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# HEAT TRANSFER TO NON-NEWTONIAN FLUIDS IN COILED PIPES IN LAMINAR FLOW

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**Abstract**—Data on isothermal pressure drop and heat transfer to non-Newtonian fluids (both pseudoplastic and dilatant) in different coiled pipes (helical and spiral coils) at constant tube wall temperature conditions are presented. The present data have been compared with the corresponding correlations and theoretical results available for straight and coiled pipes in the literature, both for Newtonian and non-Newtonian fluid flow and heat transfer. A correlation is set up which can represent the present data as well as the data available in literature for Newtonian fluids.

## NOMENCLATURE

$C_p$, specific heat at constant pressure [cal/g degC];
$D_c$, diameter of the helical coil [cm]; suffix Avg denotes the average diameter of the spiral coil;
$D_i$, inner diameter of the tube [cm];
$D_i/D_c$, curvature ratio of the helical coil;
$(D_i/D_c)_{Avg}$, average curvature ratio of the spiral coil;
$f_c$, Fanning's friction factor for helical and spiral coils [dimensionless]: the suffix $S$ denotes friction factor for straight pipe;
$g_c$, acceleration due to gravity [cm/s$^2$];
$K$, consistency index [kg/m$^2$ . h$^n$];
$h_{am}$, heat-transfer coefficient evaluated at arithmetic mean temperature difference [kcal/hm$^2$ degC];
$k$, thermal conductivity [kcal/m h degC];
$L$, length of the coil [cm];
$n$, flow behaviour index [dimensionless];
$N_{Dean}$, Dean's number $= N_{Re}\sqrt{(D_i/D_c)}$;
$N_{Eck}$, $K^2u^2/(n+1)^2 D_i^{2+1/n}/C_p\,\Delta T$;
$N_{Gz}$, Graetz number $= WC_p/k_bL$;
$N_{Nu}$, Nusselt number $= h_{am}D_i/k_b$;
$N_{Pr}$, Prandtl number $= C_p\mu/k_b$; $'$ denotes modified Prandtl number for non-Newtonian fluids;
$N_{Re}$, Reynolds number $= D_iV\rho/\mu$; $'$ denotes modified Reynolds number $= (D^nV^{2-n}\rho/g_cK\,8^{n-1})$;
$\Delta P/L$, pressure drop per unit length of capillary [kg force/m$^3$];
$r$, radius in cylindrical polar coordinate system;
$T$, dimensionless temperature;
$t$, temperature [deg C];
$V$, velocity [cm/s]; suffix $m$ denotes the mean velocity in the capillary tube [m/h];
$u$, component of velocity in $x$ direction [cm/s];
$V$, component of velocity in $y$ direction [cm/s];
$w$, component of velocity in $z$ direction [cm/s];
$w_1$, component of velocity along the axis [cm/s];
$W$, weight rate of flow [g/s];
$\rho$, density [g/cm$^3$];
$\mu$, viscosity [cp];
$\tau_{xy}$, shear stress [kg/m$^2$].

The suffixes $b$, $w$, indicate bulk and wall temperatures respectively.

## INTRODUCTION

WHENEVER a Newtonian fluid flows in a helical or spiral coil, there exists a secondary flow in addition to the main flow caused by longitudinal pressure gradient. The straight streamlines in the flow of a fluid (Newtonian) in a straight cylindrical pipe are replaced by curved streamlines in coils and higher rates of momentum and heat transfer are obtained. These secondary flows are in the form of free vortex flow characterised by the equation $V_w \cdot R_c =$ constant, where $V_w$ is the velocity of whirl and $R_c$ is the radius of curvature. The velocity of whirl will be greater at the inside of the pipe bend than at the outside. Hence there exists a pressure gradient across the diameter of the curved pipes as shown in Fig. 1.

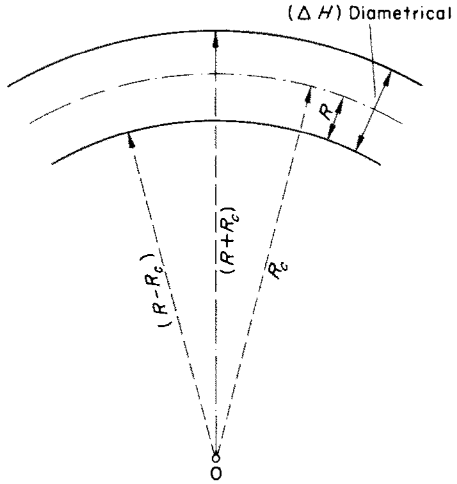


FIG. 1. Diametrical pressure drop.

This pressure gradient is referred to as the diametrical pressure drop. The intensity of secondary flows is dependent on the diametrical pressure drop. It is these secondary flows which will alter the mechanism of momentum and heat transfer in coiled pipes both in laminar and turbulent flow. The diametrical pressure drop for Newtonian fluids flowing in helical and spiral coils has been measured by Kubair and Kuloor [1, 2] in laminar and turbulent flow. It has been found that the diametrical pressure drop varies considerably along the length of the coiled pipes and is dependent on the geometry of the coil both in laminar and turbulent flow. The values of diametrical pressure drop increases as the flow rate is increased. The contribution of the diametrical pressure drop in increasing the rates of momentum transfer is lower in laminar flow and higher in turbulent flow. Accordingly the parabolic velocity profile encountered in straight cylindrical pipes, in laminar flow is replaced by a skew profile in coiled pipes. Since the centrifugal forces suppress the transverse motion, which are characteristics of turbulent flow, the onset of turbulence in coiled pipes is delayed and the transition occurs at a higher Reynolds number than in a straight pipe (even up to $N_{Re} = 10000$) depending on the geometry of the coil.

