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Study of the Effect of Non-Newtonian Behavior of Power-Law Non-Newtonian Fluids Flowing across Tube Banks

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ABSTRACT: Aim of the present work is to study the heat transfer rates based on the theoretical analysis of hydrodynamic boundary layer and thermal boundary layer of Power-law fluids and to compare these values from the results obtained experimentally. This study indicates that the thickness of the hydrodynamic boundary layer is a function of the flow behavior index (n) and angular position (θ).
Assuming flow to be two dimensional and consistency index (K) and flow behavior index (n) remaining constant over the complete range of wall shear stress, integration of momentum integral equation gives:

$$\frac{\delta}{d_o} = \left[\frac{\varphi(n)\varphi(\theta)}{Re_o}\right]^{1/(n+1)}$$

Where,

$$\phi(n) = 2^{(n-3)}\,(n+1)^n\,(n+2)(2n+3)$$

$$\phi(\theta) = \frac{1}{Sin^{(4n+5)}\theta}\int_0^\theta Sin^{(5n+3)}\theta\, d\theta$$

$$Re_o = \frac{d_o^n\, u_o^{2-n}\,\rho}{K}$$

KEYWORDS: Angular position (θ), boundary layer, convergence, creeping flow, Nusselt number, Prandtl number, pseudo plasticity, separation

I. INTRODUCTION

Present work has been carried out to study the influence of non – Newtonian behavior of different Power- law fluids on heat transfer in flow across a single tube and bundle of tubes. Experimental data were obtained for the effects of fluid rheology, tube diameter and neighbouring tubes on the heat transfer characteristics. The test fluids included water (Newtonian), 1.0 % and 1.5% aqueous solutions of CMC (pseudo plastic) and 0.5% aqueous PVA solution (dilatant).

II. THEORETICAL

Theoretical analysis of the boundary layer, experimental studies to viscous flow and heat transfer from vertical tubes immersed in Power-law fluids clearly indicate that the thickness of the hydrodynamic boundary layer is a function of the flow behavior index (n) and angular position (θ).
For two dimensional flow of power – law fluids, the integration of momentum integral equation gives:

$$\frac{\delta}{d_o} = \left[\frac{\varphi(n)\varphi(\theta)}{Re_o}\right]^{1/(n+1)} \qquad (1)$$

It indicates that for laminar flow, at a given Reynolds number, the boundary layer thickness is dependent only on the flow behavior index (n) and angular position (θ).
An expression for practicable Reynolds number which is independent of the local conditions for Power-law fluids has been derived as:

$$Re' = (Re_0)^{2/(n+1)} = \left[\frac{d_0^n u_0^{2-n}\rho}{K}\right]^{2/(n+1)} \qquad (2)$$

It can be rewritten as:

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$Re_o = [\mathrm{Re}']^{(n+1)/2}$ (3)

By substituting the value of Re_o in equation (1), following expression is obtained for boundary layer thickness:

$$\left(\frac{\delta}{d_0}\right) = \left[\frac{\varphi(n).\varphi(\theta)}{\mathrm{Re}_0}\right]^{\frac{1}{n+1}} = \left[\frac{\varphi(n).\varphi(\theta)}{[Re']^{(n+1)/2}}\right]^{\frac{1}{n+1}} \quad (4)$$

where, $\varphi(n) = 2^{(n-3)}(n+1)^n(n+2)(2n+3)$

$$\varphi(\theta) = \frac{1}{\sin^{(4n+5)}\theta}\int_0^\theta \sin^{(5n+3)}\theta \, d\theta$$

$$\mathrm{Re}_0 = \frac{d_o{}^n u_0^{2-n}\rho}{K}$$

and

$$\delta^{n+1} = (n+1)^n(n+2)\,(2n+3)\,\frac{K}{\rho}\left(\frac{R}{(2u_0)^{2-n}}\right)\frac{1}{\sin^{4n+5}\theta}\int_0^\theta \sin^{5n+3}\theta \, d\theta \quad (5)$$

Or $\left(\frac{\delta}{d_0}\right)^{n+1} = \dfrac{2^{n-3}(n+1)^n(n+2)(2n+3)}{\frac{d_o{}^n u_0^{2-n}\rho}{K}}\,\dfrac{1}{\sin^{(4n+5)}\theta}\int_0^\theta \sin^{(5n+3)}\theta \, d\theta$

The values of the right hand side in the above expression (5) for n = 0.2, 0.4, 0.6,….., 2.0 and at various angular positions(θ) have been computed and are listed in table A.1 of Appendix. Typical plots, depicting the variation of boundary layer thickness with flow behavior index (n), are shown in Figs.1 and 2, where boundary layer thickness is represented by radial distances between the surface of the cylinder and the edge of boundary. From Fig.1, it is observed that boundary layer thickness decreases with the decreasing value of flow behavior index. It converges in the region near the point of stagnation. For pseudo plastic fluids, the boundary layer thickness is seen to reduce in the region near the point of stagnation and then it increases with an increase in angular position. However, for dilatant fluids, a reverse trend in the variation of the boundary layer thickness near the stagnation point is observed. The boundary layer thickness increases continuously for all values of angular position (θ).

