Experimental Investigaion Of Natural Convection Heat Transfer Through Heated Vertical Tubes

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ABSTRACT

An experimental investigation of natural convection heat transfer in heated vertical ducts dissipating heat from the internal surface is presented. The ducts are open-ended and circular in cross section. The test section is electrically heated imposing the circumferentially and axially constant wall heat flux. Heat transfer experiment is carried out for four different channels of 45mm. internal diameter and 3.8mm thickness with length varying from 450mm to 850mm. Ratios of length to diameter of the channel are taken as L/D = 10, 12.22, 15.56, and 18.89. Wall heat fluxes are maintained at $q^{//}$ = 250 to 3341 W/m^2. The studies are also carried out on intensified channels of the same geometrical sizes with the discrete rings of rectangular section. Thickness of the rings are taken as t = 2.0, 3.0, 3.8mm; step size (s) of the rings are varying from 75mm to 283.3mm and the other ratios are taken as: ratio s/D = 1.67 to 6.29, ratio t/D = 0.044 to 0.084 and ratio s / t = 19.74 to 141.65. A systematic experimental database for the local steady state natural convection heat transfer behaviour is obtained. The effects of L/D ratio and wall heating condition on local steady-state heat transfer phenomena are studied. The effects of ring thickness and ring spacing on heat transfer behaviour are observed. The present experimental data is compared with the existing theoretical and experimental results for the cases of vertical smooth tubes and also for tubes with discrete rings.

Keywords- ***Heat flux,*** **heat transfer,** ***natural convection, ring spacing, ring thickness***

I. INTRODUCTION

Natural convection heat transfer has been a reliable, cost-effective cooling method for the fast growing electronic industry where hundreds of thermal connection modules are accommodated on a small base. As the density of these heat producing modules increases day by day, for more compactness, the heat released should be transferred from the surface not only to protect them but also for longer life. There is often the need to cool the internal surfaces of vertical open-ended ducts by natural convection, despite the low rates of heat transfer that this convection process affords. Thus information on the behaviour of natural convection flow through confined spaces has been found useful especially in the thermal fluid systems encountered in the diverse fields of nuclear and solar energy. Due to its importance, the natural convection problem has received increasing attention in the literature in recent years.

At present, flow of gaseous heat carriers in vertical channels with natural convection is widely encountered in science and engineering. For example, in domestic convectors, cooling systems of radio electronic and electrical equipment, nuclear reactors with passive cooling systems,dry cooling towers, ground thermo siphons, etc.

The purpose of this work is to study experimentally the natural convection pipe flows at different heating levels. The test section is a vertical, open-ended cylindrical pipe dissipating heat from the internal surface. The test section is electrically heated imposing the circumferentially and axially constant wall heat flux. As a result of the heat transfer to air from the internal surface of the pipe, the temperature of the air increases. The resulting density non-uniformity causes the air in the pipe to rise.

Heat transfer experiment is carried out for four different channels of 45mm internal diameter and 3.8mm thickness with length varying from 450mm to 850mm. Ratios of length to diameter of the channel are taken as L/D = 10, 12.22, 15.56, and 18.89. Wall heat fluxes are maintained at $q^{//}$ = 250 to 3341 W/m^2.

The studies are also carried out on intensified channels of the same geometrical sizes with the discrete rings of rectangular section. Thickness of the rings are taken as t = 2.0, 3.0, 3.8mm; step size (s) of the rings are varying from 75mm to 283.3mm and the other ratios are taken as: ratio s / D = 1.67 to 6.29, ratio t / D = 0.044 to 0.084 and ratio s / t = 19.74 to 141.65.

Although extensive work has been done on the study of natural convection hydrodynamics and heat exchange in vertical open-ended channels without intensifiers, but the works on internal heat transfer with presence of intensifiers are not adequate in literature. Investigations are still going on to determine the effects of various parameters on hydrodynamics and heat transfer coefficients.

Sastri and Mallik [1] studied experimentally the natural convection heat transfer over an array of staggered discrete vertical plates and found that the use of discrete vertical plates in lieu of continuous plates gives rise to enhancement of natural convection heat transfer. The highest local heat transfer values are encountered at the leading edge and least at the trailing edge of each plate for a particular temperature level and spacing. The highest value corresponds to the thinnest thermal boundary layer and as the thermal boundary layer starts growing from the leading edge of each plate, the heat transfer values starts decreasing and reach a minimum at the trailing edge. Had the plates been continuous, there would have been decrease in the heat transfer values continuously along the height of the vertical plate for same input conditions. They also found that the heat transfer quantities at the leading edge of the top plate are more than that at the trailing edge but less than that at the leading edge of the bottom plate. Degree of enhancement increases with increase in spacing.

