction Factor with Reynolds Number for Water, 2%, 1% and 0.5% CMCA Through HT Test Coil

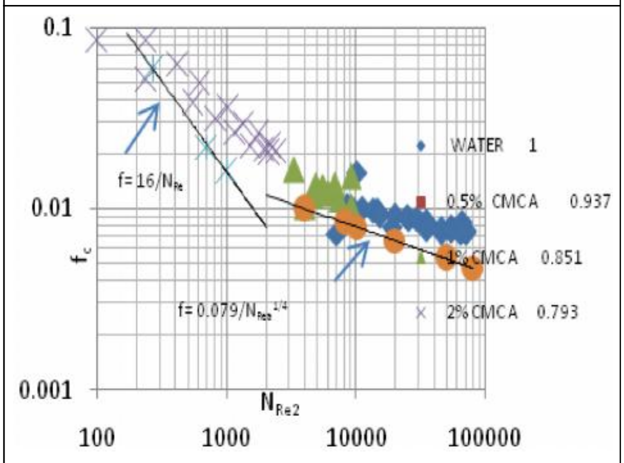
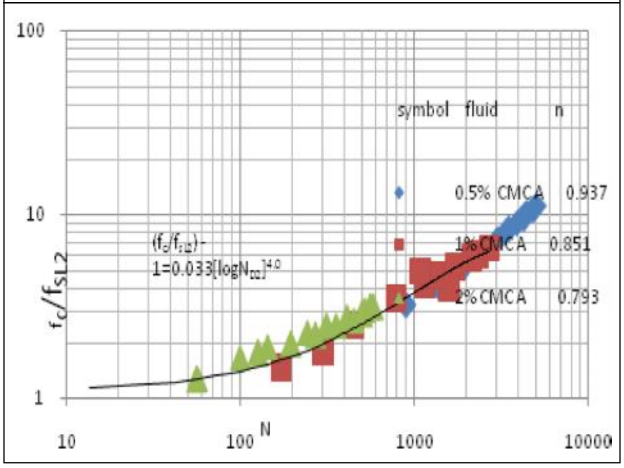


Figure 3: Laminar Flow Friction Factor in Helical Coils as a Function of Dean Number for Non-Newtonial Fuids



The ratio of friction factor, f_c, for coil to that of straight pipe, f_{SL} evaluated at Reynolds number, N_{Re_2}, from equation $f_{SL} = 16/N_{Re_2}$ is plotted against N_{D_2} in Figure 3.

$$\left(\frac{f_c}{f_{sl}}\right) - 1 = 0.33\left(\log N_{D_2}\right)^{4.0} \qquad \text{...(11)}$$

The Equation (11) proposed by Berger *et al.* (1983) is also shown in this figure. The excellent agreement of the data with above equation indicates the fitness of the coil and the heat transfer data obtained in this coil could be placed for its accuracy.

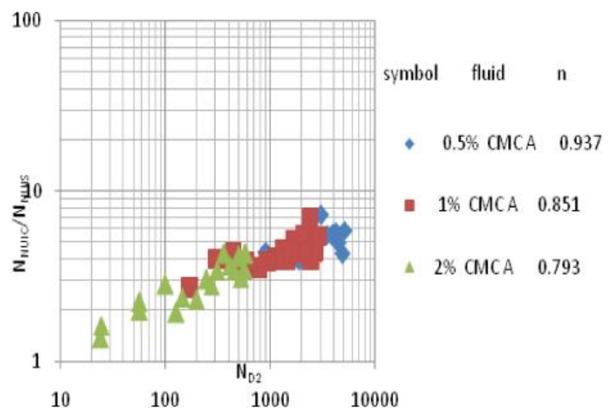
Three non-Newtonian fluids: 0.5% (n = 0.937), 1% (n = 0.851) and 2% (n = 0.793) CMC in water, representative of pseudo plastic fluids obeying power law, and water, a Newtonian fluids were investigated. Most of the data, obtained, were in laminar region and only a few non-Newtonian data could fall in turbulent region.

Om Prakash *et al.* have shown that the fluid flow data in helical coils over wide range of coil diameter and fluid behavior can successfully be correlated by the use of apparent viscosity μ_z evaluated at the wall shear stress in Reynolds number N_{Re_2}. This suggests the possibility of analogous correlation in terms of the ratio $\left(\frac{N_{Nuic}}{N_{Nuis}}\right)$. N_{Nuis} was evaluated by Equation (3).

Effect of Dean Number

In order to find the effect on the ratio of heat transfer in coil to heat transfer in straight pipe, the first attempt was to plot the ratio $\left(\frac{N_{Nuic}}{N_{Nuis}}\right)$ against N_{D_2} on a logarithmic scale as shown in Figure 4.