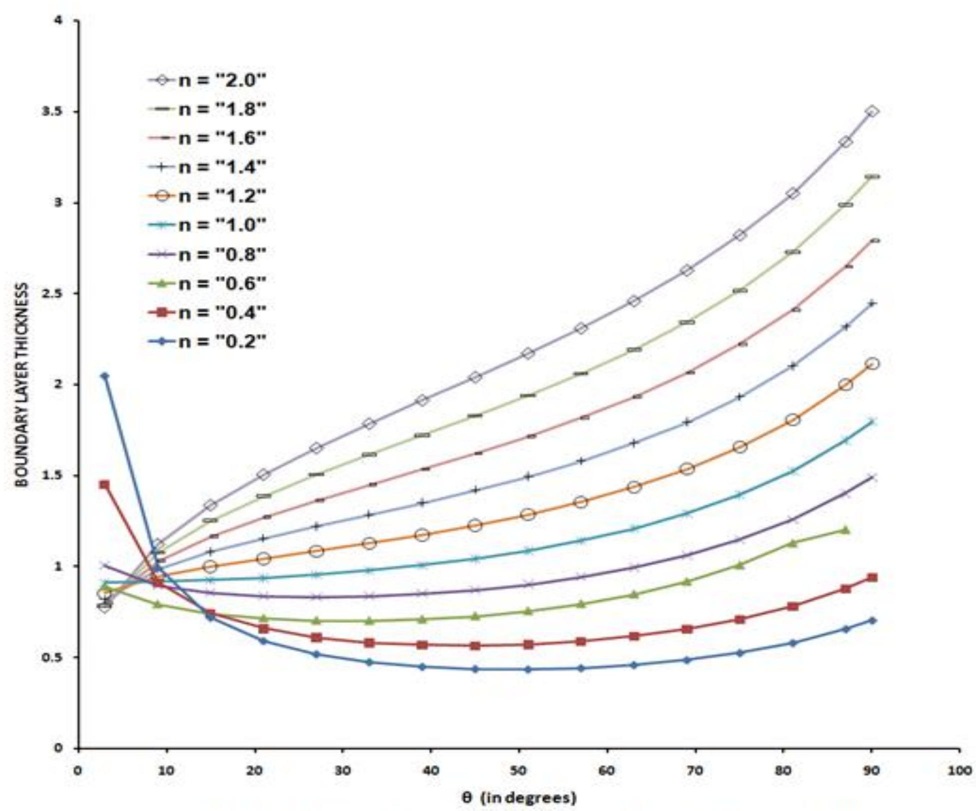

FIG. 1 VARIATION OF BOUNDARY LAYER THICKNESS WITH THE ANGULAR POSITION THETA(θ) FROM THE FRONT STAGNATION POINT

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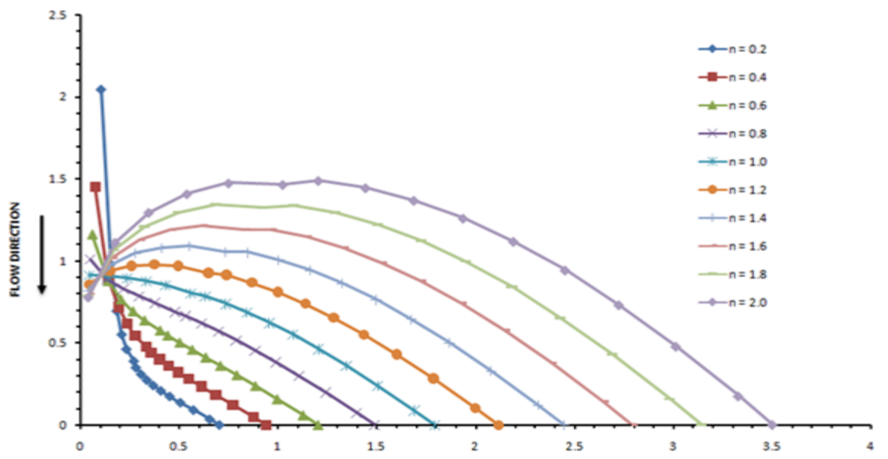

