EFFECT OF TUBE BANK CONFIGURATION AND GEOMETRY ON HEAT TRANSFER COEFFICIENT FOR THE FLOW OF NEWTONIAN AND POWER LAW NON-NEWTONIAN FLUIDS FLOWING ACROSS TUBE BANKS

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ABSTRACT: Aim of this work is to study and investigate experimentally the effects of rheological behavior, tube diameter and tube arrangement on heat transfer to Newtonian and non-Newtonian fluids flowing across tube banks. Use of dimensionless numbers Nu, Pr and Re has been made on the basis of a bank of tubes represented by a set of diverging and converging parallel plate channel model being superior to the conventional capillary flow model. A methodology of determination of heat transfer coefficient experimentally at any specified (desired) location in a tube bank with desired numbers of tubes has been developed. Heat flux is supplied through a heated tube with desired number of dummy tubes placed around it at desired location. The heat transfer has been determined for different configuration and geometry by changing the number and diameter of dummy tubes and also their direction of location with respect to heated tube. The data has been obtained experimentally for the flow of Newtonian (water) and three non-Newtonian (1%, 1.5% aqueous CMC and 0.5% aqueous PVA) fluids flowing across each configuration and geometry in the present study. Based on experimental data following single valued correlations for both Newtonian and Power law non-Newtonian fluid is being suggested,

$$\frac{Nu_{av}\ Pr^{-1/3}}{c(n)}\left[\frac{p}{d}\right]^{0.2} = 1.46\ Re^{1/3} \quad \text{for } 1 < Re \le 60$$

$$\frac{Nu_{av}\ Pr^{-1/3}}{c(n)}\left[\frac{p}{d}\right]^{0.2} = 0.37\ Re^{2/3} \quad \text{for } 60 < Re < 10^4$$

KEY WORDS: aqueous, dummy tube, Newtonian, non-Newtonian, , single valued, tube bank, void fraction.

I. INTRODUCTION

The interest in studying the phenomena of heat transfer between a moving fluid and surfaces of bank of tubes immersed in it stems both from fundamental considerations, such as the better insights into the underlying physical processes, and from practical considerations. Such geometry also provides the testing ground for checking the validity of theoretical analysis. Consequently, considerable research effort has been expended in exploring and understanding the convective transport processes between moving fluids and the submerged tube banks having different arrangements.

Unfortunately, most of the fluids encountered in chemical and allied processing applications do not adhere to the classical Newtonian postulate and are accordingly known as non-Newtonian. Most particulate slurries (China clay and coal in water, sewage sledges etc.), multiphase mixtures such as oil-water emulsions, gas-liquid dispersions such as froths, foams and butter are non-Newtonian fluids, as are melts and solutions of high molecular weight and synthetic polymers. Further examples of systems displaying a variety of non-Newtonian characteristic include pharmaceutical formulations, cosmetics and toiletries, paints, synthetic lubricant, biological fluids (blood, synovial fluid, saliva etc.) and food stuffs (jams, jellies, soups, marmalades etc.). Indeed non-Newtonian behavior is so widespread that it would be no exaggeration to say that the Newtonian fluid behavior is an exception rather than a rule. Frequent occurrences of such fluids in industries have created considerable interest in the study of the behavior of these fluids in various process-equipments.

Different models and techniques have been put forward to describe the flow behavior and heat transport phenomena in cross flow heat exchangers. These models suffer from one discrepancy or the other. The results based on these models are fairly inaccurate. These models not only use an incorrect characteristic length parameter but also do not account for the real flow situation in the tube bank and also the influence of the tube geometry has been ignored. A cell model was attempted by Hwang and Yao (1986) in which the hypothetical fluid cell was assumed to have zero shear stress at its outer surface. But the solution provided by this model is not of much practical importance because it is applicable to creeping flow for which the Reynolds number is restricted to Re ≤ 1 and Peclet number Pe ≥ 1.