Following the work of Ito [3], Kubair and Kuloor [4] have suggested that the formula

$$(N_{Re})_{Cr} = 10^4 (1{\cdot}273)(D_1/D_c)^{0{\cdot}2} \qquad (1)$$

can be used to estimate the critical Reynolds number for helical and spiral coils, replacing the curvature ratio by the average curvature ratio for spiral coils.

The friction factors for coiled pipes are higher than in straight pipes, owing to the presence of secondary flows. The difference in the friction factors of coiled pipes and straight pipes is greater in laminar flow than in turbulent flow, owing to the higher contribution of centrifugal forces in laminar flow than in turbulent flow.

Similarly higher rates of heat transfer are obtained in laminar flow than in turbulent flow, since the thermal resistance is considerably reduced in laminar flow. The rate at which thermal resistance is reduced depends on the intensity of secondary flows, which in turn depend on $(L/D_i)$ ratio and $(D_i/D_c)$ ratio of the coils. Therefore, coiled pipe heat exchangers are preferable to straight pipe heat exchangers. Although multitubular straight pipe heat exchangers give higher area and higher rates of heat transfer, they are beset with a number of problems like a complicated common header, cleaning of tubes and economy. Hence coiled pipes are preferred and offer promising results as efficient heat exchangers.

The state of knowledge of pressure drop and heat transfer for Newtonian fluids in coils can be summarized as:

1. The onset of turbulence is in coiled pipes is delayed and can be estimated from the equation (1).
2. The rates of momentum and heat transfer in coiled pipes are higher than in straight pipes both in laminar and turbulent flow.
3. The friction factors for coiled pipes can be estimated from the correlation [5] for laminar flow,

$$f_c/f_s = [1 - \{1 - (11{\cdot}6/N_{Dean})^{0{\cdot}45}\}^{2{\cdot}2}]^{-1} \quad (2)$$

for $11{\cdot}6 < N_{Dean} < 2000$ $f_s = 16/N_{Re}$ and for turbulent flow friction factors can be estimated from the correlation [6]

$$f_c - f_s = 0{\cdot}120\,(D_i/D_c)^{0{\cdot}5} \quad (3)$$

where $f_s = 0{\cdot}079\ N_{Re}^{-\frac{1}{4}}$

4. The heat-transfer coefficients for laminar flow can be calculated using the correlation of Kubair and Kuloor [7]; viz

$$N_{Nu} = [1{\cdot}98 + 1{\cdot}8\,(D_i/D_c)]\,N_{Gz}^{0{\cdot}7} \quad (4)$$

for $80 < N_{Re} < 6000$, and for turbulent flow it can be estimated from the equation [8]

$$N_{Nu} = [1 + 3{\cdot}54\,(D_i/D_c)]\,N_{Re}^{0{\cdot}8}\,N_{Pr}^{0{\cdot}4} \quad (5)$$

for $N_{Re} > 10000$.

A number of non-Newtonian fluids (time independent and time dependent) like starch, clay suspensions, polyox, carboxymethyl cellulose and many polymer solutions in water are extensively used in industries. When coiled pipe heat exchangers are used to handle non-Newtonian fluids, the above discussions are not applicable and separate correlations are to be set up to provide a basis for design of coiled pipe heat exchangers for non-Newtonian fluids.

During the last twenty years there has been an increasing interest in the flow behaviour of non-Newtonian fluids both theoretical and experimental. Of various time independent non-Newtonian fluids, the power-law model characterised by the equation

$$\tau_{xy} = K(\mathrm{d}u/\mathrm{d}y)^n \qquad (n > 0) \quad (6)$$

where $K$ is the consistency index and $n$ is the flow behaviour index, has been recommended as the suitable simplification to the flow of many non-Newtonian fluids handled in industries.

Various analytical treatments have been given [9–14] for the flow of non-Newtonian fluids in curved pipes. However, because of the simplifications required, the predictions do not agree with the results of Rajasekaran *et al.* [15] who have measured the diametrical pressure drop for the flow of non-Newtonian fluids in helical coils and found that:

1. The curvature and the rheological properties of non-Newtonian fluids appreciably affect the axial and diametrical pressure drop values.
2. The diametrical pressure drop for non-Newtonian fluids indicate that the intensity of secondary flows is a function of the curvature ratio and the $(L/D_i)$ ratio.
3. Since the rates of momentum transfer depend upon the above variables, the rates of heat transfer should depend upon the same variables both in laminar and turbulent flows.

The theoretical predictions of Ratna [9] and Kanakaraju and Ratna [14] do not reveal the effect of curvature on momentum and heat transfer in coiled pipes, which is not evident by the preliminary work of Rajasekharan *et al.* [15]. Perhaps the order of correction of curvature introduced in equations of motion and energy may not have been sufficient to elucidate the effect of curvature on momentum and heat transfer.

Hence in order to understand the role of the secondary flows on pressure drop and heat transfer, this work is taken up, to present data on pressure drop and heat transfer to non-Newtonian fluids of power-law type, and establish correlations with the geometrical parameters of the coiled pipes.