FIG. 2 DEVELOPMENT OF BOUNDARY LAYER ON A CIRCULAR CYLINDER: EFFECT OF NON-NEWTONIAN BEHAVIOUR

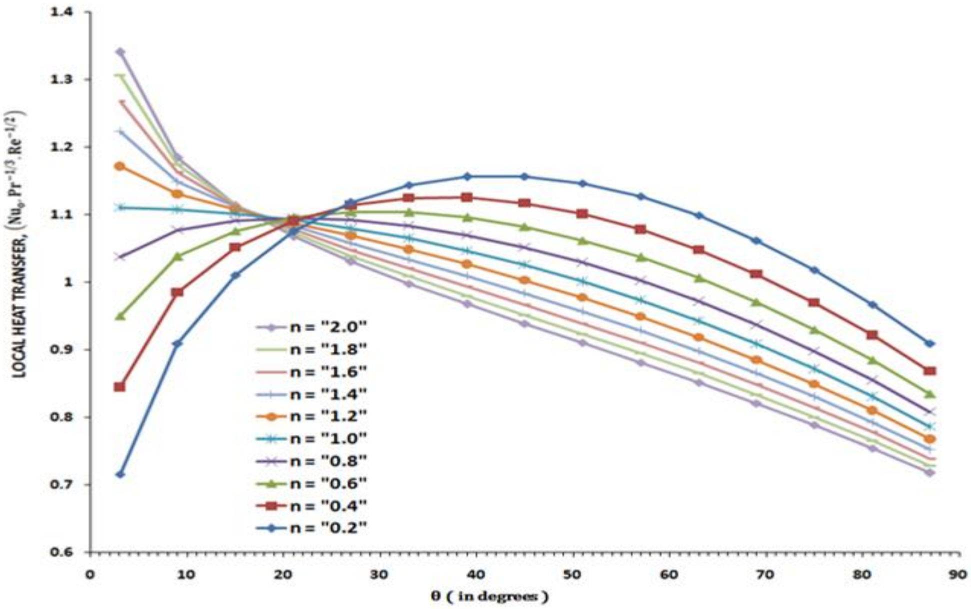

FIG. 3 CONVECTIVE HEAT TRANSFER FROM A VERTICAL CYLINDER: VARIATION OF LOCAL HEAT TRANSFER WITH ANGULAR POSITION THETA (θ)

A plot showing the variation in the boundary layer thickness with angular position (θ) from the point of stagnation is given in Fig.2. This figure shows the formation of boundary layer on the first quadrant of the circular cylinder, i.e. upto an angle of $\pi/2$ for the creeping flow without boundary layer separation. From the plot it is evident that at stagnation point, the boundary layer thickness does not reduce to zero. This aspect is also supplemented by the fact that a contradiction would yield an infinite heat transfer rate at the point of stagnation. Although, there is no distinct point of convergence of the boundary layer thickness but difference is minimum for angular position between 5^0 to 10^0. The convergence, however, should have been at one point only and the observed range can be attributed to the limitation of the numerical integration procedure used for computing the values.

For a dilatant fluid, the boundary layer thickness increases with angular position (θ), whereas, for pseudo plastic fluid it first decreases with θ upto convergence range and then increases. However, for both pseudo plastic as well as dilatant fluids, the extent of rise of boundary layer thickness is seen to be dependent on flow behavior index (n). As the value of

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n decreases, the angular position at which the boundary layer starts rising also increases. For example, for a Newtonian fluid, the boundary layer starts rising approximately at 20^0 while for a pseudo plastic fluid the rise starts approximately at 50^0. For the dilatant fluid the rise in the boundary layer thickness is found to start from the very beginning.

2.2 HEAT TRANSFER FROM A CIRCULAR CYLINDER IN CROSS FLOW:

Thus it may be concluded that the increase in dilatancy (i.e. increasing n beyond 1.0) and decrease in pseudo plasticity (i.e. increasing n towards the value 1.0) cause the formation of a thinner boundary layer over the surface. For fluids having higher Prandtl number, it further reduces the thermal boundary thickness. Expressions derived for local heat transfer rates are given as:

$$\frac{Nu_\theta}{Re'^{1/2}\,Pr'^{1/3}} = \frac{0.855\,(n+1)^{1/3}}{[\varphi(n)]^{1/3(n+1)}} \cdot \frac{\mathrm{Sin}^{\frac{5n+6}{3(n+1)}}\theta}{\theta^{1/3}} \left[\int_0^\theta \mathrm{Sin}^{(5n+3)}\theta\, d\theta\right]^{-1/3(n+1)} \tag{6}$$

By integrating equation (6), the average heat transfer rate is given as:

$$\frac{\mathrm{Nu_{av}}}{\mathrm{Re'^{1/2}\,Pr'^{1/3}}} = \frac{0.855\,(n+1)^{1/3}}{[\varphi(n)]^{1/3(n+1)}} \mathrm{x} \frac{1}{\theta}\int_0^\theta \mathrm{Sin}^{\frac{5n+6}{3(n+1)}}\theta \left[\int_0^\theta \mathrm{Sin}^{(5n+3)}\theta\, d\theta\right]^{-1/3(n+1)} d\theta \tag{7}$$

As both Prandtl number and Reynolds number are dependent on the effective viscosity, it can be seen that the Prandtl number is a function of Reynolds number and may be defined as:

$$\mathrm{Re'} = \frac{d_o\, u_o\, \rho}{\mu_{eff}}$$

$$\mu_{eff} = \frac{d_o\, u_o\, \rho}{Re'} = \frac{d_o\, u_o\, \rho}{{Re_o}^{2/(n+1)}}$$

Therefore, Prandtl number, $\mathrm{Pr} = \frac{\mu_{eff}\, C_p}{k} = \frac{d_o\, u_o\, \rho\, C_p}{k(Re_o)^{2/(n+1)}}$ (8)

Fig.(3) shows the variation of local heat transfer with angular position (θ) from the front stagnation point. The values of flow behavior index (n) have been taken equal to 0.2, 0.4, 0.6,…,2.0 For pseudo plastic fluids (n<1),the heat transfer is found to increase rapidly in the regions close to the stagnation point. This may be attributed to a decrease in boundary layer thickness. A reverse characteristic is seen in the case of dilatants fluids, the heat transfer rate decreases in a small region around the stagnation point (due to increase in boundary layer thickness). This trend continues up to a certain point, after which, increase in heat transfer is evident. Further, increase in dilatancy (increasing value of n) decreases heat transfer rates.