Zukauskas (1972) presented a novel approach in the analysis of heat transfer across the tube bank by proposing that the flow across a bundle of tubes is comparable to that in the parallel plate channel. In this approach the Reynolds number calculation is based on equivalent hydraulic diameter and average velocity. The common use of maximum flow velocity at minimum passage does not give a single valued correlation for the tube banks of different configurations. It is reported by Prakash et al. (1987) that the parallel plate channel model can give better correlation of the pressure drop data for the flow of both Newtonian and non-Newtonian fluids.

Zukauskas (1968, 1972, 1982, 1985 and 1987), Adams (1968), Achenbach (1975), Hwang and Yao (1986), Snyder (1953), Aiba (1976, 1980 and 1990) Mandhani et.al (2002), Mangadoddy et.al (2004) and Khan et.al (2006) extensively studied the heat transfer and flow pattern across single tube and bundle of tubes. The results presented by Zukauskas clearly showed that the heat transport phenomena in a tube bank is influenced by several factors, like tube geometry, nature of flow and temperature of heating or cooling surface and that of the bulk of fluid etc.

A thorough review of the currently available literature reveals that the attention has mainly been focused on the heat transport phenomena of Newtonian fluids across tube banks. Only the average value of heat transfer across the tube bank has been reported and no effort has been made to study the transport behavior at any specified location within the tube bank for non-Newtonian fluids. Also the effect of tube spacing (i.e., p/d) and study of dilatants behavior remain untouched so far.

AIMS OF STUDY:

Objective of this work is to investigate and study experimentally the effects of rheological behavior, tube diameter and that of the tube arrangement on heat transfer in tube banks and to find a single valued correlation for heat transfer to Newtonian and non-Newtonian fluids flowing across tube banks.

II. THEORETICAL ANALYSIS

2.1 CONVERGING DIVERGING PARALLEL PLATE CHANNEL MODEL:

Many workers have made attempts to extend the capillary flow model for flow through packed beds to flow across tube bundles (Galloway and Sage 1964, Hughmark 1972, Zukauskas 1972, Vossoughi and Seyer 1974 and Prakash et al. 1987). Vossoughi and Seyer (1974) modeled the tube bundle as a series of parallel plate channels and modified the capillary tube bundle expression. Prakash et al. (1987) argued that for flow across staggered tube bundles, the

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fluid flows through the gaps between adjacent tubes where the cross section is periodically converging and diverging. This assumption also gets support from the photographic studies of the flow in staggered tube bundles as reported by Lohrish (1929). Based on these arguments Prakash et al. (1987) proposed a converging diverging channel model for analysing the pressure drop data during flow across staggered tube bundles. In the present work this model has been extended to analyse the heat transfer from such tube bundles.

Prakash et al. (1987) proposed that the staggered tube arrangement can be considered as a bundle of parallel converging-diverging channel. They further assumed that each channel consists of two plates and the gap D_X between the two surface of the channel is a sinusoidal function of axial position x (Refer Fig. 1):

$$Dx = D_1 + B\left[1 - cos2\pi(x/x_0)\right] \quad (1)$$

where, D_x= gap between the plates at any axial position x

D_1 = minimum gap at x=0 and D_2 = maximum gap at $x=x_0/2$

Hence, $B_1 = \frac{D_2 - D_1}{2}$

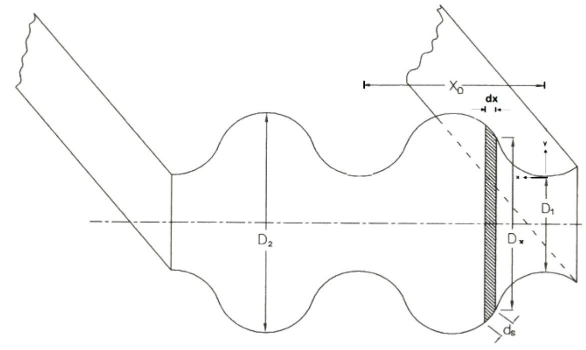

FIG. 1 PERIODICALLY CONVERGING AND DIVERGING PARALLEL PLATE CHANNEL

From Fig 1 it is seen that the cross-section is a function of axial position, suggesting that the velocity profile and the pressure gradient are also a function of the axial position 'x'. For simplicity it is further assumed that the pressure gradient at any position is equal to that prevailing in a straight parallel plate channel, of the same hydraulic radius as that existing in the converging-diverging channel, at that position.