Sparrow and Prakash [2], and Prakash and Sparrow [3] have analyzed the free convection from a staggered array of discrete vertical plates. They compared the performance of a staggered array of discrete vertical plates with that of a parallel flat channel, considering the wall at uniform temperature. Their results indicated that larger spacing, shorter plate and smaller heights of the channels provide enhancement of heat transfer. Anug et al. [4] attempted to derive a general expression to account for the effect of flow restriction, while still considering the governing equation to be parabolic. Flow restrictions encountered in Aung's study are in the form of staggered cards and baffles.

Kalinin et al. [7] found that the low cross ring lugs is rather effective with heat transfer enhancement in tubes with forced convection. It is experimentally revealed that in range of comparatively small Reynolds numbers and comparatively large relative steps of lugs a turbulence of results in a favorable ratio between heat transfer enhancement and increase of hydraulic resistance. The optimum height of lugs in a tube is in the range $0.1 > 2h/D > 0.02$ and the optimum steps is in the limits $25 \geq t/h \geq 10$; with increase of h/D the optimum moves to the range of large t/h, which is checked experimentally in the range of Reynolds numbers from 10^4 to 10^5.

Gortysov et al. [8] studied experimentally the natural convection hydrodynamics and heat exchange in vertical open-ended channels. They found that when discrete rings are provided in the internal surface of the tube there is increase in heat transfer from the channel walls.

Elenbass [9] carried out extensive analytical and experimental work on natural convection flow in such cross sectional geometries as the equilateral triangle, square, rectangle, circle and infinite parallel plates and his results are often compared with analytical results for those channels. Natural convection heat transfer measurements for vertical channels with isothermal walls of different temperature are presented in the work of Sernas et al. [10]. Experimental study of Sparrow and Bahrami [11] encompasses three types of hydrodynamic boundary conditions along the lateral edges of the channel. Akbari and Borges [12] solved numerically the two dimensional, laminar flow in the Trombe wall channel, while Tichy [13] solved the same problem using the unseen type approximation. Lavy et al. [14] address the problem of optimum plate spacing for laminar natural convection flow between two plates. Churchill, using the theoretical and experimental results obtained by a number of authors for the mean rate of heat transfer in laminar buoyancy-driven flow through vertical channels, developed general correlation equations for these results.

M. Capobianchi, A. Aziz [15] A scale analysis is presented for natural convection from the face of a vertical plate. Three types of thermal boundary conditions are considered: (1) constant surface temperature; (2) constant surface heat flux, and (3) plate heated from its back surface. Basant K. Jha, Abiodun O. Ajibade This article investigates the natural convection flow of viscous incompressible fluid in a channel formed by two infinite vertical parallel plates. Fully developed laminar flow is considered in a vertical channel with steady–periodic temperature regime on the boundaries. The effect of internal heating by viscous dissipation is taken into consideration. Separating the velocity and temperature fields into steady and periodic parts, the resulting second order ordinary differential equations are solved to obtain the expressions for velocity, and temperature.

M. Sankar et al [16]. In this paper natural convection flows in a vertical annulus filled with a fluid-saturated porous medium has been investigated when the inner wall is subject to discrete heating. The outer wall is maintained isothermally at a lower temperature, while the top and bottom walls, and the unheated portions of the inner wall are kept adiabatic. Through the Brinkman-extended Darcy equation, the relative importance of discrete heating on natural convection in the porous annulus is examined.

D.X. Du et al [17]. A mathematical separated flow model of annular upward flow has been developed to predict the critical heat flux (CHF) in uniformly heated vertical narrow rectangular channels. The theoretical model is based on fundamental conservation principles: the mass, momentum, and energy conservation equation of the liquid film and the momentum conservation equation of the vapor core together with a set of closure relationships.

Bum-Jin Chung et al. [18] Natural convection experiments inside a vertical cylindrical cavity were performed for Rayleigh numbers and for four different geometrical arrangements: both-open (pipe-shape), bottom- closed (cup-shape), top-closed (cap), and both-closed (cavity) cylinders.

