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Research Paper

THE EFFECT OF NEIGHBOURING TUBES ON HEAT TRANSFER COEFFICIENT FOR NEWTONIAN AND NON-NEWTONIAN FLUIDS FLOWING ACROSS TUBE BANK

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The experimental investigation of heat transfer coefficient for Newtonian and non-Newtonian fluids, obeying Power-law, has been carried out by measuring local surface temperature of heated tube and bulk inlet and outlet temperatures of test fluid. The effect of neighbouring tubes was studied by placing dummy tubes around it one by one. Experiment was conducted with single tube and using four more tube arrangements. The following correlation has been obtained for heat transfer for the flow of water, 0.5% PVA and 1% and 1.5% CMC solutions across tube bank.

$$\frac{Nu\,\mathrm{Pr}^{1/3}}{c(n)} = 1.35\ \mathrm{Re}^{1/3} \quad \text{for Re} \le 60$$

$$\frac{Nu\,\mathrm{Pr}^{1/3}}{c(n)} = 0.35\ \mathrm{Re}^{2/3} \quad \text{for Re} > 60$$

Keywords: Non-newtonian, Power-law, Tube bank, Cross-flow, Dummy

INTRODUCTION

The flow of fluids over an array of rods or cylinder represents an idealization of several important industrial processes. The power generation field, the chemical industry, and other technology based industries increasingly employ heat exchangers involving tubes in cross-flow. The cross-flow over the bank of tubes represents the true picture of the flow encountered in the shell side heat exchangers. Consequently, considerable research effort has been expended in exploring and understanding the convective transport process between moving fluids and submerged tube banks having different type of arrangements.

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.

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Most particulate slurries (China clay, coal in water, sewage sludge, multiphase mixtures such as oil water emulsions, gas-liquid dispersions such as froths and foams and butter) are non-Newtonian fluids. Melts and solutions of high molecular weight naturally occurring and synthetic polymers are found to be non-Newtonian. Further examples of systems displaying a variety of non-Newtonian characteristics include pharmaceutical formulations, cosmetics and toiletries, paints, synthetic lubricants, biological fluids (blood, synuvial fluid, saliva, etc.), and food stuffs (jams, jellies, soups, marmalades, etc.).

Frequent occurrence of such fluids in industries has created considerable interest in the study of bahaviour of these fluids in various process equipments where these are subjected to heating and/or cooling in heat exchangers. The design of cross flow heat exchanger is a complex task requiring the examination and optimization of a wide variety of design parameters such as friction factor, heat transfer, Reynolds number, etc. Heat transfer is considerably influenced by flow regime around the tube. Therefore, the need of an appropriate model describing the real flow situation with in the heat exchanger cannot be overlooked.

Several models and techniques have been put forward to describe the flow behaviour and heat transfer phenomena in cross flow heat exchangers which suffer from one discrepancy or the other. The results based on above models are fairly inaccurate. Zukauskas (1972) proposed that the flow across a bundle of tubes is comparable to that in a parallel plate channel. The fluid flow was approximated as that of flow through a series of parallel plates with spacing equal to minimum space between the tubes. The model was later modified to converging-diverging parallel plate channel model to account for fluid contraction between two adjacent tubes and expansion elsewhere. The flow pattern is assumed to follow a sinusoidal path in the direction of flow and Reynolds number calculation is based on equivalent hydraulic diameter and average velocity. It is reported by Prakash *et al.* (1987) that the parallel plate channel model can give better correlation of pressure drop data for the flow of both Newtonian and Non-Newtonian while the analogy between heat and momentum is restricted to the condition of low Reynolds number range.