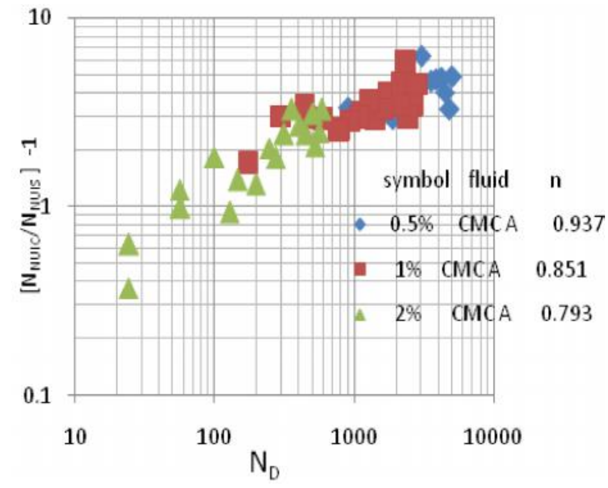
Figure 4: Variation of N_{Nuic}/N_{Nuis} with N_{D2} for Heat Transfer to Non-Newtonian Fluids



No direct relationship is seen between the two variables in this manner. At very low Dean Number or very small curvature, effect of secondary flow will be negligible and then N_{Nuic} will be equal to N_{Nuis}. Therefore, the next attempt was to plot $\left(\frac{N_{Nuic}}{N_{Nuis}}\right) - 1$ against N_{D_2}. This plot is shown in Figure 5.

No net conclusion could be drawn even from this figure except for some qualitative

Figure 5: Variation of $[(N_{Nuic}/N_{Nuis}) - 1]$ with N_{D2} for Heat Transfer to Non-Newtonian Fluids



ones. The heat transfer coefficient in coil is always higher than in straight pipe as it is seen that $\frac{N_{Nuic}}{N_{Nuis}}$ is always greater than unity. The ratio h_{ic}/h_{is} increases with Dean number, N_{D_2}. These data do not follow any particular trend. This may be explained by the fact that due to non Newtonian behavior Prandtl number changes by change of flow rate affecting the heat transfer rate. Data plotted for three fluids in Figure 5 have wide range of Prandtl number. However, an exponent of Dean Number ½ correlated the data much better. This trial has been made on the basis of theoretical results presented by previous investigators.

Effect of Prandtl Number

Prandtl number N_{pr2} was calculated by using apparent viscosity, «$_2$ evaluated at wall shear stress, $^2{}_w$, obtained from the measured pressure drop. The «$_2$ may be evaluated by the following derived expression

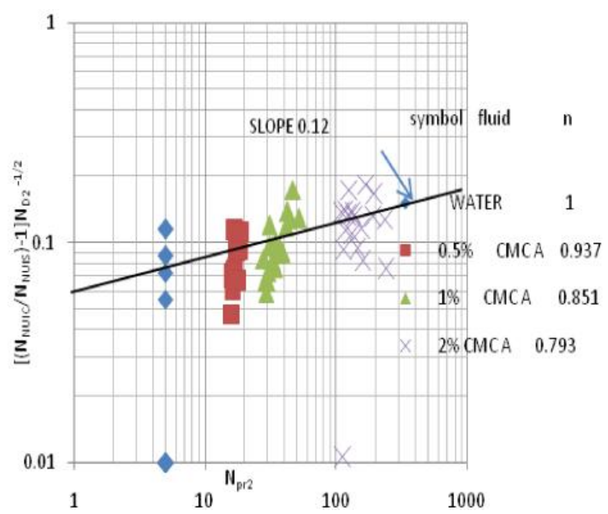
$$«_2 = k'({}^2{}_w / k')\frac{n-1}{n}$$

Subsequently $N_{Re_2} = DtU°/«_2$ and

$$N_{D_2} = N_{Re_2} (D_t/D_c)^{1/2}$$

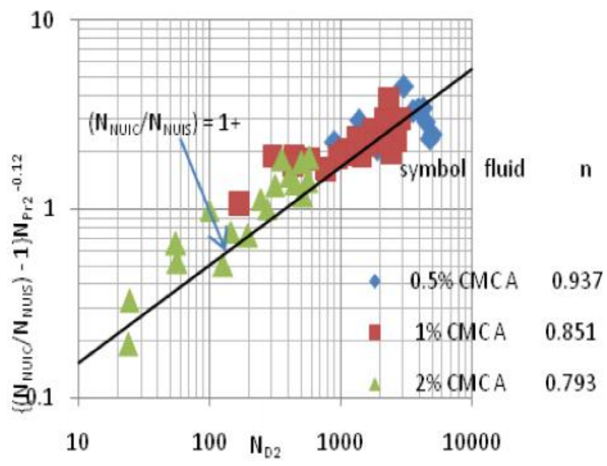
In order to understand the effect of N_{pr2} on the ratio h_{ic}/h_{is}, a logarithmic plot of $\left(\frac{N_{uic}}{N_{uis}}\right)^{-1} N_D^{-1/2}$ versus N_{pr2} is shown in Figure 6. Figure 6 clearly shows the effect of N_{pr2}. Many other trails were made by changing the exponent of Dean number N_{D_2}, in the ordinate variable of Figure 6.