For convenience the equation (1) is simplified as

$$D_x = \frac{D_2 + D_1}{2}\left[1 - \frac{D_2 - D_1}{D_2 + D_1} cos2\pi\left({}^{x}/_{x_0}\right)\right]$$

$$\text{Or, } Dx = \frac{D_1}{1-b}\left[1 - b\, cos2\pi\left({}^{x}/_{x_0}\right)\right] \quad (2)$$

Where, $b = \frac{D_2 - D_1}{D_2 + D_1}$

Volume of one cycle of length x and width z may be obtained as

$$= \int_0^{x_1} D_x . z\, dx = \int_0^{x_1} \frac{D_1 z}{(1-b)}\left[1 - \mathrm{b}\cos 2\pi(\mathrm{x}_{\mathrm{x}_0})\right] dx$$

$$= \frac{\boldsymbol{D_1 z}}{\boldsymbol{(1-b)}} \frac{\boldsymbol{x_0}}{\boldsymbol{2\pi}} \int_{\boldsymbol{0}}^{\boldsymbol{x_1}} (1 - \mathrm{b}\cos\theta)\, d\theta \quad = \frac{\boldsymbol{D_1 z\, x_0}}{\boldsymbol{(1-b)}} \quad (3)$$

where, $\theta = 2\pi\left({}^{x}/_{x_0}\right)$

The average cross-sectional area may be given as :

$$A_c = \frac{Volume}{x_0} = \frac{D_1 z\, x_0}{(1-b)\, x_0} = \frac{D_1 z}{(1-b)} = \left(\frac{D_1 + D_1}{2}\right) z \quad (4)$$

average velocity :

$$\bar{U} = \frac{Q}{A_c} = \frac{Q}{\left({}^{volume\ \ of\ the\ unit}/_{x_o}\right)} = \frac{Q\, x_0}{D_1\, x_0\, z}\,(1-b) = \frac{Q}{z}\left[\frac{1-b}{D_1}\right]$$

$$\text{Or, } \bar{U} = \frac{2\,Q}{z\,(D_2 + D_1)} \quad (5)$$

Thus average velocity is obtained at the position where the gap between the plates equals to

$\frac{D_1 + D_2}{2}$

The length of the curved surface for one unit:

$$S_{x_0} = \int_0^{x_0} ds$$

$$= \int_0^{x_0}\left[1 + \left(\frac{dy}{dx}\right)^2\right]^{1/2} dx$$

$$= \int_0^{x_0}\left[1 + \left(\frac{1}{2}\frac{dD_x}{dx}\right)^2\right] dx$$

Differentiating equation (2), we get

$$\frac{dD_x}{dx} = \frac{2\pi\, b\, D_1}{x_0\,(1-b)}\, Sin\left[2\pi\frac{x}{x_0}\right]$$

$$\text{Thus, } S_{x_0} = \int_0^{x_0}\left[1 + \left\{\frac{2\pi\, b\, D_1}{2\, x_0 (1-b)}\right\}^2 Sin^2\left(2\pi\frac{x}{x_0}\right)\right]^{1/2} dx$$

Let $\left[2\pi\frac{x}{x_0}\right] = \theta$, $x = \frac{x_0}{2\pi}\theta$ and $B = \frac{\pi\, b\, D_1}{x_0(1-b)}$

$$S_{x_0} = x_0\left[1 + \frac{1}{4}B^2 - \frac{3}{64}B^4 + \cdots\right]$$

For B << 1

$$S_{x_0} = x_0\left[1 + \frac{B^2}{4}\right] \quad (6)$$

2.2 LAMINAR FLOW OF NEWTONIAN AND NON-NEWTONIAN FLUIDS ACROSS BANKS OF CIRCULAR TUBES:

For the analysis and application of the converging-diverging parallel channel model describing the cross flow patterns in a tube bundle, the following few parameters are defined as shown in Fig.2.

