EFFECT OF DEAN NUMBER ON HEAT TRANSFER COEFFICIENTS IN COILED TUBE AGITATED VESSEL

Ansar Ali sk , Vipin Sharma
Meerut Institute of Engineering and Technology, Meerut, India

Ashok Kumar Ojha
Saraswati Institute of Engineering and Technology, Hapur, INDIA

ABSTRACT -The purpose of the experimental phase of the present investigation is to obtain data for heat transfer to purely viscous non-Newtonian fluids in coil and agitated vessel. The set up consisted of a flat bottomed cylindrical test vessel 'A' of 45.25 cm inner diameter and 60 cm height, made from 1/8 inch thick copper sheet. An annular space 'F' around the test vessel was provided by surrounding it with another cylinder of 44 cm height and made from 1/8 inch thick G.I. sheet. The jacket and the vessel assembly was adequately insulated by keeping it in a wooden tank and then filling the surrounding space with glass wool and thermocouple. Three aqueous solutions containing 0.5% (n=0.973), 1% (n= 0.851) and 2% (n=0.793) CMC and water were studied in the coil and 1%, 2% and 4% (n=0.698) aqueous solutions of CMC in the vessel. In our present study h_{ic} / h_{is} Vs N_{D2} , N_{Nuic} / N_{Nuis} Vs N_{D2} were studied.

***KEYWORDS* -Dean Number, Agitated Vessel, helical coil, CMC solution, heat transfer coefficient**

INTRODUCTION

Several investigators have studied the average heat transfer from helical coils in the mixing vessels. The heat transfer rate at the coil surface may be written as

$$Q = -k \int_A \left(\frac{\partial T_i}{\partial x}\right)_s dA \qquad \textbf{(1.1)}$$

Where A is the total surface area and x is the distance measured outward from and normal to the coil surface. Flow from the impeller passing around the coil surface, forms boundary layer which in turn offers resistance to heat transfer. Thus the velocity and temperature profile will depend on the impeller Reynolds number, Prandtl number and the diameter of the coil tube. On the basis of the discussions made it can be shown that the temperature gradient is a function of (N''_{Rea}) and(N''_{Pra}). Thus, in order to describe the heat transfer from the coil surface, a correlation can be developed between the Nusselt number$\left(h_{oc} \ {}^{D_t}/_{k}\right)$, Reynolds number, (N''_{Rea}), and Prandtl number, (N''_{Pra}). For coils in mixing vessel and for given geometrical situation and isothermal condition this results in the following equation

$$\frac{h_{oc} \ D_o}{k} = C_2 \ (N''_{Rea})^{b_3} \ (N''_{Pra})^{b_4} \qquad \textbf{(1.2)}$$

Laminar forced convection in curved pipes has received considerable attention in recent years. From the work of Akiyama and his co-workers (1971), Dravid (1971), Kubair and Kuloor (1966) and Rajshekhran et al. (1966), it is seen that very little work has been done for non-Newtonian fluids in curved pipes. The existing heat transfer results 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 Mori and Nakayama (1965) 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 recent and improved results of Akiyama (1971) shows that the ratio of heat transfer coefficients in coil and in straight pipe is a function of Dean Number as well as Prandtl number.

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

$$\frac{N_{Nuic}}{N_{Nuis}} - 1 = \emptyset_1 \ (N_D \, , N_{Pr}) \qquad (1.\textbf{3})$$

The advantage of the above form of correlation is that for small N_D, the equation satisfies the condition that $N_{Nuic} \approx N_{Nuis}$ for very very small N_D. N_{Nuis} may be calculated for isothermal laminar heat transfer in straight tube for the case of uniform wall temperature and parabolic velocity distribution by the following correlation:

$$N_{Nuis} = 1.75 \ (N_{Gz})^{\frac{1}{3}} \qquad \textbf{(1.4)}$$

For non-Newtonian fluids equations (1.3) and (1.4) take the following form:

$$N_{Nuis} = 1.75 \left(\frac{3n+1}{4n}\right) (N_{GZ})^{\frac{1}{3}} \qquad \textbf{(1.5)}$$

Here again it is important to note that Dean Number contains Reynolds number and the viscosity term appears in both the Reynolds and Prandtl numbers. It is suggested that the effective viscosity μ_2 at the shear stress prevailing at the wall should be used for evaluating the Reynolds and Prandtl numbers. From the above discussions it may be concluded that the correlation may be written as below:

$$\frac{N_{Nuic}}{N_{Nuis}} - 1 = C_3 \, N_{D_2}^{b_5} \, N_{Pr_2}^{b_6} \quad \textbf{(1.6)}$$

EXPERIMENT

The following observations were recorded.

Inlet and exit fluid temperatures in the jacket and the coil, amount of the water from the jacket and the coil collected separately for a known time interval, rpm of the agitator, temperature of the fluid in the agitated vessel and that in the storage tanks.

The readings were duplicated to ensure the steady state and to eliminate any error in the 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 with all the non-Newtonian fluids for different agitational speeds and coil flow rates. Adequate care was taken to maintain constant temperatures in both the storage tanks during a particular run period.