Snyder (1953), Zukauskas (1968), (1972), (1982), (1985) and (1987), Adams (1968), Achenbach (1975), Hwang and Yao (1986), Aiba (1976), (1980) and (1990) extensively studied the heat transfer and flow pattern across single tube and bundle of tubes. Zukauskas presented extensive theoretical as well as experimental information regarding design parameters for flow across a tube and tube bank. The results presented by Zukauskas clearly showed that the heat transfer phenomena in a tube bank is influenced by several factors like tube geometry, nature of flow, temperature of heating or cooling surface and that of bulk of fluid, etc., He, however, focused his attention only on the average value of heat transfer across tube bank like many other workers and no effort has been made to study the transport at any specified location within the tube bank for the flow of non-Newtonian fluids. It is rather interesting and useful to determine as to how addition of tubes around a single tube, in the formation of tube bank, change the flow pattern which in turn changes the heat transfer coefficient. Also the effects of rheological properties of non-Newtonian fluids and tube diameter be found to analyse the heat transfer. In order to find the above experimentally a new methodology has been adopted in which desired number of tubes at desired location may be arranged and for different tube geometries the heat transfer coefficient may be calculated for different flow rates of Newtonian as well as non-Newtonian fluids.

THEORETICAL

The visualization of the physical picture of cross flow and heat transfer pattern from the photograph published by Lorich (1929) reveals that in tube bank the flowing fluids is not in continuous contact with heat transfer surface but experiences a short contact and after contacting a part of tube, find a gap and then moves to the next tube following either a slightly curved path (inline arrangement) or a sinusoidal path (staggered arrangement). Refer Figure 1. Due to short contact for a short period, the temperature distribution does not attain a fully developed profile but the gradient of the temperature with respect to direction perpendicular to the flow remains confined to a very thin region near to the surface.

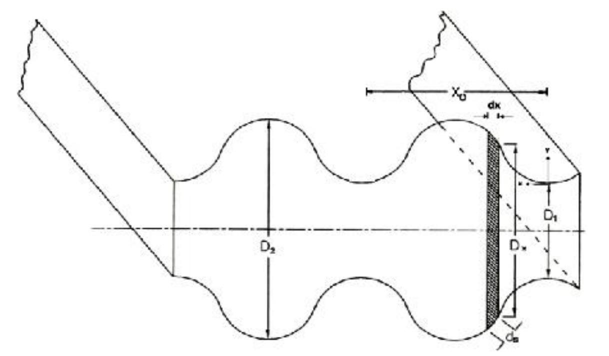
Thus

$$v << u \text{ and } \frac{\partial^2 T}{\partial x^2} << \frac{\partial^2 T}{\partial y^2} \quad ...(1)$$

Following the above assumptions, for the present model, the steady-state differential Equation (1) reduces to

$$u\frac{\partial T}{\partial x} = \sim \frac{\partial^2 T}{\partial y^2} \quad ...(2)$$

Figure 1: Periodically Converging and Diverging Parallel Plate Channel



For a constant property fluid in laminar flow the solution of Equation (2) may be obtained by assuming high velocity across short contact surface of the tubes. The velocity distribution in the boundary layer is assumed to be confined to a very thin region over the tube surface for laminar flow. The velocity profile thus becomes:

$$u = \left(\frac{\partial u}{\partial y}\right)_w y \quad ...(3)$$

and consequently Equation (2) takes the form,

$$\left(\frac{\partial u}{\partial y}\right)_w \frac{\partial T}{\partial x} = \sim \frac{\partial^2 T}{\partial y^2} \quad ...(4)$$

Applying the following boundary conditions

at $x = 0$, $y = 0$, $T = T_b$

at $x > 0$, $y = 0$, $T = T_w$

and solving we get

$$h_x = \frac{K}{0.893}\left(\frac{v}{9\sim x}\right)^{1/3} \quad ...(5)$$

and

$$h_{av} = \frac{3}{2}\frac{K}{0.893}\left(\frac{v_a}{9\sim x_s}\right)^{1/3} \quad ...(6)$$

where

$$_{v} = \left(\frac{2n+1}{3n}\right)\left(\frac{12U}{2D_x}\right) \quad ...(7)$$

and

$$_{v_a} = \left(\frac{2n+1}{3n}\right)\left(\frac{12U}{D_H}\right) \quad ...(8)$$