Subhrajit Dey , Debapriya Chakrborty Tailoring [19] The local flow field around a fin can substantially enhance the forced convection heat transfer from a conventional heat sink. A fin is set into oscillation leading to rupture of the thermal boundary layer developed on either side of the fin. This enhancement in heat transfer is demonstrated through an increase in the time-averaged Nusselt Number (Nu) on the fin surfaces.

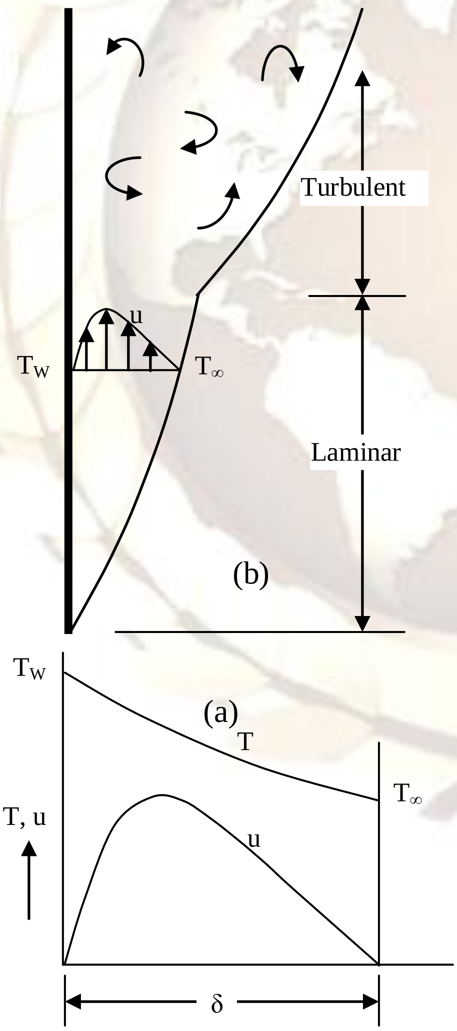


Fig. 1.(a) Boundary layer on a vertical flat plate (b) Velocity and Temperature distribution in the boundary layer

Approximately 4×10^8. Values ranging between 10^8 and 10^9 may are observed for different fluids and environment "turbulence levels".
Over the years it has been found that average free-convection heat transfer coefficients can be represented in the following functional form for a variety of circumstances:

$$\overline{Nu_f} = C\left(Gr_f \ \Pr_f\right)^m \quad (1)$$

$$T_f == \frac{T_w + T_\infty}{2} \quad (2)$$

The product of the Grashof and Prandtl is called Rayleigh number:

Ra= Gr Pr (3)

The characteristic dimensions used in the Nusselt and Grashof numbers depend on the geometry of the problem. For a vertical plate it is the height of the plate L; for a horizontal cylinder it is the diameter d; and so forth. Experimental data for free convection problems appear in a number of references, which may be given in a summery form as follows;

For vertical surfaces, the Nusselt and Grashof numbers are formed with L, the height of the surface as the characteristic dimension. If the boundary layer thickness is not large compared to the diameter of the cylinder, the heat transfer may be calculated with the same relations used for vertical plates. The general criterion is that a vertical cylinder may be treated as a vertical flat plate when,

$$\frac{D}{L} \geq \frac{35}{Gr_L^{1/4}} \quad (4)$$

where D is the diameter of the cylinder. For isothermal surfaces, the values of the constants are given in [39] with appropriate references noted. There are some indications from the analytical work of various investigators that the following relation may be preferable.

$$Nu_f = 0.10\left(Gr_f \ \Pr_f\right)^{1/3} \quad (5)$$

More complete relations have been provided by Churchill and Chu and are applicable over wider ranges of the Rayleigh number :

$$\overline{Nu} = 0.68 + \frac{0.670 Ra^{1/4}}{\left[1+\left(0.492/\Pr\right)^{9/16}\right]^{4/9}} \quad for\ Ra_L < 10^9 \quad (6)$$

$$\overline{Nu}^{1/2} = 0.825 + \frac{0.387 Ra^{1/6}}{\left[1+\left(0.492/\Pr\right)^{9/16}\right]^{8/27}} \quad for\ 10^{-1} < Ra_L < 10^{12} \quad (7)$$

Equation is also a satisfactory representation for constant heat flux. Properties for these equations are evaluated at the film temperature.