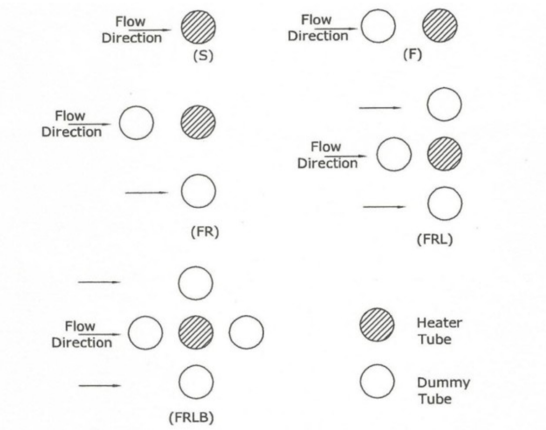

FIG. 2 ARRANGEMENT OF DUMMY TUBES AROUND HEATER TUBE

S_L Centre to centre distance between adjacent transverse rows (longitudinal pitch.)

S_T Centre to centre distance between the tubes in longitudinal rows (transverse pitch).

The distance between tubes in adjacent transverse rows for staggered arrangement is given as :

$$S_L' = \left[S_L^2 + \left({}^{S_T}/_{2}\right)^2\right]^{1/2} \quad (7)$$

d_0 = outer diameter of tube

N_T = number of transverse rows of tubes

L = length of the tube bundle.

ε = void fraction of the tube bundle.

A common equivalent hydraulic diameter for the tube based on the free cross-sectional area and the wetted perimeter is defined as:

$$D_H = 4\left[\frac{free\ cross-sectional\ \ area}{wetted\ \ perimeter}\right]$$

$$= 4\left[\frac{free\ \ volume}{wet\,ted\ \ surface\ \ area}\right]$$

$$= 4\left[\frac{S_T.S_L - \pi d_0^2/4}{\pi d_0}\right] \quad (8)$$

Void fraction, $\varepsilon = \frac{S_T.S_L - \pi d_0^2/4}{S_T.S_L} = 1 - \frac{\pi}{4}\left[\frac{d_0}{S_T}\right]\left[\frac{d_0}{S_L}\right]$

$$S_T\, S_L = \frac{\pi}{4}\left(\frac{d_0^2}{1-\varepsilon}\right)$$

$$D_H = 4\left[\frac{\frac{\pi}{4}\left(\frac{d_0^2}{1-\varepsilon}\right) - \frac{\pi}{4}d_0^2}{\pi\, d_0}\right] \quad (9)$$

$$= \left(\frac{\varepsilon}{1-\varepsilon}\right) d_0$$

Actual average velocity in the tube banks

$$\bar{U} = \text{superficial velocity}/\varepsilon = \frac{U_s}{\varepsilon} \quad (10)$$

In staggered arrangement, the flow patterns are entirely different than that in an in-line arrangement. In both these cases a boundary layer that forms on the forward portion of the tube ultimately separates from the rear portion of the tube surface producing a turbulent wake. In the in-line arrangement the turbulent wake continues to the next transverse row. In the unobstructed space the fluid experiences a contraction and expansion once while flowing through in the length S_L.

In staggered tube arrangement the turbulent wake behind each tube reduces considerably, while the fluid experiences contraction and expansion during flow through a converging diverging space at least four times in a span of length of $2S_L$. For an inline arrangement, the length of the flow path is actually the length of the tube bank L. The sinusoidal nature of the flow path in a staggered tube bank makes the flow length more than the actual length of the tube bank.