## EXPERIMENTAL

The isothermal pressure drop data have been collected using different non-Newtonian fluids in different coiled pipes. The range of variables covered in present studies is given in Table 1.

*Table* 1.

| Helical or spiral coil | Curvature ratio | Range of $N_{Re'}$ | Liquids used |
|---|---|---|---|
| $H_I$ | 0·037 | 6–21260 | 1% CMC<br>0·5% CMC<br>1% sodium silicate |
| $H_{IV}$ | 0·097 | 3–25000 | 1% sodium silicate |
| $S_{II}$ | 0·021 | 10–25000 | 1% sodium silicate |

The pressure drop data are collected using the same apparatus as used by Kubair and Kuloor [7] in an earlier paper.

### *Heat transfer*

The arrangement of apparatus used for collecting heat transfer data is exactly the same as that reported by Kubair and Kuloor [7] for spiral and helical coils.

The range of variables covered in the different types of coiled pipes used in the present investigation are given in Table 2.

### *Method of operation*

For each flow rate the solutions which are prepared by dissolving a known amount of powders CMC, CPM and sodium silicate are used as the test fluids, which are metered by using a calibrated rotameter. The properties of the solutions of CMC, CPM and sodium silicate used in the present work are summarized in the Table 3. Steam from a boiler, is condensed on the helical and spiral coils. The non-condensable gases are removed by purging steam through vents. The calibrated thermocouples of copper–constantan of 32 gauge are used to measure the tube wall temperature. The thermo e.m.f. is noted on a Kaycee type potentiometer. When steady state is reached as denoted by the constant and reproducible readings of the thermometer at inlet and outlet points, thermo e.m.f. readings on potentiometer, the flow rate and inlet, outlet temperatures of liquid, steam temperature and the weight of condensate collected for a known time and flow rate of the liquid are noted. At the beginning and the end of every experiment, a sample of liquid is preserved which is used to determine the physical and rheological properties.

*Physical properties.* The physical properties of liquid such as density, specific heat are determined at the bulk temperature of the liquid. A pyknometer immersed in an ultra-

*Table* 2.

| Helical or spiral coils | Curvature ratio $D_i/D_c$ or $D_i/D_c$(avg.) | $D_i$ (cm) | No. of turns | Solutions used | Range of $N_{Gz}$ |
|---|---|---|---|---|---|
| $H_I$ | 0·037 | 0·6549 | 7 | 1% CMC<br>0·5% CMC | 20–90<br>40–90 |
| $H_{II}$ | 0·056 | 0·6407 | 12 | 1% CMC | 36–130 |
| $H_{III}$ | 0·074 | 0·954 | 10 | 1% CMC<br>0·5% CMC | 22–113<br>63–120 |
| $H_{IV}$ | 0·097 | 1·27 | 10 | 1% CMC<br>0·5% CMC | 48–135<br>48–139 |
| $S_{II}$ | 0·222 | 0·642 | 4·5 | 1% CMC<br>1·25% CMC | 30–77<br>33–91 |
| $S_{III}$ | 0·031 | 0·9468 | 5·0 | 1% CMC<br>0·5% CMC<br>1% sodium silicate<br>0·5% CPM | 60–140<br>50–96<br>75–103 |

thermostat is used to determine the density. The specific heat and thermal conductivity of the test liquid is assumed to be same as that of water.

*Rheological properties.* The range of wall shear stress used during heat transfer runs are calculated. An ultra thermostat is used to maintain the bulk temperature of the liquid. Water from the thermostat at the bulk temperature of the liquid is allowed to circulate through the jacket surrounding the capillary of the capillary tube viscometer. The different capillaries of different diameters are used to cover the different range of shear stress. The readings of pressure drop, weight of liquid collected for a known amount of time are noted under steady conditions of state. The plot $D\Delta P/4L$ vs. 8 $V_m/D$ are drawn on logarithmic coordinates for the test liquid. The slopes (flow behaviour index $n$) and the intercepts (consistency index $K$) are calculated using least square method. The values of $n$ and $K$ are similarly determined for different test liquids, for different helical and spiral coils (at different wall shear stresses) and at different bulk temperatures. The typical rheogram is presented in Fig. 6 and the properties in Table 3.

*Table 3. Power law parameters*

| | $C$ (wt.%) | $n$ | $\log K$ (kg/m$^2$ × s$^n$) |
|---|---|---|---|
| CMC in water 73°C | 1% | 0·73 | 4·3558 |
| | 0·5% | 0·63 | 4·6088 |
| CPM in water 75°C | 0·25% | 0·47 | 4·3103 |
| | 0·5% | 0·6886 | 4·0183 |
| Sodium silicate 75°C | 1·5% | 2·16 | 11·9735 |

## RESULTS AND DISCUSSION

The results of pressure drop data for Newtonian fluids in the same apparatus has been published by Kubair and Kuloor [7], and are found to be in good agreement with the various correlations reported for Newtonian fluids, as shown in Fig. 2.