where,

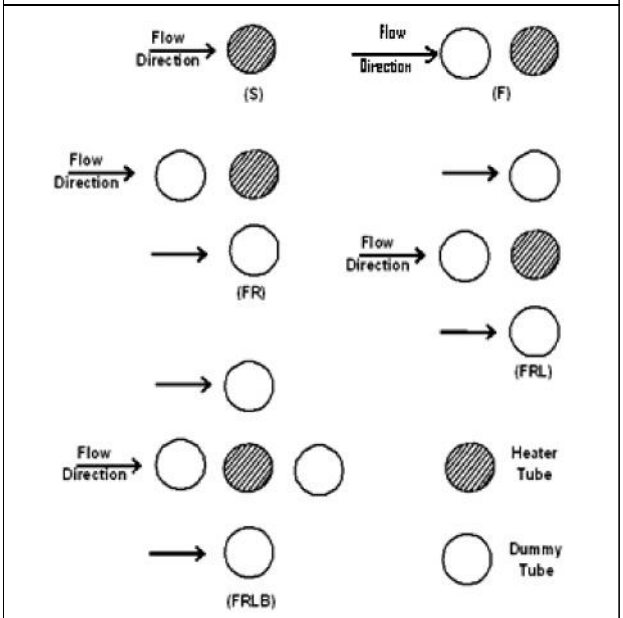
h_x represents the local heat transfer at any position x and

h_{av} represents the average heat transfer coefficient.

EXPERIMENTAL

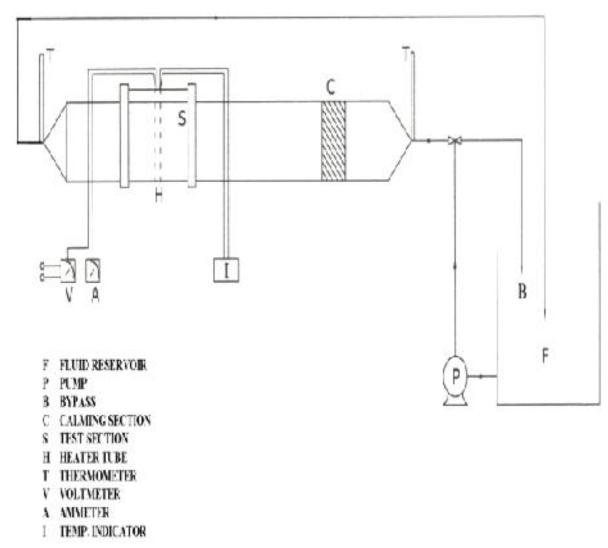
In the present experimental programme, the heat transfer data have been obtained by measuring the local surface temperature of heat flux probe and bulk inlet and outlet temperatures of the test fluid. The effect of neighbouring tubes on heat transfer was studied by placing dummy tubes around the heated probe as shown in Figure 2.

Figure 2: Arrangement of Dummy Tubes Around Heater Tube



Experiments were conducted using five different tube arrangements. In order to investigate the effect of tube diameter on heat transfer, different tubes of diameter 3.18, 2.54 and 1.91 cm have been used. A schematic diagram of the experimental setup is given in Figure 3.

Figure 3: Schematic Diagram of Experimental Set-Up



The test section of dimension 18 cm x 14.7 cm x 23.5 cm was fabricated with Perspex sheet. Three tubes of diameter 2.54 cm half exposed, were fixed on either side walls. The test section 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 tube against the pressure of flowing fluids. 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 fixed 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 required the central position in the test section. Arrangement was made to accomodate different sized tubes of three diameters that were studied in the present work. To accommodate the corresponding dummy tubes plastic adopters (plugs) were used in top and bottom plates of the test section.

The heat flux probe was fabricated from a Copper tube of 6 mm wall thickness. A pencil heater was inserted inside to fit in the tube tightly. Four slots were cut on the outer surface of the tube along its length at locations 0°, 90°, 180° and 270° at central axis of the tube, i.e., diametrically opposite to each other. Copper Constantan thermocouple wires, embedded in each slot, measured the local surface temperature of the heated tube which, when averaged, gave the average surface temperature of heated probe. A tube bank with longitudinal pitch 7.0 cm and transverse pitch 3.5 cm was used.

A capillary tube viscometer was used to measure the rheological properties of aqueous solutions of Carboxyl Methyl Cellulose and Poly Vinyl Alcohol while a parallel plate apparatus was used to measure the thermal conductivities of these fluids. Since aqueous CMC and PVA solutions followed Ostwald Power Law hence the values of flow behaviour index (n) and consistency constant (K) were determined by plotting wall shear stress ($^{2}_{w}$) versus shear rate on a log-log plot.