The flow pattern in tube banks is completely changed due to presence of neighboring tubes. A converging diverging parallel plate channel model for predicting the pressure drops when extended to predict the heat transfer from tube in tube bank give the following relations.

$$\frac{h_{av}\, D_H}{k} = \frac{3}{2}\,\frac{1}{0.893}\left[\frac{12}{9}\right]^{1/3}\left[\frac{D_H\, U\, \rho}{\mu_{eff}}\,\frac{\mu_{eff}}{\rho\, \alpha}\,\frac{D_H}{x_s}\right]^{1/3}\left[\frac{2n+1}{3n}\right]^{1/3} \quad (11)$$

where effective viscosity is

$$\mu_{eff} = K\left[\frac{2n+1}{3n}\right]^{n}\left[\frac{12\, U}{D_H}\right]^{n-1} \quad (12)$$

$$Nu = \frac{h_{av}\, D_H}{k} = 1.85\left[\frac{2n+1}{3n}\right]^{1/3} Re^{1/3}\, Pr^{1/3}\left[\frac{D_H}{x_s}\right]^{1/3} \quad (13)$$

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.

3.1 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. 2. 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.

3.2 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 the most common Newtonian fluid. 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.

3.2.1 PHYSICAL AND THERMAL PROPERTIES:

For conversion of heat transfer data into dimensionless groups, the physical properties like density and viscosity and thermal properties like specific heat, thermal conductivity, etc. are required to be determined. Although for well explored systems, the data available in the literature can be used, yet, for a new experimental system it is mandatory to determine these properties.

The properties for water have been taken from literature. For CMC and PVA solutions these properties were determined. The densities at different temperatures were measured by a density bottle. A capillary tube viscometer was used to determine the rheological properties of non- Newtonian test fluids which eventually resulted into calculation of viscosity. Measurement of thermal conductivity was done with the help of a parallel plate apparatus.

IV. EXPERIMENTAL SET UP:

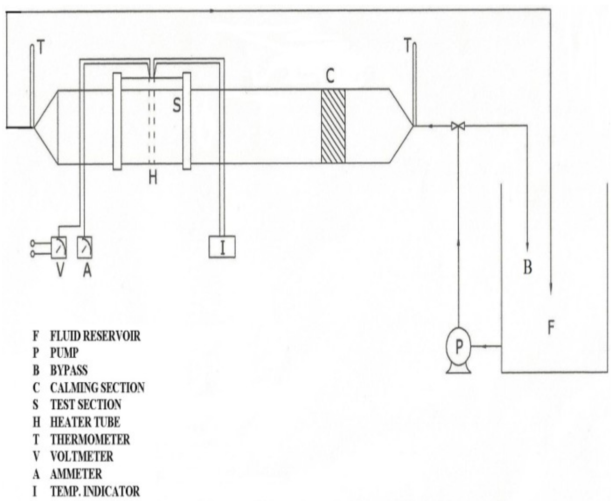

FIG.3 SCHEMATIC DIAGRAM OF EXPERIMENTAL SET-UP

The schematic diagram of the experimental rig is shown in Fig. 3. It consisted of a 2.5 H.P. pump (P), connected to a steel 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:

4.1 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 pensil 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, 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.

4.2. 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 0-1 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.

4.3. 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.

4.4. 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.

4.5 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

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. RESULTS AND DISCUSSION

5.1 Heat Transfer with Multi-tube Arrangement:

5.1.1 Average Heat Transfer Coefficient:

First of all the variation of h_{av} with U_{av} were examined for various tube arrangements F, FR, FRL, FRLB, for all the four test fluids used in this work. It was found that the heat transfer coefficient is decreasing with the increasing viscosity of the fluid. Similarly the h_{av} were plotted against U_{av} for all tube arrangements exposed to 0.5% PVA (tube diameter= 2.54 cm), water, 1% CMC and 1.5% CMC (tube diameter=3.18 cm). It is observed from those plots that except for water, the tube arrangement has no significant effect on the value of h_{av} for a given flow velocity. This could be attributed to the suppression of vortices in the wake by the non-Newtonian fluids.