The following sets of the experiments were conducted:

1. In order to check the accuracy of the experimental set up, data were taken on Newtonian fluids for different flow rates and the agitational speeds.
2. In order to propose the laminar and turbulent heat transfer correlations for the flow of power law non-Newtonian fluids in coils, the data were taken with four aqueous CMC solutions at different flow rates.
3. Data were taken with different agitational speeds to propose correlations for jacket and coil outside heat transfer.
4. Pressure drops were measured during each coil flow rate studied, to present a correlation for momentum transfer to non-Newtonian fluids flowing through the coil.

RESULT AND DISCUSSION

The ratio of friction factor, f_c, for coil to that of straight tube, 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 (1).

Three non-Newtonian fluids: 0.5 (n= 0.937), 1(n= 0.851) and 2% (n= 0.793) CMC-A in water, representatives of pseudoplastic fluids obeying power law, and water, a Newtonian fluid were investigated. Most of the data obtained were in laminar region and very few data of non-Newtonian fluids could fall in turbulent region. Due to lack of turbulent flow data for non-Newtonian fluids in coil, only laminar region data are processed and correlated.

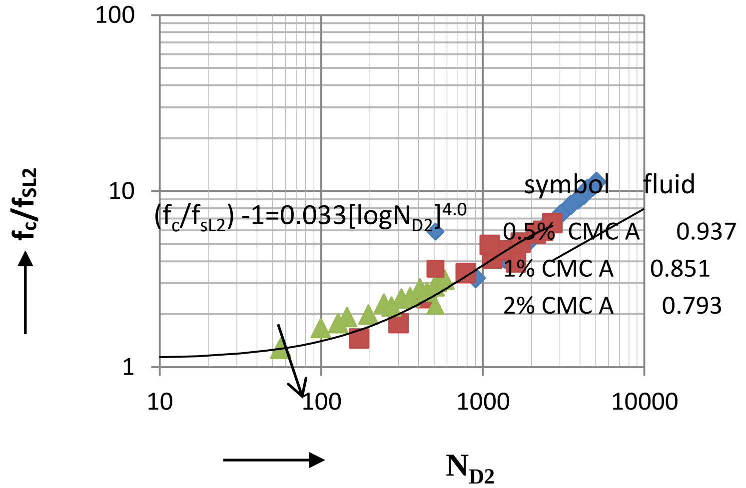

FIG 1: laminar flow friction factor in helical coils as a function of dean number for non-Newtonian fluids

In order to find the effect N_{D2} on the ratio of heat transfer in coil to heat transfer in straight pipe, the first attempt was to plot the ratio ($\frac{NNuic}{N_{Nuis}}$) against N_{D2} on a logarithmic scale as shown in figure (2).

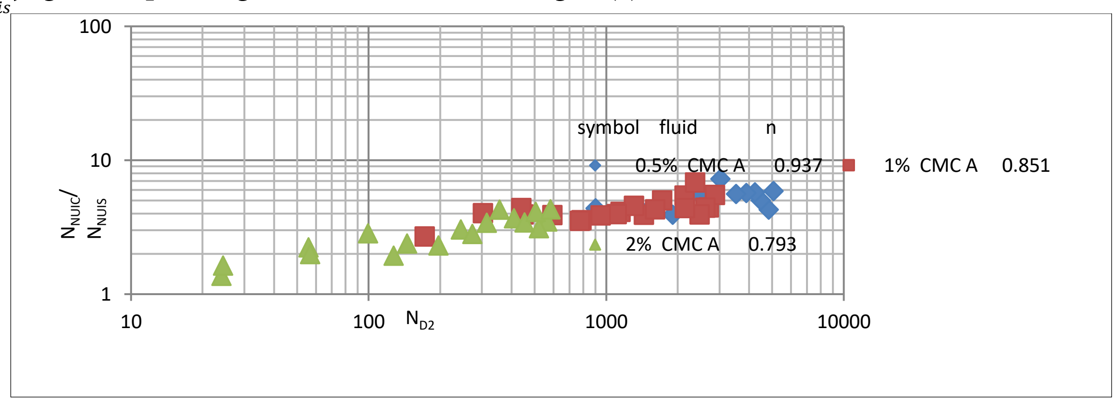

FIG 2: Variation of n_{nuic}/ n_{nuis} with n_{d2} for heat transfer to non-Newtonian fluids

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 $\left(\frac{NNuic}{N_{Nuis}}\right) - 1$ against N_{D2}. This plot is shown in figure (3).