EXPERIMENTAL PROCEDURE

For a particular tube diameter and tube bank configuration the flow rate of a particular fluid was initially set to a predetermined value. A constant electrical flux was supplied to the heated probe. After about an interval of 15 minutes when steady state was achieved the temperature of four thermocouples, placed at four different locations on the probe was recorded with the help of a multi-channel digital temperature indicator. Simultaneously the inlet and outlet temperatures of the fluid were also noted. The mass flow rate of fluid was found by collecting the fluid for known interval of time. The flow rate was varied and all the above mentioned parameters were measured. Similar sets of observation were made using different fluids for each configuration and also for the whole tube tank consisting of thirty effective tubes of a selected diameter at a time. The experiment was conducted with three tubes of different diameters.

RESULTS AND DISCUSSION

In case of aqueous CMC solutions, it is observed that the flow behaviour index (n) decreases with increase in polymer concentration resulting in increased pseudo plastic behaviour. Also no change has been observed in flow behaviour index (n) with temperature variation in the range investigated in this work. However the value of consistency constant (K) decreases with increase in temperature. In case of aqueous PVA solution, parameter K remains constant irrespective of change in temperature for the range investigated. The evaluated values of consistency constant (K) and flow behaviour index (n) are given in Table 1.

Viscosity, density, thermal conductivity and specific heat for water have been taken from the literature (Brown and Marko, 1958).

Table 1: Experimental Value of Flow Behavior Index and Consistency Constant

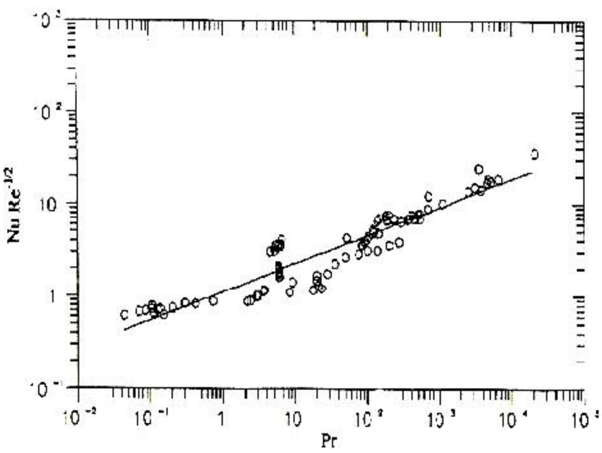
Fluid	n	Temperature °C	Consistency Constant Kg/m.sec$^{(2-n)}$
1% CMC	0.803	27.0	0.02287
1% CMC	0.803	33.0	0.02016
1% CMC	0.803	40.0	0.01836
1.5% CMC	0.637	22.5	0.15988
1.5% CMC	0.637	27.0	0.14465
1.5% CMC	0.637	31.0	0.13856
0.5% PVA	1.15	25.5-35.0	6.24 x 10^{-4}

The heat transfer is closely related to fluid dynamics, therefore the two phenomenon are considered simultaneously around the curved surface. These are complex processes and mainly depend upon the type of fluid and Reynolds number. A laminar boundary layer is formed on the portion of circular tube facing the flow with its thickness increasing downstream. There is also a longitudinal pressure gradient caused by 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 directions. The fluid assuming reverse flow contracts the boundary layer and changes to vortex which begins to rotate. The whole phenomenon of fluid behaviour is reflected with local heat transfer rate. The fluid dynamics and heat transfer 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 in local heat transfer. For a Newtonian fluid on the front part, upto separation of 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 decreases with the developing 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 dynamics and heat transfer phenomenon is more complex in nature.

To study the effect of Prandtl number on heat transfer a graph showing the variation of $NuRe^{-1/2}$ versus Pr has been plotted in Figure 4.