5.1.2 Correlation of Experimental Data:

The flow pattern around the tube in a bundle differs considerably from that around an isolated tube immersed in the fluid flowing across it, as the former is influenced by the presence of neighboring tubes, Also change in the boundary layer, formed around a cylinder is affected considerably by the neighboring tube which consequently results into considerable change in the heat transfer characteristics. Further, the turbulence generated by the leading tube in general enhances the heat transfer; this change is more significant at higher Reynolds numbers. The effects on boundary layer and extent of turbulence, both are dependent upon the arrangement tube spacing etc. The heat transfer in a bundle of tubes is thus a function of tube arrangement, spacing between the adjacent tubes and other geometrical parameters. The present investigation has been, therefore, carried out to study the effect of pitch to diameter ratio (i.e., p/d) on heat transfer for different tube arrangements. For this purpose the experimental data for all the tube diameters, fluids and different arrangements viz., ‘F’, ‘FR’, ‘FRL’ and ‘FRLB’ have been plotted together in **Fig .4**.

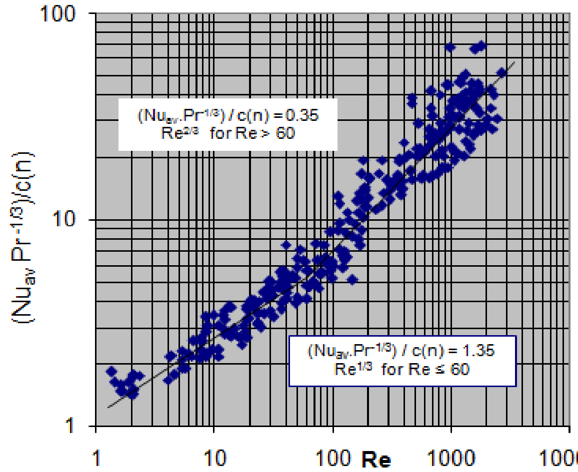

FIG.4 HEAT TRANSFER DURING FLOW THROUGH ASSEMBLY OF TUBES: VARIATION OF (Nu_{av} $Pr^{-1/3}$)/c(n) WITH REYNOLDS NO.

A careful examination of this plot reveals that experimental values show different trend for different range of Reynolds number. For lower Reynolds number (Re≤ 60), the slope is somewhat lower than that for the data in the higher Reynolds number range (Re>60). For Re≤ 60, the slope is found to be 1/3 while for Re> 60 a slope of 2/3 is observed. Separate analysis of each of the above regions results into the following correlations:

$$\frac{NuPr^{-1/3}}{c(n)} = 1.35\ Re^{1/3} \quad \text{for Re} \le 60 \tag{14}$$

$$\frac{NuPr^{-1/3}}{c(n)} = 0.35\ Re^{2/3} \quad \text{for Re} > 60 \tag{15}$$

To investigate the effects of tube diameter and tube spacing on heat transfer characteristics, a dimensionless number defined as p/d has been chosen, where p is the centre to centre distance between the tubes and d is the outside tube diameter. The parameter (p/d) directly affects the heat transfer and can be incorporated in the correlation as:

$$\frac{NuPr^{-1/3}}{c(n)} = C\ Re^{m} \tag{16}$$

where, $C = C_1\ (p/d)^b$; both C and C_1 are constants.

To determine the value of index (b), data for each arrangement for different tube diameters have been examined separately. $\frac{Nu\,Pr^{-1/3}}{c(n)}$

are plotted against Reynolds number for different arrangements and tube diameters. From these plots it is seen that the data for each system follow some general trend, but the values of constant (C) are dependent upon tube arrangement and tube diameter.

These plots result into separate equations for the two ranges of Reynolds number (Re $\leq$ 60 and Re > 60) for different tube configurations. These relations are tabulated below:

Arrangement	Tube Diameter Cm	$\frac{Nu_{av} Pr^{-1/3}}{c(n)}$	
		for Re $\leq$ 60	for Re > 60
F	3.18	1.25 $Re^{1/3}$	0.32 $Re^{2/3}$
F	2.54	1.0 $Re^{1/3}$	0.26 $Re^{2/3}$
F	1.91	1.15 $Re^{1/3}$	0.29 $Re^{2/3}$
FR	3.18	1.55 $Re^{1/3}$	0.40 $Re^{2/3}$
FR	2.54	1.33 $Re^{1/3}$	0.34 $Re^{2/3}$
FR	1.91	1.17 $Re^{1/3}$	0.30 $Re^{2/3}$
FRL	3.18	1.35 $Re^{1/3}$	0.35 $Re^{2/3}$
FRL	2.54	1.21 $Re^{1/3}$	0.31 $Re^{2/3}$
FRL	1.91	1.31 $Re^{1/3}$	0.34 $Re^{2/3}$
FRLB	3.18	1.21 $Re^{1/3}$	0.31 $Re^{2/3}$
FRLB	2.54	1.30 $Re^{1/3}$	0.33 $Re^{2/3}$
FRLB	1.91	1.21 $Re^{1/3}$	0.31 $Re^{2/3}$