For dilatant fluids, the local heat transfer rate decreases right from the beginning to the end. Heat transfer rate for Newtonian fluid (n=1.0) remains almost constant up to about 20^o and then decreases. It is also evident from this figure that with decrease in the flow behavior index (n), the local heat transfer increases over most of the region ($\theta > 20^o$). In order to have a clear picture of the effect of flow behavior index (n) on the local heat transfer rates, a few calculated values are summarized in the table A-2 of Appendix.

III. EXPERIMENTAL

The experimental programme has been aimed to develop a tool for studying heat transfer for fluids flowing across cylindrical tube(s). The effect of tube diameter (or p/d) on heat transport phenomena for both Newtonian and non-Newtonian fluids has been studied. In the present work a attempt has been made to incorporate the effects of fluid rheology, tube diameter and tube configuration on the heat transfer phenomenon. The heat transfer measurements were carried out in a closed loop circulation rig with removable tubes (active and dummy both) such that different heat exchanger configurations could be tested.

A. Outline of the Experimental Programme:

In the present experimental programme¸ heat transfer data have been obtained by measuring local surface temperature of heat flux-probe and the bulk inlet and outlet temperatures of the test fluid. The effect of neighboring tubes on heat transfer from the surface of heated tube was studied by placing dummy tubes around the heater tube one by one. Experiments were conducted using five different tube arrangements as shown in Fig. 5. In order to investigate the effect of tube diameter (or p/d) on heat transfer, different tubes of diameter 3.18 cm, 2.54 cm and 1.91 cm have been used.

B. Test Fluid:

Water, aqueous solution of CMC (Carboxy Methy1 Cellulose) and PVA (Poly Viny1 Alcohol) have been taken as the test fluids. Water was selected as the test fluid because of its well known physical and thermal properties, as it is one of

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the most common Newtonian fluids. PVA and CMC solutions were found to obey the Ostwald power law model. These, being widely used in industries, are cheap and readily available and also for the fact that they do not suffer from thixotropy or ageing effects, were found to be the most suitable non-Newtonian test fluids. Two different concentrations of aqueous CMC solution viz. 1.0% and 1.5% and 0.5% PVA solutions were prepared. The polymer solutions were prepared by dissolving an appropriate weight of powder in required weight of water. The test fluid was vigorously stirred to dissolve the lumps till a homogeneous solution was obtained. These solutions were left in the tank for two days before starting the experiment.