Figure 4: Effect of Prandtl Number on Heat Transfer from a Cylinder: Plot of $Nu.Re^{-1/2}$ versus Pr



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

$$\frac{Nu}{Re^{1/2}} = C\,Pr^{1/3}$$

The index 1/3, thus obtained for Prandtl number is in excellent agreement with theoretical analysis and also with the values reported by Richardson (1968) and Zukauskas

(1987) and others. The equation for the Nusselt number may be derived as:

$$Nu_{av} = \ddot{}\,(n, \,^{-})\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}$$

For $^{-} = {}^{®}\!/_{2}$, the above equation reduces to

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

where

$$c(n) = \ddot{}\,\left(n, {}^{®}\!/_{2}\right)$$

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

Table 2: Values of Factor c(n) for Different Experimental Fluids

Fluid	*c*(*n*)
0.5% PVA Solution	0.95
Water	0.97
1% CMC Solution	0.99
1.5% CMC Solution	1.01

Figure 5 through 8 show the variation of h_{av} with U_{av} for various tube arrangements- F, FR, FRL, FRLB for all the four test fluids studied in this work. Here it is found that the heat transfer coefficient is decreasing with increasing viscosity of fluids.

Figure 9 through 12 show plots of h_{av} versus U_{av} for all tube arrangements for the flow of 0.5% aqueous PVA solution (through 2.54 cm diameter tube), water and 1% and 1.5% aqueous CMC solutions respectively (passing through 3.18 cm diameter tube). It is seen from these plots except for water, that the tube arrangement has no significant effect on the value of h_{av} for a given flow velocity. This could be attributed to suppression of vortices in the wake by non-Newtonian fluids.

Figure 5: Variation of Average Heat Transfer Coefficient (h_{av}) with Average Velocity (U_{av}) for Heat Transfer from a Vertical Tube Placed in an Assembly of Tubes: Effect of Flow Behaviour Index

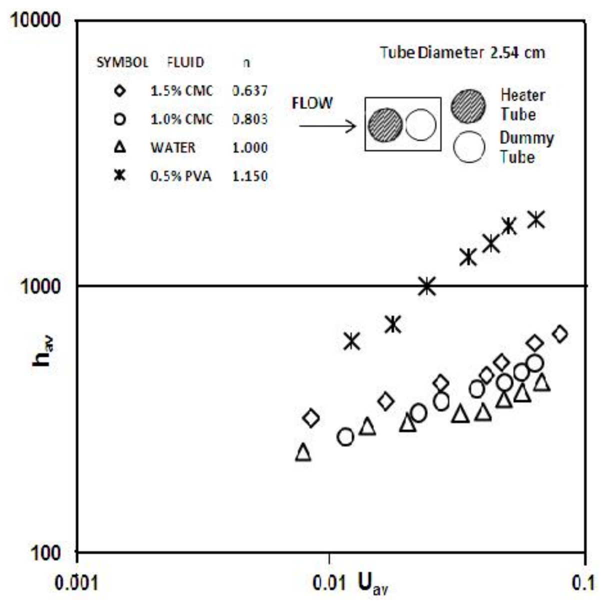


Figure 6: Variation of Average Heat Transfer Coefficient (h_{av}) with Average Velocity (U_{av}) for Heat Transfer from a Vertical Tube Placed in an Assembly of Tubes: Effect of Flow Behaviour Index

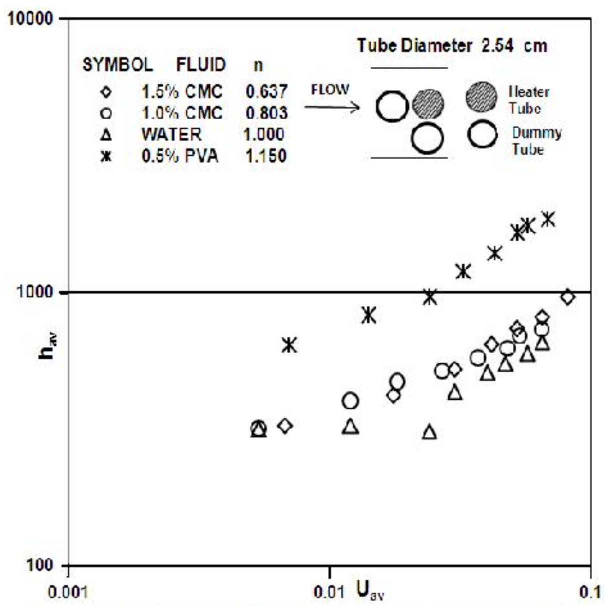


Figure 7: Variation of Average Heat Transfer Coefficient (h_{av}) with Average Velocity (U_{av}) for Heat Transfer from a Vertical Tube Placed in an Assembly of Tubes: Effect of Flow Behaviour Index