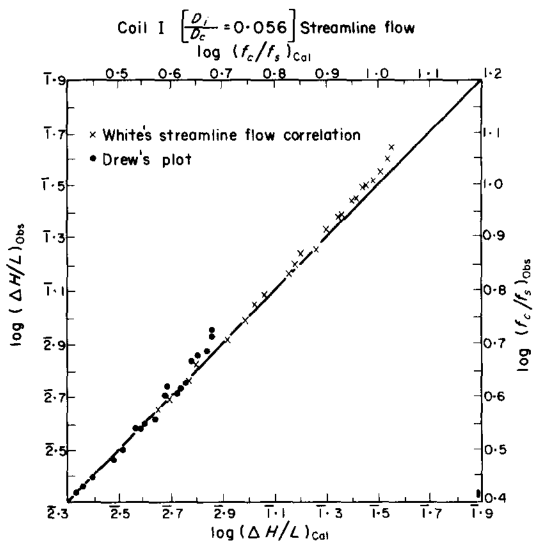


FIG. 2. Comparison of the present data with the existing correlations.

The typical results of pressure drop data for non-Newtonian fluids flowing through helical and spirals have been presented in Figs. 3–5 (spirals).

It can be found that as modified Reynolds number increases, the friction factors decrease and are much higher than the friction factors calculated from standard correlations available for laminar [16] and turbulent [17] flow of non-Newtonian fluids in straight pipes. But as turbulent flow ls reached, the difference in friction factors for straight and coiled pipes is not greater as observed in the figures. The difference is appreciable in the case of laminar flow owing to the higher contribution of centrifugal forces towards total pressure drop values whereas in the case of turbulent flow, the suppressed motion of secondary flows by dominating inertia force is responsible for smaller difference in friction factors between curved and straight pipes. Moreover, the onset of turbulence is delayed in all the coils owing to the fact that transverse motions characteristic of turbulent flow are suppressed by centrifugal forces.

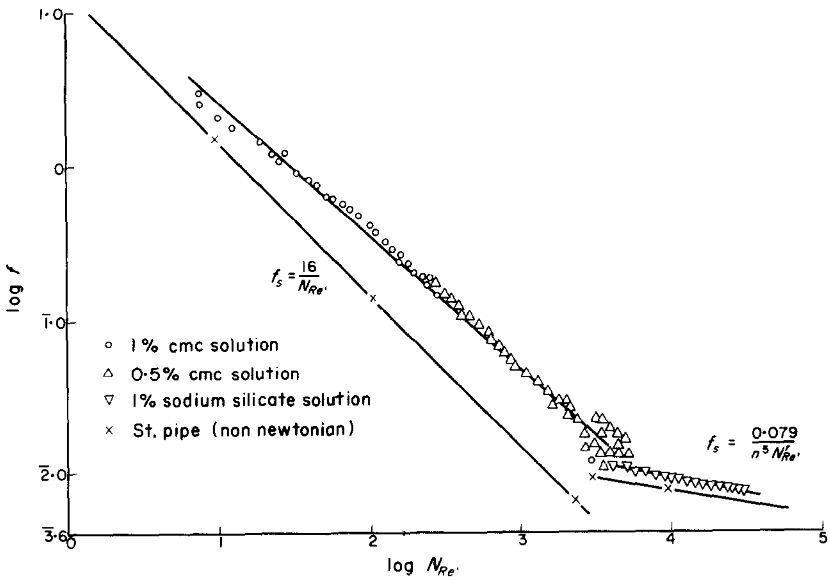


FIG. 3. Isothermal pressure drop data Helical coil I ($D_i/D_c$ = 0·037).

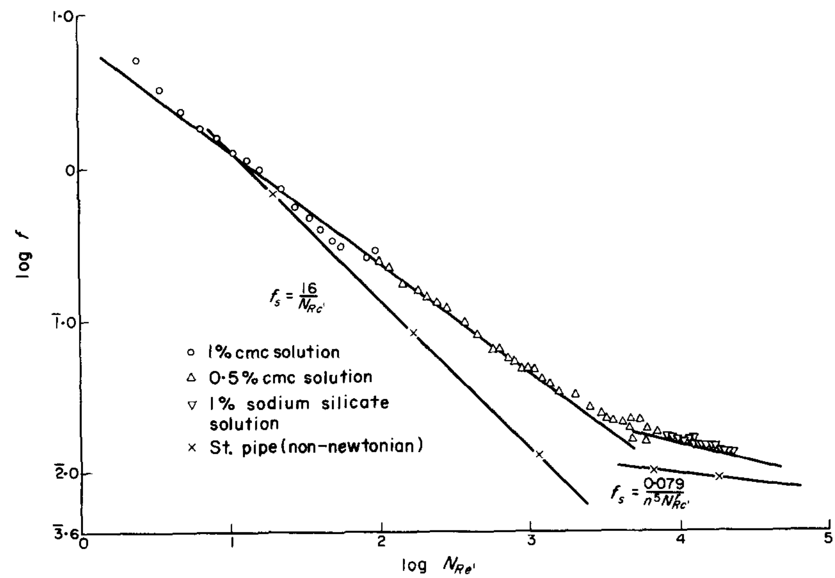


FIG. 4. Isothermal pressure drop data coil IV ($D_i/D_c$ = 0·097).