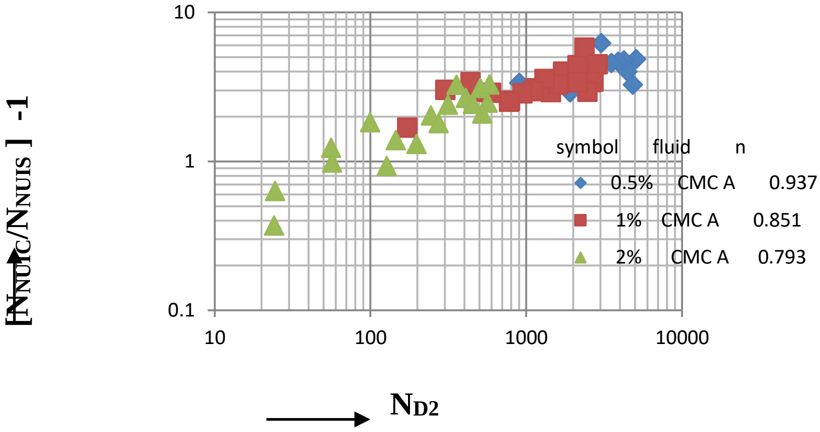

FIG 3: Variation of $[(n_{nuic}/n_{nuis}) -1]$ with n_{d2} for heat transfer to non-Newtonian fluids

No net conclusion could be drawn even from this figure except for some qualitative ones. The heat transfer coefficient in coil is always higher than in straight pipe as it is seen that $\frac{NNuic}{N_{Nuis}}$ is always greater than unity. The ratio h_{ic}/ h_{is} increases with Dean number, N_{D2} . 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 3.3. 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.

CONCLUSION

As a result of the present investigation on the heat transfer to agitated fluids flowing through immersed helical coils in jacketed vessels, the following conclusions are drawn:

1. For small temperature driving forces the non isothermal correlation is negligible.
2. The heat transfer data for agitated Newtonian and non-Newtonian fluids have been successfully correlated by using the viscosity of the fluid evaluated at the impeller tip assuming a cylinder of diameter equal to that of impeller rotating in an infinite fluid. Data of 1, 2 and 4% CMC-A, for three impeller diameters, have been correlated by the following equation:

For coil

$$N_{Nuoc} = 0.036\, N''^{2/3}_{Rea}\, N''^{1/3}_{Pra} \left(\frac{D_a}{D_c}\right)^{0.1}$$

(Standard deviation 15.03%)

Using the above concepts of Reynolds and Prandtl numbers it is also possible to correlate the available published data for other non-Newtonian fluids obtained with different impeller geometries.

3. The use of same Reynolds and Prandtl number expressions in correlation of jacket side and coil side heat transfer data provides a method of comparison between h_j and h_{oc}. The ratio h_j/h_{oc} is given by

$$\frac{h_j}{h_{oc}} = 8.39 \left({}^{D_c}/_{D_T}\right)^{0.1} \left({}^{D_t}/_{D_T}\right)$$

4. Under the identical flow conditions, laminar flow heat transfer rate in helical coils are higher than those in straight pipes. The ratio ${}^{N_{Nuic}}/_{N_{Nuis}}$ is found to be a function of N_{D_2} and N_{pr_2}. The data obtained have been correlated by the following equation.

$${}^{N_{Nuic}}/_{N_{Nuis}} = 1 + 0.0666\, N_{D_2}^{1/2}\, N_{pr_2}^{0.12}$$

(Standard deviation 15.52%)

$24 < N_{D_2} < 2000, 40 < N_{pr_2} < 225$ and $0.793 < n < 1$

NOMENCLATURE

b_3, b_4	constants in equation (1.2)
b_5, b_6	constants in equation (1.6)
D_a	agitator diameter, cm
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
D_T	diameter of the agitated vessel
f_c	friction factot in coil
f_{sL}	laminar flow friction factor in straight pipe
h_j	heat transfer coefficient for jacketed vessel wall to fluid, Kcal/hr m^2 ^0C
h_{oc}	coil outside heat transfer coefficient , kcal/hr m^2 ^0c
h_{ic}	coil inside heat transfer coefficient , kcal/hr m^2 ^0c
k	thermal conductivity
n	flow behavior index
n'	generalized flow behavior index
r	radial distance, cm
t	time, sec
Tci	inlet fluid temperature in the coil, ^0C
Tco	outlet fluid temperature in the jacket, ^0C
T_i	inlet temperature
T_s	surface temperatue
U	average velocity,cm
U_j	jacket overall heat transfer coefficient, Kcal/hr m^2
U_{OC}	coil overall heat transfer coefficient, Kcal/hrm^2 ^0c
u	local velocity in the x direction at r or y, cm/sec

Dimensionless groups

N_{Nu}	Nusselt number, h D/k
N_{Nuj}	Nusselt number, h_j D_T/k
N_{Nuoc}	Nusselt number, h_{oc} D_o/k
N_{Nuic}	Nusselt number, h_{ic} D_t/k
N_{NUIS}	Nusselt number, h_{is} D_t/k
N_{Re}	Reynolds number, $D_tU\rho/\mu$
N_{Re2}	Reynolds number defined by equation
N_{Re}"	Reynolds number defined by equation
N"$_{Rea}$	Reynolds number defined by equation
N_{Pra}	Prandtl number defined by equation
N_{pr}	Prandtl number
N''$_{pr}$	Prandtl number defined by equation
N_D	Dean number
N_{prd}"	prandtl number defined by equation

Subscript

j	jacket
c	coil
I	inner, inlet
m	mean value
o	outlet, outer
y	yield
SL	laminar flow in straight pipe

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