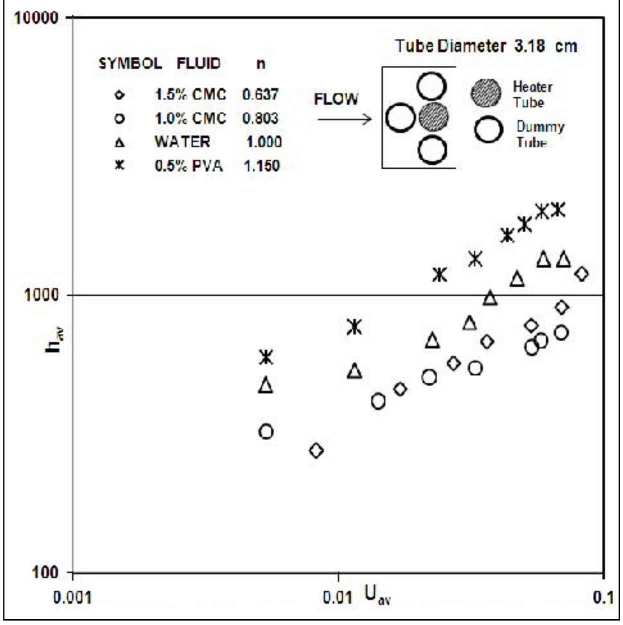


Figure 8: Variation of Average Heat Transfer Coefficient (h_{av}) with Average Velocity (U_{av}) for Heat Transfer from a Vertical Tube Placed in an Assembly of Tubes: Effect of Flow Behaviour Index

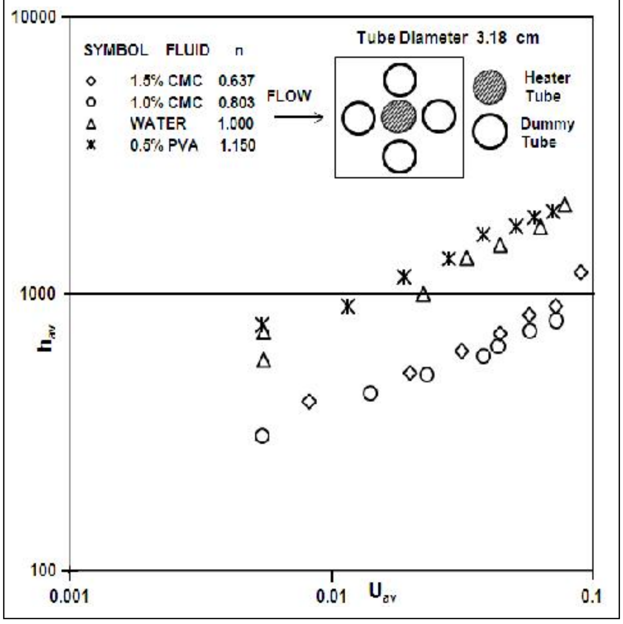


Figure 9: Variation of Average Heat Transfer Coefficient (h_{av}) with Average Velocity (U_{av}) for Heat Transfer from a Vertical Tube Placed in an Assembly of Tubes: Effect of Tube Arrangement (Tube Diameter 2.54 cm Fluid 0.5% PVA)