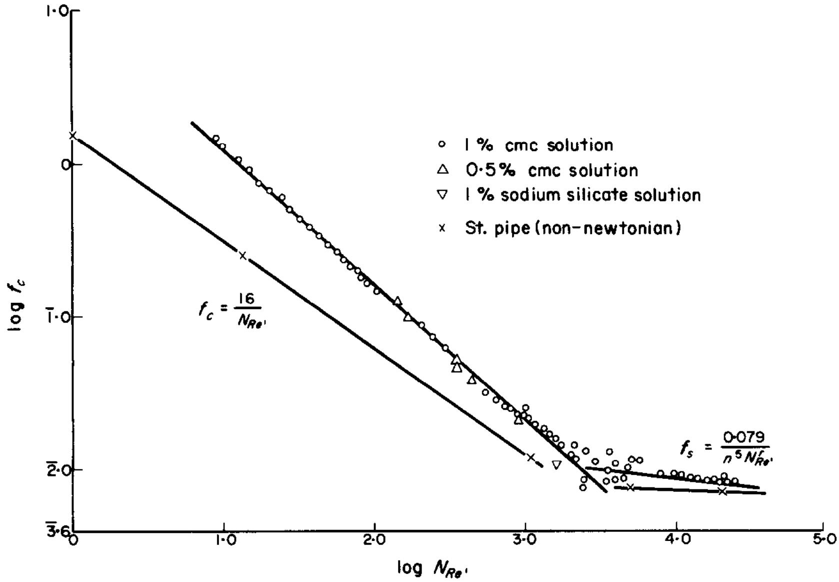


FIG. 5. Isothermal axial pressure drop data spiral II.

### *Heat transfer*

The results of heat transfer are presented in Figs. 7–12 as plots of log $N_{Nu}$ vs. log $N_{Gz}$. It is found that the heat transfer coefficients are very much higher than for Newtonian and non-Newtonian fluids in straight pipes, and Newtonian fluids in coiled pipes.

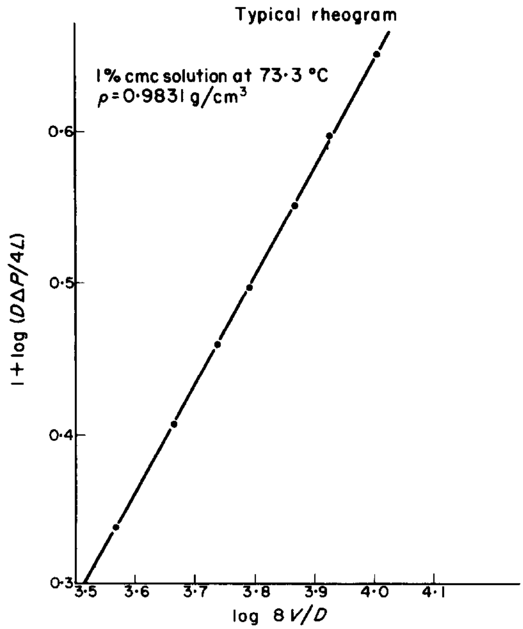


FIG. 6. Plot of 1 + log ($D\Delta P/4L$) vs. log ($8\ Vm/D$).

The heat-transfer coefficients for Newtonian and non-Newtonian fluids for straight pipes are taken from Pigford's correlation [18] viz

$$\frac{h_{am}D_i}{k_b} = 1{\cdot}75[(3n + 1)/4n]^{\frac{1}{3}}(N_{Gz})^{\frac{1}{3}} \qquad (7)$$

which has been experimentally verified by Metzner *et al.*, in the range $0{\cdot}2 < n < 0{\cdot}7$. For $n > 1$, no data seem to exist for heat transfer to non-Newtonian fluids. Since the consistency index is found to vary, Metzner *et al.* [19] have recommended the correction $(K_b/K_w)^{0{\cdot}14}$ corresponding to Sieder Tate's correction for Newtonian fluids, Kubair and Kuloor [7] found that such corrections are not necessary since the contribution of this term did not appreciably influence the correlation.

Since in the present studies, constant tube wall temperature conditions are maintained, Pigford's correlation [18] has been used for comparison.

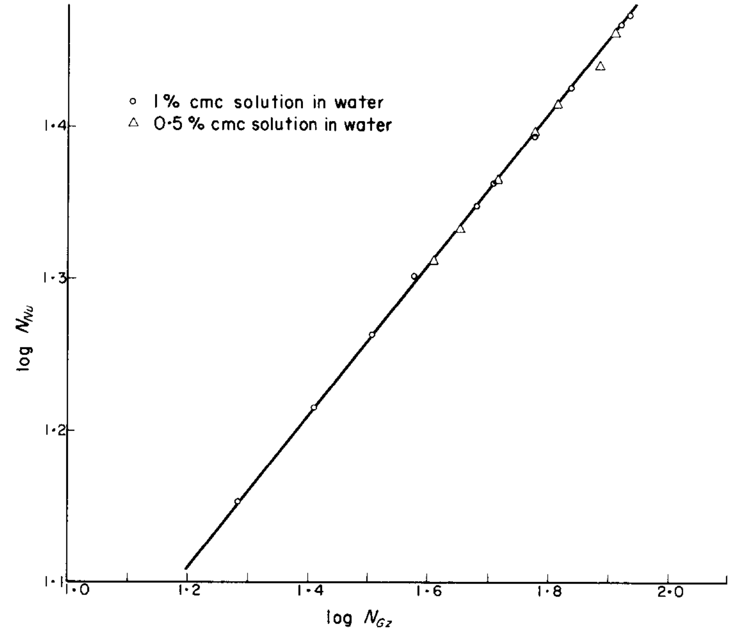


FIG. 7. Plot of log $N_{Gz}$ vs. log $N_{Nu}$ for helical coil I.