It is seen that coefficients of $Re^{1/3}$ and $Re^{2/3}$ are dependent on (p/d). To determine the exponent of (p/d), values of coefficients were plotted against (p/d) as shown in Fig.5.

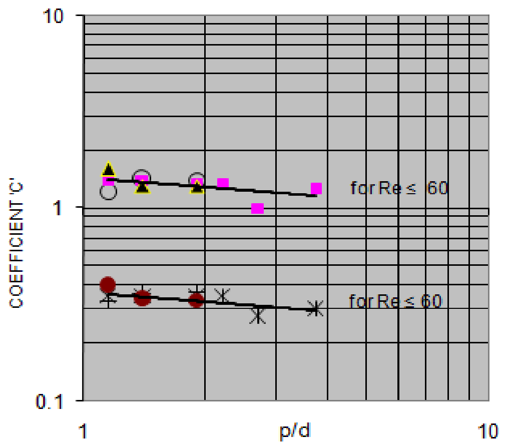

FIG.5 VARIATION OF COEFFICIENT 'C' OF Eq.16 with Pitch/ Diameter (p/d) Ratio

The value of the exponent was found as 0.2 for both the regions i.e. for Re $\leq$ 60 as well as Re $\geq$ 60.

Thus, $C = C1 \left[\frac{p}{d}\right]^{-0.2}$ (17)

Hence, the heat transfer $\frac{Nu_{av}\ Pr^{-1/3}}{c(n)} \propto \left[\frac{p}{d}\right]^{-0.2}$

Now from equation (16)

$$\frac{Nu_{av}\ Pr^{-\frac{1}{3}}}{c(n)} = C_1 \left[\frac{p}{d}\right]^{-0.2} Re^m$$

Or $\frac{Nu_{av}\ Pr^{-1/3}}{c(n)} \left[\frac{p}{d}\right]^{0.2} = C_1\ Re^m$ (18)

Where, m = 1/3 for Re $\leq$ 60 and 2/3 for Re > 60.

The heat transfer parameter, thus obtained, has been plotted against Reynolds number in **Fig. 6**.

The comparison of Fig. 4 and Fig. 6 reveals that data treated with the factor (p/d) has converged and the deviation has reduced to about 11.73% as compared to 12.54% in the former case, for the region Re $\leq$ 60 Similarly, for the higher Reynolds number range, i.e., Re > 60, also a decrease in deviation of the experimental data from the line representing proposed correlation has been observed. Thus the final correlations obtained are:

$$\frac{Nu_{av}\ Pr^{-1/3}}{c(n)} \left[\frac{p}{d}\right]^{0.2} = 1.46\ Re^{1/3}, \text{ for } 1 < Re \leq 60$$

$$\frac{Nu_{av}\ Pr^{-1/3}}{c(n)} \left[\frac{p}{d}\right]^{0.2} = 0.37 Re^{2/3}, \text{ for } 60 < Re < 10^4$$