Extensive experiments have been reported by Vliet [39] for free convection from vertical and inclined surfaces to water under constant heat-flux conditions. In such experiments, the results are presented in terms of a modified Grashof number, Gr^* :

$$Gr_x^* = Gr_x\, Nu_x = \frac{g\beta q_w x^4}{k\upsilon^2} \qquad (8)$$

Where q_w is the wall heat flux in watts per square meter. The local heat transfer coefficients are correlated by the following relation for the laminar range:

$$Nu_{xf} = \frac{hx}{k_f} = 0.60\left(Gr_x^* \Pr_f\right)^{1/5} \quad for\, 10^5 < G_x^* < 10^{11}; \qquad (9)$$

For the turbulent region, the local heat-transfer coefficients are correlated with

$$Nu_x = 0.17\left(Gr_x^* \Pr\right)^{1/4} \quad for\, 2\times10^3 < Gr_x^* \Pr < 10^{16}; \qquad (10)$$

All properties in the above correlation are evaluated at the local film temperature. Although these experiments were conducted for water, the resulting correlations are shown to work for air as well. Writing Eq. (27) as a local heat transfer form gives.

$$Nu_x = C\left(Gr_x \Pr\right)^m \qquad (11)$$

Substituting the value of Gr_x :

$$Nu_x = C^{1/(1+m)}\left(Gr_x^* \Pr\right)^{m/(1+m)} \qquad (12)$$

Churchill and Chu show that Eq. may be modified to apply to the constant heatflux case if the average Nusselt number is based on the wall heat flux and the temperature difference at the center of the plate (x = L/2). The result is

$$\overline{Nu}_L^{1/4}\left[\overline{Nu}_L - 0.68\right] = \frac{0.67\left(Gr_L^*\right)}{\left[1+\left(0.492/\Pr\right)^{9/16}\right]^{4/9}} \qquad (13)$$

Where

$$\overline{Nu}_L = q_w L/\left(k\,\overline{\Delta T}\right) \quad and \quad \overline{\Delta T} = T_w\ (at\ L/2) - T_\infty$$

II. EXPERIMENTAL SET-UP AND PROCEDURE

Fig. 2, shows the overall experimental set-up, which consists of the entire apparatus and the main instruments. The experimental set-up consists of a test section, an electrical circuit of heating and a measuring system. The test section is a cylindrical tube. The cross-sectional view of the test section is shown in fig. 3. In this study a hollow tube is made of alumunium which is 45mm in diameter and 3.8mm thick. Nine Copper-Constantan thermocouples are fixed to monitor temperatures on the internal surface at various locations as shown in the figure. Holes of 0.8 mm diameter are drilled at these locations for inserting the thermocouples. After inserting the thermocouple beads, the holes are filled with alumunium powder for getting good thermal contact with the tube. Then the opening of the thermocouple wells are closed by punching with a dot punch. Epoxy is used for sealing the opening of the thermocouple wells and for holding the thermocouples in position.

After mounting the thermocouples a layer of asbestos paste (10mm thick) is provided on the outer surface of the tube. A layer of glass tape is provided over the asbestos paste and then the nichrome wire heater coil is helically wound around the external surface with equal spacing. Then asbestos rope of diameter approximately 7mm is wound over the heater coil with close fitting.

After that another layer of asbestos paste is provided. A layer of glass-fibers of approximately 15mm thickness is wrapped around it. A thick cotton cloth is wrapped over the glass fibers. It is covered with a very thin alumunium foil for reducing the radiation heat transfer. Two traversing type thermocouples are provided; one at the entrance and the other at the exit, to determine the temperature profile of fluid entering and leaving the tube at different radial distances. Another;