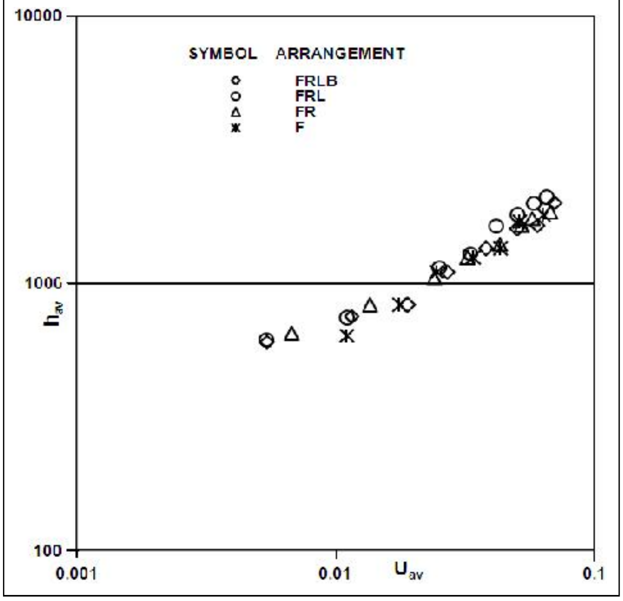


Figure 10: Variation of Average Heat Transfer Coefficient (h_{av}) with Average Velocity (U_{av}) for Heat Transfer from a Vertical Tube Placed in an Assembly of Tubes: Effect of Tube Arrangement (Tube Diameter 3.18 cm Fluid Water)

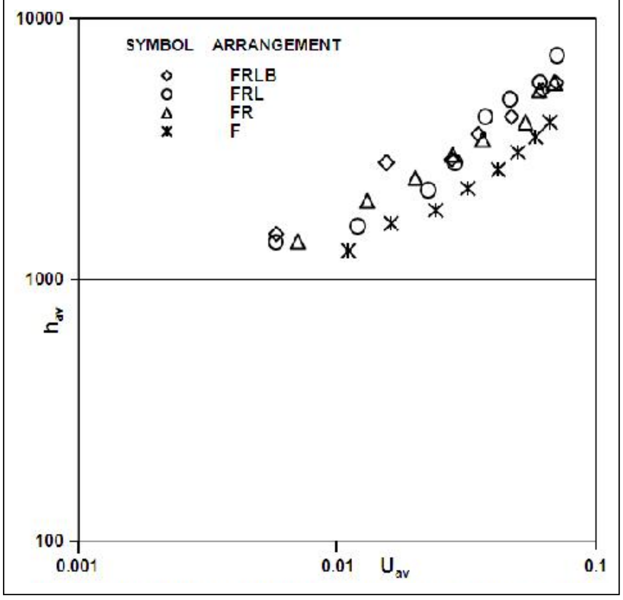


Figure 11: Variation of Average Heat Transfer Coefficient (h_{av}) with Average Velocity (U_{av}) for Heat Transfer from a Vertical Tube Placed in an Assembly of Tubes: Effect of Tube Arrangement (Tube Diameter 2.54 cm Fluid 1.0% CMC)

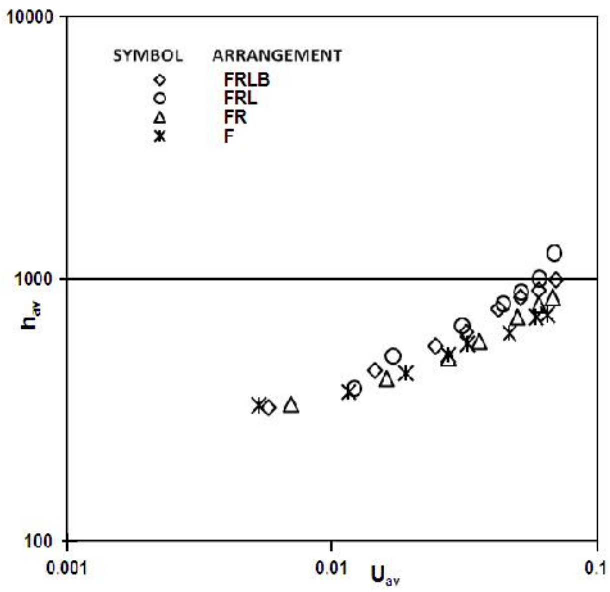


Figure 12: Variation of Average Heat Transfer Coefficient (h_{av}) with Average Velocity (U_{av}) for Heat Transfer from a Vertical Tube Placed in an Assembly of Tubes: Effect of Tube Arrangement (Tube Diameter 3.18 cm Fluid 1.5% CMC)