Rajasekharan *et al.* [12] found that intensity of secondary circulation for non-Newtonian fluids depends on $L/D_i$ ratio and $D_i/D_c$ ratio. Therefore the rate of reduction of thermal resistance i.e. heat-transfer coefficient must depend on these variables. Therefore, the heat transfer coefficients depend on:

$$N_{Nu} = F(N_{Re'}, N_{Pr'}, D_i/D_c, L/D_i\, n)$$
$$= F(N_{Gz'}, n).$$

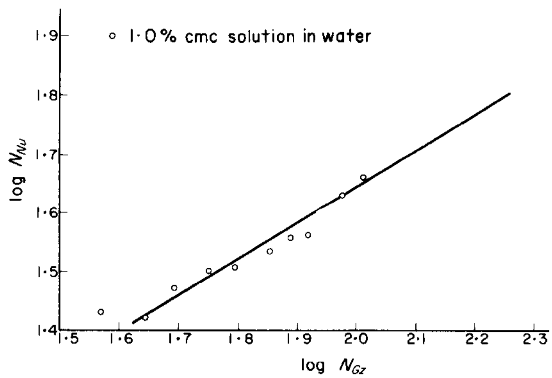


FIG. 8. Plot of log $N_{Gz}$ vs. log $N_{Nu}$ for helical coil II.

Kanakaraju and Ratna [14] have found theoretically

$$N_{Nu} = F(N_{Re}, N_{Pr}, N_{Eck}, n)$$

and that Nusselt's number increases with the decrease in the value of $n$. No appreciable effect of curvature has been reported by them.

Hence based on the above considerations it is necessary $N_{Gz'}$ is the correct characteristic dimensionless group to correlate the heat transfer coefficients for non-Newtonian fluids, similar to the correlation of Kubar and Kuloor [7] for Newtonian fluids viz.

$$N_{Nu} = [1{\cdot}98 + 1{\cdot}8\,(D_i/D_c)]\, N_{Gz}^{0{\cdot}7} \qquad (8)$$

for $80 < N_{Re} < 6000$ and $20 < N_{Pr} < 100$.

In the Table 4, the results in the present studies are compared with heat-transfer data, for Newtonian and non-Newtonian fluids in straight pipes, and Newtonian fluids in coiled pipes.

It can be found from Table 4, that curvature has a definite increasing effect on heat-transfer

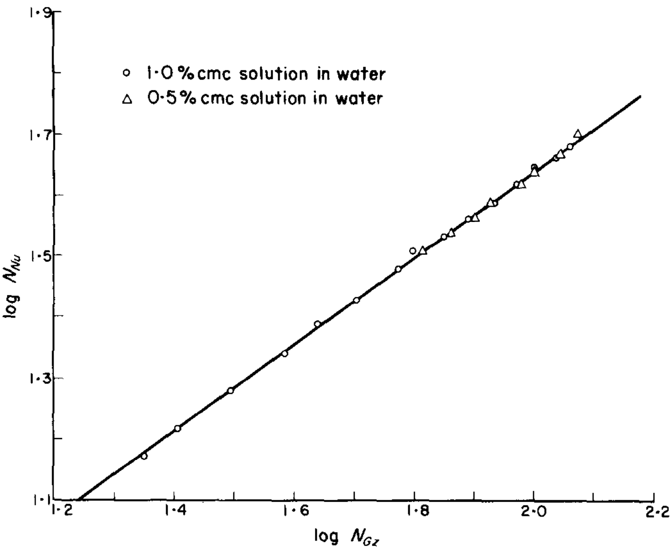


FIG. 9. Plot of log $N_{Gz}$ vs. log $N_{Nu}$ for helical coil III.

coefficients which has not been reported by earlier workers. But the heat-transfer coefficients are increasing as $n$ decreases which is in agreement with the conclusion of Kanakaraju and Ratna [14], based on their theoretical analysis. But the conclusion reported by Kanakaraju and Ratna [14] holds good even for straight pipes [22] and also for boundary-layer flow [23]. But the effect of curvature on heat transfer is not revealed from their analysis since in their analysis they did not apply higher order corrections of curvature. Since the calculations and theoretical analysis are too tedious for higher order corrections, it is found necessary to propose an empirical correlation.

From Table 4, it can be found that the increase in the rates of heat transfer is about 10 times than in straight pipes. For curved pipes the prediction of Ratna for dilatent fluids does not hold good since higher heat-transfer coefficients than pseudoplastic fluids are obtained. But only limited data of heat transfer to sodium silicate indicates that further work is necessary in this direction.

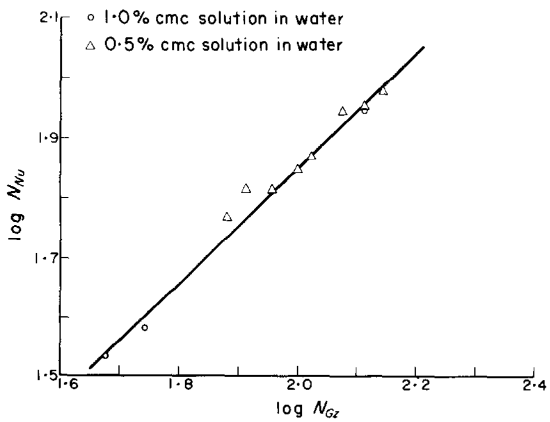


FIG. 10. Plot of log $N_{Gz}$ vs. log $N_{Nu}$ for helical coil IV.