IV. EXPERIMENTAL SET-UP

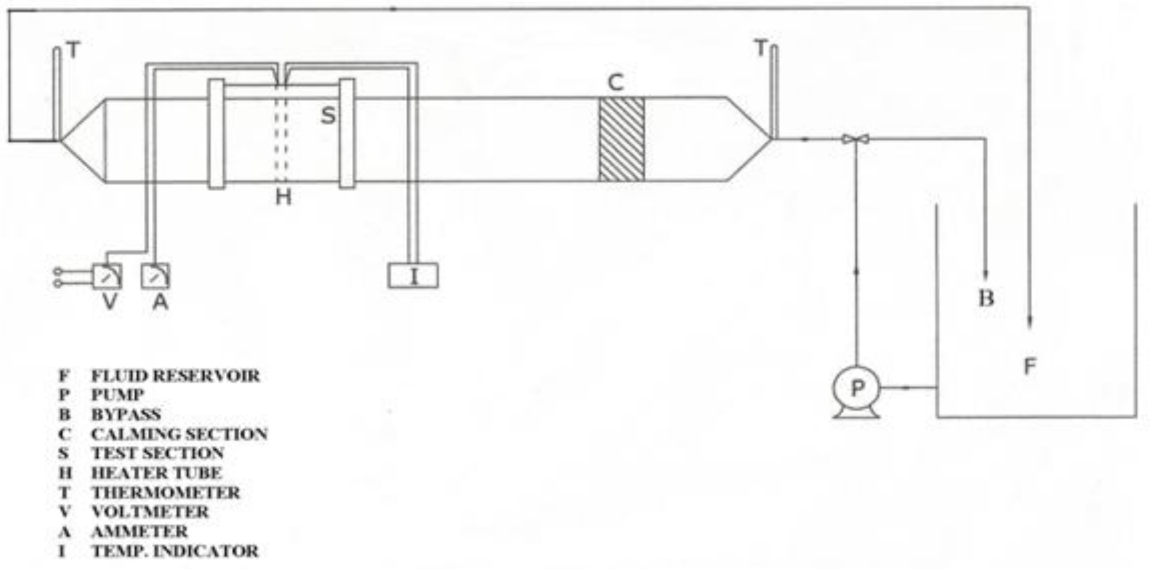

FIG.4 SCHEMATIC DIAGRAM OF EXPERIMENTAL SET-UP

A The schematic diagram of the experimental rig is shown in Fig. 4. It consisted of a 2.0 H.P. pump (P), connected through 2 inch diameter pipe to a fluid reservoir (F). A bypass (B) was provided at the delivery side of the pump to regulate the flow. The entrance portion of the test rig had divergent portion in shape which finally matches the dimension of test section. A calming section was provided with wire net at the entrance to eliminate the entrance effect and also to reduce eddies and reversal of flow.

The entire test rig may be divided into three major units:

A1. Test Section:

The test section of dimension 18.0 cm x 14.7 cm x 23.5 cm was fabricated with Perspex sheets. Three tubes of diameter 2.54 cm were half exposed on either side walls. The test section was so fabricated that it could accommodate desired number of tubes of different diameters. The top portion of the test section was provided with through holes of diameter 3.18 cm while in the bottom portion, partial grooves were provided to support the base of the tubes against the pressure of flowing fluids. To cover the top portion of the test section a top cover plate made of Perspex sheet was provided. The top cover plate was fastened with nuts and bolts thus making it removable, so that, whenever required the top cover plate could be removed and required number of dummy tubes could be placed in the test section. The heat flux probe or active tube was tightly fitted in the removable top cover plate and its location was so adjusted that it acquired the central position in the test section. Arrangement was made to accommodate any of different sized tubes of three different diameters in the present work that were studied. To accommodate the corresponding dummy tube, plastic adopters (plugs) were used to accommodate the tubes in the top and bottom plates. The removable heat flux probe placed centrally in the test section measured the heat flow per unit area.

The heat flux probe was fabricated from a copper tube 6 mm wall thickness. A pencil heater was inserted inside the copper tube to fit in the tube tightly. Four slots were cut at the outer surface of the rod which were at the location 0^0,

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90^0, 180^0 and 270^0 i.e., diametrically opposite to each other. Copper-constantan thermocouple wire, embedded in each slot, measured the local surface temperature of the heated tube, which, when averaged gave the average surface temperature of the heating probe.

To study the effect of neighbouring tubes and arrangement with longitudinal pitch S_L = 7.0 cm and transverse pitch S_T = 3.5 cm was used.

A whole tube bank consisting of thirty effective tubes [twenty seven tubes (including heater tube) and six half exposed tubes in side walls] of diameter 2.54 cm was also used for experiments with test fluids.

A2. Heating Arrangement:

A 300 watt pencil heater was placed inside the copper heating probe. The heater tube was provided an electrical heat flux through a rheostat. An ammeter (A) (range 0-2 amp.) and a voltmeter (V) (range 0-230 V) were used to measure the current and voltage. A constant supply of heat flux was provided by supplying constant voltage and current.

A3. Heat Transfer and Flow Measurement Unit:

The flow rate of the fluid was measured by collecting the fluid in a container for a known interval of time and weighing it on a balance having an accuracy of 2 gm. Two thermometers with accuracy of 0.1 ^{0}C were used to measure the inlet and outlet temperatures of the fluid flowing through the test section. The average value of inlet and outlet temperature gave the bulk temperature of the fluid flowing across the tubes. A calibrated digital temperature indicator having a range of 0-300^{0}C and a accuracy of 0.1^{0}C was used to record the surface temperature of the heater tube through four thermocouples fitted at different locations on the tube surface.

B. Experimental Procedure:

For a particular tube diameter and tube configuration the flow rate of a particular fluid was initially set to a predetermined value. A constant electrical heat flux was supplied to the heater and the surface temperatures were recorded through thermocouples placed at four difference location on the heater tube with the help of a multi-channel digital temperature indicator. Simultaneously the inlet and outlet temperatures of the fluid were also noted. The flow rate was varied and similar sets of data were recorded. This procedure was repeated for each tube diameter and tube arrangement.

Similar sets of observations were made using different fluids for each configuration and also for the whole tube bank consisting of thirty effective tubes of different diameters.