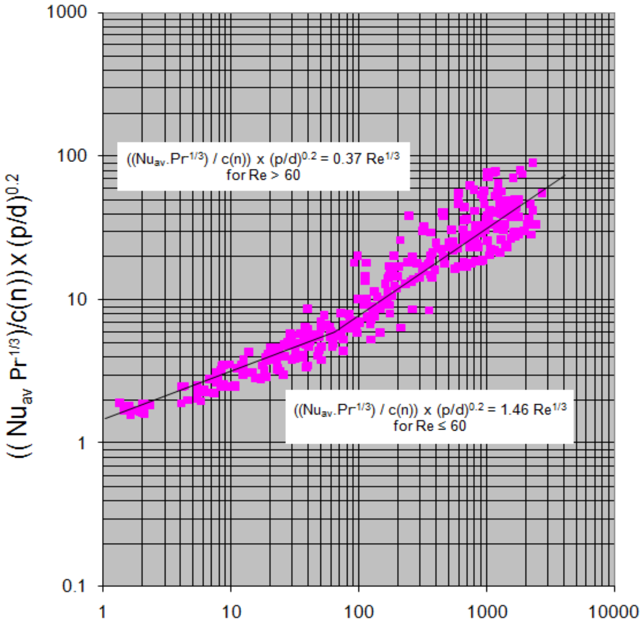

FIG.6 CORRELATION OF AVERAGE HEAT TRANSFER DATA: PLOT OF $((Nu_{av}\ Pr^{-1/3})/c(n)))$ x $(p/d)^{0.2}$ versus REYNOLDS NUMBER

It is, therefore, evident that the factor (p/d) can successfully be used to eliminate the effect of tube diameter and tube arrangement to arrive at a unique correlation which is applicable for heat transfer from multi-tubular arrangements. Further heat transfer has been studied in a complete tube bank assembly also for the flow of 1.0% CMC, water and 0.5% PVA. The outside diameter of the tube in the arrangement was 2.54 cm. The active tube occupied a position in the fourth row of tube bank. Comparison of data for arrangements 'FRLB' and 'WTB' as shown in **Fig. 7** reveal that the difference between the data for two aforesaid arrangements is not significant. Thus the correlations suggested above are applicable for the whole of the tube bank and not only for the situations for which analysis of the experimental data has been carried out.

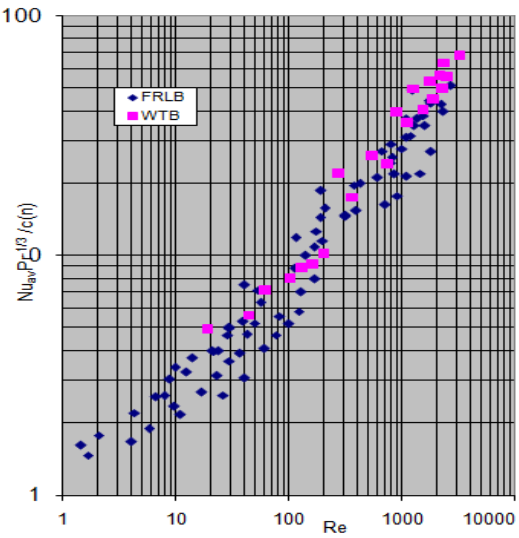

FIG 7: COMPARISON OF AVERAGE HEAT TRANSFER FOR SHORT (FRLB) AND LONG (WTB) TUBE BUNDLE ARRANGEMEMNTS: PLOT OF $Nu_{av}Pr^{-1/3}/c(n)$ VS Re

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VII. NOMENCLATURE

A_c	average area of cross-section of the tube bank, m^2
b	relative longitudinal pitch, S_l / d
B	width of the parallel place channel, m
c(n)	ø(n, π/2) a function
d	diameter of the tube bank, m
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_o /2, m
D_H	hydraulic diameter of the tube bank, m
g	acceleration due to gravity, m^2/sec
g_c	conversion factor, Kg-m/Kgf sec^2
h_{av}	average convective heat transfer coefficient, w/ m^2 °C
k	thermal conductivity, W/ m^2 °C
K	power law consistency constant, Kg/m $sec^{(2-n)}$
n	flow behavior index
Q	volumetric flow rate, m^3 /sec
S_L	longitudinal pitch or spacing of longitudinal rows, m
S_T	transverse pitch, m
U	velocity, m/sec
Ū	average velocity, m/sec
U_s	superficial 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
Pe	Peclet number
Pr	Prandtl number
Re	Reynolds number
Re	Reynolds number based on tube diameter

GREEK LETTERS:

α	thermal diffusivity
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

SUBSCRIPTS:

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

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