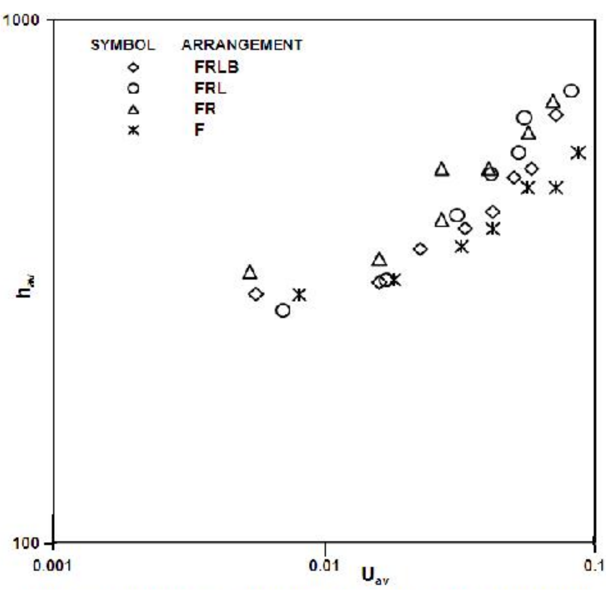
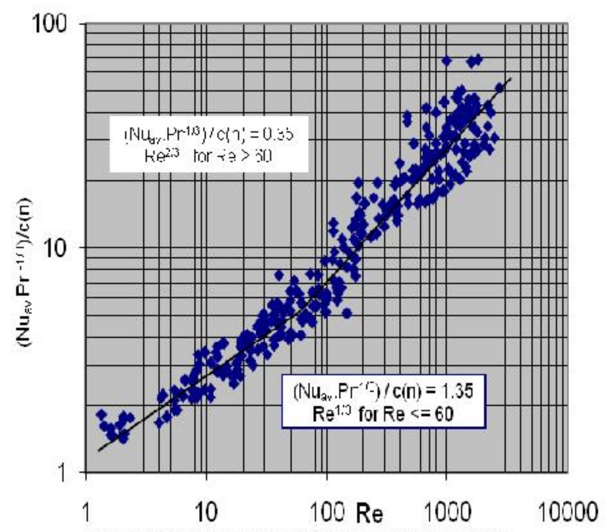


Figure 13: Heat Transfer During Flow Through Assembly of Tubes: Variation of $(Nu_{av}Pr^{-1/3})/c(n)$ with



CONCLUSION

The flow pattern around a tube in a tube bundle differs considerably from that around an isolated tube immersed in a fluid flowing across it as the former is influenced by the presence of neighbouring tubes. Also change in the boundary layer, formed around a cylinder is affected by neighbouring tubes which consequently results into considerable change in the heat transfer characteristics. Further, the turbulence generated by leading tube, in general, enhances the heat transfer and this change is more significant at higher Reynolds number. The effects on boundary layer and extent of turbulence, both are dependent on tube arrangement, spacing between adjacent tube and other geometrical parameters. For this purpose the experimental data for all the tube diameters, fluids and different arrangements viz. F, FR, FRL and FRLB have been plotted in Figure 13.

A careful examination of this plot reveals that experimental values show different trends for different range of Reynolds number. For lower Reynolds number (Re < 60) the slope is somewhat lower than for data in higher Reynolds number range (Re > 60). For Re ≤ 60 the slope of average line 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 following equations:

$$\frac{Nu\,\mathrm{Pr}^{1/3}}{c(n)} = 1.35\ \mathrm{Re}^{1/3} \quad \text{for Re} \leq 60$$

$$\frac{Nu\,\mathrm{Pr}^{1/3}}{c(n)} = 0.35\ \mathrm{Re}^{2/3} \quad \text{for Re} > 60$$

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APPENDIX

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
$c(n)$	$ø(n$, ®/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
SL	longitudinal pitch or spacing of longitudinal rows, m
S_T	transverse pitch, m
U	velocity, m/sec
$\bar{U}$	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

APPENDIX (CONT.)

Nomenclature	
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	
~	thermal diffusivity
£	void fraction calculated from tube spacing and geometry
«	coefficient of viscosity, Kg/m sec
«$_{eff}$	effective viscosity, Kg/m sec
–	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