C. Tube Arrangements and Corresponding Nomenclature:

Following tube arrangements and corresponding nomenclature have been used for this experimental study:

1. Single heater tube – S
2. Test cylinder with a dummy tube in front – F
3. Test cylinder with a dummy tube in front and one on the right hand side – FR
4. Test cylinder with a dummy tube in front and one each on left and right hand side – FRL
5. Test cylinder with a dummy tube in front, one in back and one each on left and right hand side – FRLB
6. Whole tube bank consisting of thirty effective tubes – WTB

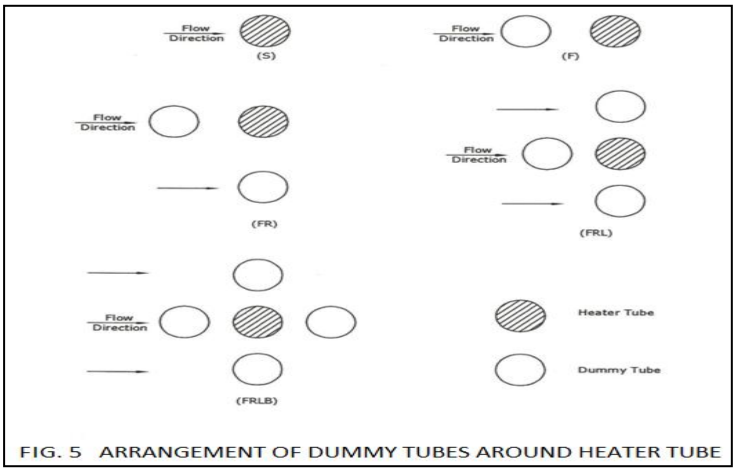

FIG. 5 ARRANGEMENT OF DUMMY TUBES AROUND HEATER TUBE

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In the arrangement 'WTB' the tubes were arranged in staggered manner ($S_T = S_L$ = 3.5cm) and the heater tube occupied central position in the fourth row of the bank. The experimental study on the arrangement 'WTB' has been carried out only with the tubes having a diameter 2.54 cm for the test fluids 1% CMC, water and 0.5% PVA only.

V. COMPARISON WITH EXPERIMENTAL RESULTS

Heat transfer is closely related to fluid dynamics, therefore, the two phenomena are considered simultaneously. Around the curved surface, there is complex process which mainly depends on the type of fluid and Reynolds number, A laminar boundary layer is formed on the front part of the circular tube facing the flow with its thickness increasing downstream. There is also a longitudinal pressure gradient caused by the curved surface. It may be noted that the point at which shear stress acquired the zero value the boundary layer can separate from the surface. The separation point is followed by a reverse flow, where the velocity vectors of fluid masses near the wall are in opposite direction. The fluid assuming reverse flow contacts the boundary layer and changes in to a vortex, which begins to rotate. The whole phenomenon of fluid behaviour is reflected into the local heat transfer rate. The fluid dynamics and heat transfer rate are also influenced by free stream turbulence, geometry and surface roughness etc.
The variation in the flow of fluid over a cylinder in cross flow gives rise to similar variation of local heat transfer.
For a Newtonian fluid starting from the front part, up to the separation boundary layer, heat transfer can be found analytically. For low Reynolds number flow of a Newtonian fluid, the heat transfer on the front part of the cylinder is maximum which decrease with the developing laminar boundary layer. However, at higher Reynolds number, the heat transfer gradually increases downstream of laminar boundary layer separation and is mainly determined experimentally.
For Non- Newtonian fluids, the fluid dynamic and heat transfer phenomenon are more complex in nature.
Zukauskus (1987) has suggested the use of following relation for evaluating heat transfer from bodies in cross flow of viscous fluids:

$$\mathrm{Nu} = \mathrm{c}\, Re^{a}\, Pr^{b} \left[\frac{Pr}{Pr_w}\right]^{r} \qquad (9)$$

In the present case , as the variation in the temperature is small ,the term (Pr/pr_w) may be considered approximately equal to 1.
The experimental result of 0.5% PVA, water, 0.5% CMC and 1.5% CMC have been processed and converted into the dimensionless groups viz. Nusselt number, Prandtl number and Reynolds number.
To study the affect of Prandtl number on heat transfer, a graph has been plotted in Fig. 6, showing the variation of $Nu.Re^{-1/2}$ versus Pr.