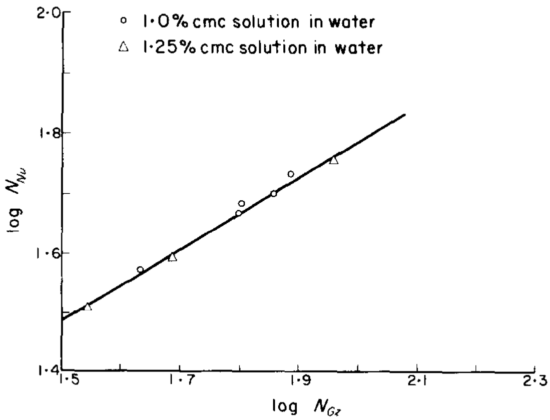


FIG. 11. Plot of log $N_{Gz}$ vs. log $N_{Nu}$ for spiral II.

*Table 4. Values of $(hD/k)$ at $N_{Gz} = 80$*

| Coil | $D_i/D_c$ | $n = 0.63$ 0.5% CMC solution | $n = 0.73$ 1% CMC solution | $n = 1.0$ glycerol solution | $n = 2.2$ |
|---|---|---|---|---|---|
| $H_{I}$ | 0.057 | 28.0 | 28.0 | 31.0 | |
| $H_{II}$ | 0.056 | | 35.0 | 32.0 | |
| $H_{III}$ | 0.074 | 38.0 | 37.2 | 33.0 | |
| $H_{IV}$ | 0.097 | 84.39 | 54.0 | 39.0 | |
| $S_{II}$ | 0.022 | | 56.0 | 31.0 | 72.7 |
| $S_{III}$ | 0.031 | 42.0 | | 33.0 | |
| St. pipe [18] | | 7.711 | 7.589 | 7.367 | 3.258 |

In the Table 5, the values of $N_{Nu}$ at lower and higher ranges of $N_{G_z}$ follow the conclusion reported by Kanakaraju and Ratna [14] namely that heat-transfer coefficients increase with decrease in $n$. For dilatant fluids in coiled pipes, highest transfer coefficients are obtained, than in straight pipes. But in the absence of any theoretical and experimental verifications, generalization is not possible.

*Table 5. Values of $(hD/k)$*

| | $N_{Gz}$ | $n = 0.73$ 1% CMC | $n = 2.2$ 1% sodium silicate solution | $n = 1.0$ |
|---|---|---|---|---|
| $N_{I}$ | 19.0 | 14.23 | — | 8.9 |
| $S_{II}$ | 132.0 | — | 106.2 | 31.62 |

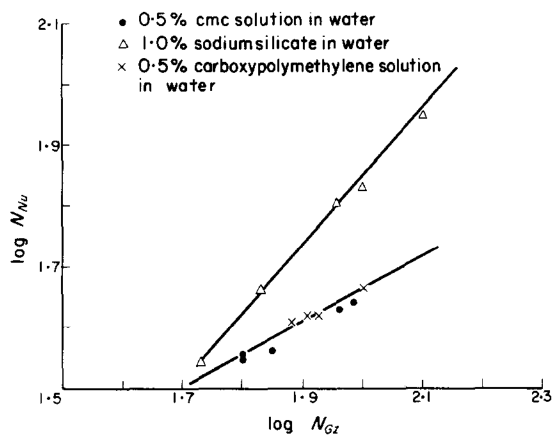


FIG. 12. Plot of log $N_{Gz}$ vs. log $N_{Nu}$ for spiral III.

Based on the same arguments as given above, the Nusselts number for coiled pipes, whenever a non-Newtonian fluid flows, is a function of modified Graetz number and the flow behaviour index. Since the variation of the consistency index does not appreciably influence the heat-transfer coefficients, because of the secondary flows, it is not included in the correlation applicable for coiled pipes. The correlation of Kubair and Kuloor [7], equation (7), applicable for Newtonian fluids is used as the basis for accounting the data on heat transfer to non-Newtonian fluids in coiled pipes, using the correction factor $[(3n + 1)/4n]$. The exponent of this correction factor for straight pipes is given as $\frac{1}{3}$ by Pigford [18] and it is same as exponent of $N_{Gz}$. The same arguments apply to non-Newtonian fluids in coiled pipes and $[(3n + 1)/4n]^{0.7n}$ if included in Kubair and Kuloor's correlation [7], can very well represent the present data on heat transfer to non-Newtonian fluids in all the coiled pipes. The final correlation can be written as:

$$N_{Nu} = [1.98 + 1.8\,(D_i/D_c)]\left[\frac{3n + 1}{4n}\right]^{0.7n} N_{Gz}^{0.7}. \tag{9}$$

For $800 < N_{Re'} < 9000$, $0.4 < n < 2.0$ and $10 < N_{Gz} < 1000$, $10 < N_{Pr'} < 100$.

The present correlation predicts the data of Kuloor [7] within $\pm 10$ per cent and average deviation less than 3 per cent, for Newtonian fluids. The agreement between the observed and calculated results is shown in Fig. 13.