Figure 6: Laminar Flow Heat Transfer Correlation for Non-Newtonian Fuids Through Coils: Effect of Prandtl Number



Values of exponent other than ½ on N_{D_2} tend to increase the scatter of data points. The slope of mean line shown represents the exponent of N_{pr2} and was estimated to be 0.12 for Prandtl number range of 4.9 to 225. Finally $\left(\frac{N_{uic}}{N_{uis}} - 1\right) N_{pr2}^{-0.12}$ is plotted against N_{D_2} in Figure 7 to verify the exponent of N_{D_2}. The mean line passing through data points show an exponent equal to ½. The resulting correlation is obtained as:

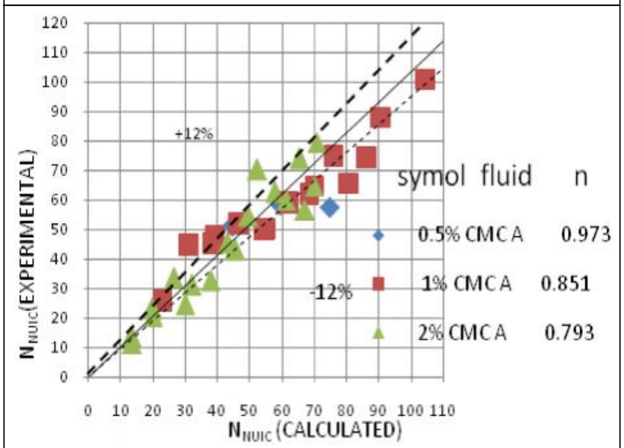
Figure 7: Laminar Flow Heat Transfer Correlation for Non-Newtonian Fuids Through Coils



$$\frac{N_{uic}}{N_{uis}} = 1 + 0.066 N_{D2}^{1/2} N_{pr2}^{0.12} \qquad \text{...(12)}$$

Figure 8 compares the experimental Nusselt number with corresponding values calculated from Equation (12). It is found that that laminar flow heat transfer data in the range $24 < N_D < 2000$, $N_{pr2} < 225$ and $0.793 < n \leq 1$ can successfully be correlated using Equation (12) with a standard deviation of 15.5%.

Figure 8: Comparison Between Experimental and Calculated Values of Laminar Nusselt Number N_{NUIC} (Heat Transfer to Non-Newtonian Fluids Through Coil)



According to Ozisik and Topa kaglu's theoretical approach (08), there is possibly of two regions to exist, namely the heated and cooled region in the fluid space and that too depends upon Prandtl number.

This equation is similar to the equation suggested by Mori and Nakjayama (09) for flow of Newtonian fluids across tube banks where the ratio $\frac{N_{uic}}{N_{uis}}$ is the function of N_{pr2} and N_{D_2}.

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APPENDIX

Nomenclature

b_5, b_6	constant in Equations (4)
C_3	constant in Equation (4)
D_c	diameter of the coil helix, cm
D_t	inner diameter of the straight or coil tube
D_o	outer diameter of the straight or coil tube
f_{sL}	laminar flow friction factor in straight pipe
f_c	friction factot in coil
h_{ic}	coil inside heat transfer coefficient, kcal/hr m^2 °C
h_{oc}	coil outside heat transfer coefficient, kcal/hr m^2 °C
k’	consistency index, gm sec n’-2/cm
N	speed (r.p.m)
N_{pr}	Prandtl number
N_{NUIC}	Nusselt number, $h_{ic}D_t$/k
N_{NUIS}	Nusselt number, $h_{is}D_t$/k
N_D	Dean number
N_{GZ}	Graetz number, wc_p/kL
N	speed (r.p.m)
n’	generalized flow behavior index
U_{OC}	coil overall heat transfer coefficient, Kcal/hr m^2 °C
h_{OC}	coil outside heat transfer coefficient, Kcal/hr m^2 °C
h_{ic}	coil inside heat transfer coefficient, Kcal/hr m^2 °C
Y_C	resistance of the jacket side fluid and the wall
Δ_X	width of the agitated vessel wall, cm
N_{Re_2}	Reynolds number defined by Equation (10)
U	average velocity, cm
P	density, gm/cm^3
«$_2$	effective or pseudo shear viscosity, gm/cm
²$_W$	shear stress, gm(f)/cm^2
«$_e$	effective viscosity
«”$_a$	apparent viscosity at the impeller tip