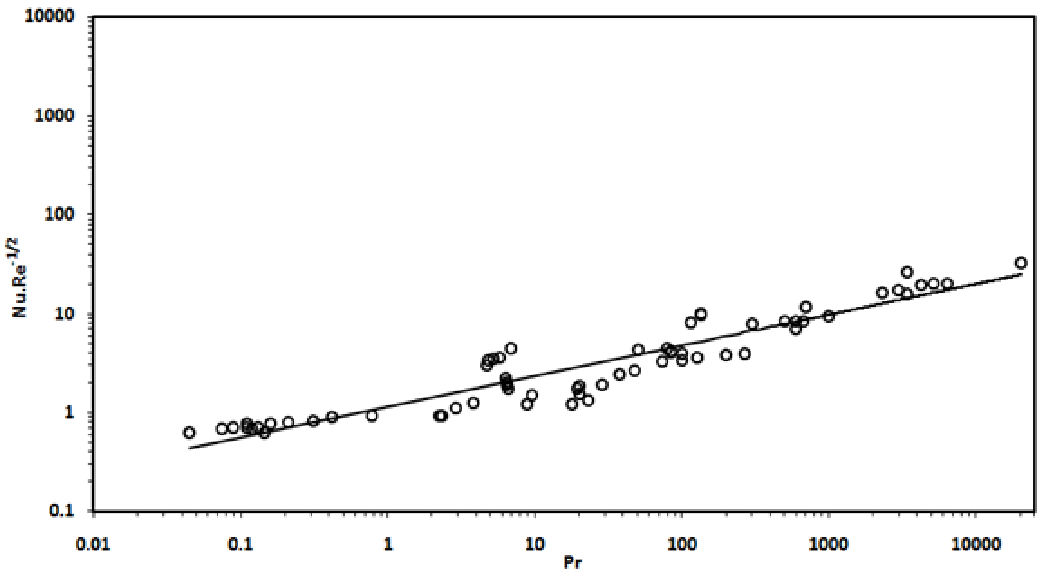

FIG. 6 EFFECT OF PRANDTL NUMBER ON HEAT TRANSFER FROM A CYLINDER : PLOT OF $Nu.Re^{-1/2}$ VERSUS Pr

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The slope of the line obtained is 0.33 giving a functional relationship:

$$\frac{Nu}{Re^{1/2}} = C\ Pr^{1/3} \quad (10)$$

The index 1/3, thus obtained for Prandtl number is in excellent agreement with the present analysis and also with the values reported (0.33 -0.37) by Zukauskus (1987), Richardson (1968) and others.

Equation (7) can also be expressed as:

$$Nu_{av} = \varphi(n,\theta) Re^{1/2}\ Pr^{1/3} \quad (11)$$

For $\theta = \frac{\pi}{2}$, the above equation reduces to,

$$Nu_{av} = c(n) Re^{1/2}\ Pr^{1/3} \quad (12)$$

where $c(n) = \varphi(n,\pi/2)$

In order to exclude the effect of flow behavior index (n) on $Nu_{av}\ Pr^{-1/3}$ a factor c(n) has been introduced as defined above. Values of c(n) for different fluids under study are:

Fluid	**c(n)**
0.5% PVA	0.95
Water	0.97
1.0% CMC	0.99
1.5% CMC	1.01

Although there is no significant variation in c(n), still it has been accounted for in the correlations.

VI. CONCLUSIONS

Based on the present theoretical analysis of the boundary layer theory and experimental studies the following significant conclusions can be drawn:

1. The theoretical analysis of the boundary layer theory, applied to the viscous flow and heat transfer from vertical tubes immersed in Power law fluids, indicates that the thickness of the hydrodynamic boundary layer is a function of the flow behavior index (n) and angular position (θ). For pseudo plastic fluids, it reduces near the stagnation point and then increases and for dilatants fluids it increases for all values of angular positions. Thermal boundary layer, however, shows a reverse trend. Accordingly, with increase in dilatancy (n > 1.0) the boundary layer thickness increases, thus decreasing the heat transfer rates. Whereas, with an increase in pseudo plasticity (n < 1.0), the boundary layer thickness decreases, thus increasing the heat transfer rates.
2. The present experimental results for heat transfer from single tube in cross flow are comparable with the theoretical analysis proving the validity of the model.
   For a single tube, $Nu \propto Re^{1/2}$
3. Based on the analysis, appropriate definition for the effective viscosity and Reynolds number for flow of non-Newtonian fluids past circular cylinders have been developed.
4. The average and local heat transfer coefficients from a cylinder have been shown to be a function of Re and Pr, and are related according to the following expression: $Nu = C\ Pr^{1/3}\ Re^{1/2}$
   The average heat transfer coefficient has been correlated as:
   $$\frac{Nu_{av} Pr^{-\frac{1}{3}}}{c(n)} = 0.35 + 0.87\ Re^{\frac{1}{2}}$$
5. In case of multi-tubular systems, the heat transfer for less viscous fluids has been found to be affected by the tube in front. Other neighbouring tubes do not seem to affect the heat transfer coefficient significantly. The data can be correlated by
   $$\frac{Nu_{av}\ Pr^{-1/3}}{c(n)} \left[\frac{p}{d}\right]^{0.2} = 1.46\ Re^{1/3} \quad \text{for } 1 < Re \leq 60$$
   $$\frac{Nu_{av}\ Pr^{-1/3}}{c(n)} \left[\frac{p}{d}\right]^{0.2} = 0.37\ Re^{2/3} \quad \text{for } 60 < Re < 10^4$$

VII. NOMENCLATURE

a	relative transverse pitch S_T/d.
b	relative longitudinal pitch S_L/d
B	width of the parallel place channel, m
c(n)	$\varphi(n, \pi/2)$ a function
C_p	specific heat at constant pressure, kJ/kg ^{0}C
d	diameter of the tubes in tube bank, m

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d_o outside diameter of the tubes in the bank, m
D gap between two plates at any axial position in the converging-diverging parallel plate channel model, m
D_1 minimum gap at x = 0, m
D_2 maximum gap at $x = x_0/2$, m
D_x gap between two plates at any axial position in the converging-diverging parallel plate channel model, m
g acceleration due to gravity, m^2 /sec
g_c conversion factor, kg-m/kg_f sec^2
k thermal conductivity, W/ m^2 °C
K power law consistency constant, kg/m $sec^{(2-n)}$
n flow behavior index
P absolute pressure, kg/m^2
S_L longitudinal pitch or spacing of longitudinal rows, m
S_T transverse pitch, m
u velocity component in x-direction, m/sec
U velocity, m/sec
x axial position in x direction, m
y axial position in y-direction, m
z height of the tube bank or exposed length of the tube in a tube bank, m