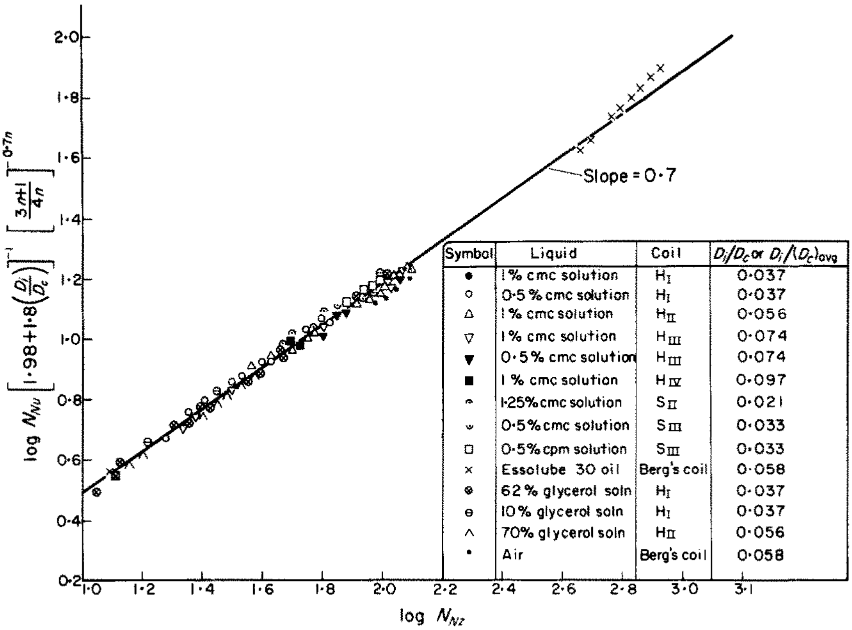


FIG. 13. Plot of $\log N_{Nu}[1{\cdot}98 + 1{\cdot}8(D_i/D_c)]^{-1}\,[(3n + 1)/4n]^{-0{\cdot}7n}$ vs. $\log N_{Gz}$.

If $n = 1$ the present correlation reduces to the correlation published earlier [7]. The present correlation can predict the data of Berg and Bonilla [20] satisfactorily at higher Graetz numbers and that of Seban and McLaughlin [21] at lower Graetz numbers. The data on heat transfer to Newtonian fluids are also included in the Fig. 13 with a view to establishing the validity of the correlation for Newtonian fluids.

## CONCLUSIONS

1. The friction factors and heat-transfer coefficients for non-Newtonian fluids in coiled pipes are higher than those for straight pipes and are affected by the curvature ratio and flow behaviour index.

2. Data on heat transfer to non-Newtonian fluids in different coiled pipes can be satisfactorily represented by equation (9).

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## TRANSPORT DE CHALEUR VERS DES FLUIDES NON-NEWTONIENS EN ÉCOULEMENT LAMINAIRE DANS DES TUYAUX ENROULÉS

**Résumé**—On présente les résultats sur la perte de charge isotherme et le transport de chaleur vers des fluides non-Newtoniens (pseudoplastiques ou dilatants) dans différents tuyaux enroulés (bobines en hélice ou en spirale) avec des conditions de température pariétale du tube constante. Les résultats actuels ont été comparés avec les corrélations correspondantes et les résultats théoriques sur les tuyaux droits et enroulés disponibles dans la littérature, pour l'écoulement et le transport de chaleur dans des fluides Newtoniens ou non-Newtoniens. On établit une corrélation qui peut représenter les résultats actuels aussi bien que les résultats disponibles dans la littérature pour les fluides Newtoniens.

## WÄRMEÜBERGANG AN NICHT-NEWTON'SCHEN FLÜSSIGKEITEN IN LAMINARER STRÖMUNG IN GEWENDELTEN ROHREN

**Zusammenfassung**—Es werden Daten aufgeführt für den isothermen Druckabfall und den Wärmeübergang an Nicht-Newton'sche Flüssigkeiten (sowohl pseudoplastische als auch elastische) in verschiedenen gewundenen Rohren (wendel- und spiralförmig) bei konstanter Rohrwandtemperatur. Die angegebenen Werte wurden verglichen mit entsprechenden Beziehungen und theoretischen Ergebnissen, die in der Literatur für Strömung und Wärmeübergang Newton'scher und Nicht-Newton'scher Fluide in geraden und gewundenen Rohren zur Verfügung standen. Es wird eine Beziehung aufgestellt, die für die vorliegenden Werte ebenso wie für die in der Literatur angegebenen Werte für Newton'sche Fluide gültig ist.

## ТЕПЛООБМЕН ПРИ ЛАМИНАРНОМ ТЕЧЕНИИ НЕНЬЮТОНОВСКИХ ЖИДКОСТЕЙ В ЗМЕЕВИКАХ

**Аннотация**—Приводятся данные по перепаду давления и теплообмену неньютоновских жидкостей (псевдопластичных и дилатантных) в различных змеевиках при постоянной температуре стенок трубы. Приведенные данные сравнивались с соответствующими корреляциями и теоретическими литературными данными для прямых труб и змеевиков, как для ньютоновских, так и неньютоновских жидкостей. Найдено соотношение, которое может обобщить полученные данные и данные, имеющиеся в литературе, на случай ньютоновских жидкостей.