DIMENSIONLESS GROUPS:

Nu Nusselt number
Nu_{av} average Nusselt number around the tube
Pr Prandtl number
Pr_w Prandtl number at wall conditions
Re Reynolds number
Re′ Practicable Reynolds number

GREEK LETTERS:

ε void fraction calculated from tube spacing and geometry
μ coefficient of viscosity, kg/m sec
μ_{eff} effective viscosity, kg/m sec
θ angular position with respect to front stagnation point, degrees
τ shear stress, N/m^2
τ_w wall shear stress, N/m^2
Δ difference
ρ density of the fluid, kg/m^3

SUBSCRIPTS:

av average
H hydraulic
L longitudinal
o outer
T transverse
x in x direction
y in y-direction

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APPENDIX

Table A.1: Calculated Value of Boundary Layer Thickness $\left[(\delta/d_o)Re^{1/(1+n)}\right]$

θ	Flow Behaviour Index (n) 0.2	0.4	0.6	0.8	1	1.2	1.4	1.6	1.8	2.0
3	2.047	1.452	1.165	1.007	0.913	0.856	0.819	0.797	0.784	0.779
9	0.993	0.914	0.891	0.896	0.918	0.949	0.987	1.03	1.076	1.125
15	0.719	0.745	0.794	0.856	0.926	1.002	1.082	1.165	1.251	1.339
21	0.591	0.66	0.744	0.838	0.939	1.045	1.156	1.27	1.388	1.507
27	0.518	0.611	0.717	0.833	0.956	1.086	1.222	1.362	1.506	1.653
33	0.475	0.583	0.704	0.837	0.979	1.129	1.286	1.448	1.615	1.787
39	0.449	0.569	0.703	0.85	1.008	1.175	1.351	1.533	1.721	1.915
45	0.437	0.566	0.711	0.872	1.044	1.227	1.42	1.62	1.828	2.042
51	0.435	0.573	0.729	0.902	1.088	1.287	1.496	1.714	1.94	2.173
57	0.442	0.59	0.757	0.942	1.143	1.356	1.581	1.816	2.059	2.31
63	0.459	0.617	0.796	0.995	1.21	1.438	1.679	1.931	2.191	2.46
69	0.486	0.656	0.849	1.062	1.292	1.537	1.794	2.063	2.341	2.628
75	0.525	0.709	0.918	1.148	1.395	1.658	1.933	2.22	2.516	2.822
81	0.58	0.782	1.009	1.258	1.525	1.808	2.103	2.41	2.727	3.051
87	0.656	0.88	1.13	1.402	1.693	1.999	2.318	2.648	2.987	3.334
90	0.704	0.941	1.204	1.49	1.795	2.115	2.447	2.79	3.143	3.502

Table A.2: Effect of flow behaviour index on Local Heat Transfer $\left[\frac{Nu_\theta}{Pr^{1/3}Re^{1/2}}\right]$

θ	Flow Behaviour Index (n) 0.2	0.4	0.6	0.8	1	1.2	1.4	1.6	1.8	2
3	2.047	1.452	1.165	1.007	0.913	0.856	0.819	0.797	0.784	0.779
9	0.993	0.914	0.891	0.896	0.918	0.949	0.987	1.03	1.076	1.125
15	0.719	0.745	0.794	0.856	0.926	1.002	1.082	1.165	1.251	1.339
21	0.591	0.66	0.744	0.838	0.939	1.045	1.156	1.27	1.388	1.507
27	0.518	0.611	0.717	0.833	0.956	1.086	1.222	1.362	1.506	1.653
33	0.475	0.583	0.704	0.837	0.979	1.129	1.286	1.448	1.615	1.787
39	0.449	0.569	0.703	0.85	1.008	1.175	1.351	1.533	1.721	1.915
45	0.437	0.566	0.711	0.872	1.044	1.227	1.42	1.62	1.828	2.042
51	0.435	0.573	0.729	0.902	1.088	1.287	1.496	1.714	1.94	2.173
57	0.442	0.59	0.757	0.942	1.143	1.356	1.581	1.816	2.059	2.31
63	0.459	0.617	0.796	0.995	1.21	1.438	1.679	1.931	2.191	2.46
69	0.486	0.656	0.849	1.062	1.292	1.537	1.794	2.063	2.341	2.628
75	0.525	0.709	0.918	1.148	1.395	1.658	1.933	2.22	2.516	2.822
81	0.58	0.782	1.009	1.258	1.525	1.808	2.103	2.41	2.727	3.051
87	0.656	0.88	1.13	1.402	1.693	1.999	2.318	2.648	2.987	3.334
90	0.704	0.941	1.204	1.49	1.795	2.115	2.447	2.79	3.143	3.502