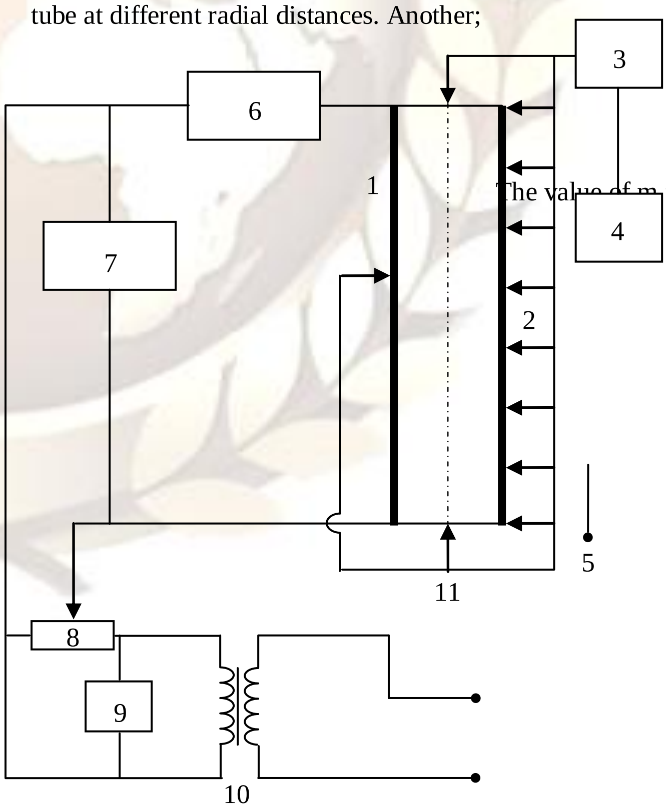


Fig. 2. Experimental Set-up

1. Test section, 2. System of thermocouples, 3. Selector switch, 4. Millivoltmeter, 5. Thermometer, 6. Ammeter, 7 & 9. Volt meter, 8. Variac, 10. Transformer, 11. Traversing type thermocouples

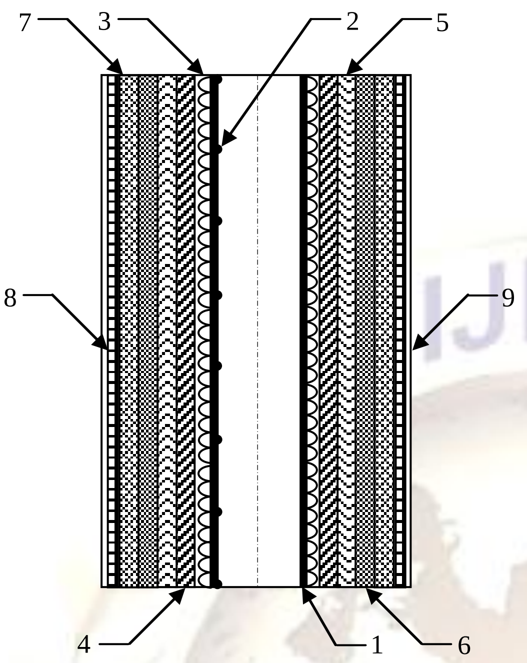


Fig. 3. Cross sectional view of test section

1. Aluminium tube, 2. Position of thermocouples, 3. Heater coil, 4. Glass tape, 5. Asbestos rope, 6. Asbestos paste, 7. Glass fibre, 8. Cotton cloth, 9. Aluminium foil.

Traversing type thermocouple is also used for measuring the external surface temperature of the test section to find out the heat loss from the external surface.

Four different channels of similar construction with height varying from 450mm to 850mm are considered. Ratio of length to diameter of the channel are taken as L/D = 10, 12.2, 15.6, and 18.9. Rings of rectangular section of thickness varying from 2mm to 3.8mm are used.

Wall temperatures at different locations are found out from the millivoltmeter readings to which thermocouples are connected. The fluid temperatures at the channel exit and entrance are found out at various radial distances by two traversing type thermocouples provided at the top and bottom of the channel respectively. The electric power input to the test section is determined from the measured voltage drop across the test section and the current along the test section.

Though adequate thermal insulation is provided on the outer surface of the tube, there is still some heat rejection from the external surface and this heat loss by natural convection from the test section through the insulation is evaluated by measuring the outer surface temperature of the insulation and the ambient temperature. At different axial locations along the pipe the outer surface temperature of the insulator are measured by thermocouples and the average insulation temperature is determined. Heat loss from the external surface is then computed by the suggtested correlation for natural convection from a vertical cylinder in air [39]. Now heat dissipated from the internal surface can be found out by subtracting this heat loss from the heat input to the test section. The wall heat flux $q^{//}$ can be found out by dividing this heat by the internal surface area. From the measured temperature profiles at the channel entrance and exit, the approximate temperature profile at any other axial distance can be calculated.

Let the local wall temperatures at different axial distance along the pipe be T_{w1}, T_{w2}, T_{w3}, …..etc, and the fluid temperature at these distances be T_{b1}, T_{b2}, T_{b3}, …..etc. respectively, then local heat transfer coefficient at these locations can be calculated by the relation:

$$h_1 = \frac{q''}{(T_{w1} - T_{b1})} \tag{14}$$

$$\overline{Nu} = \frac{\bar{h}D}{k} \tag{15}$$

Now modified Rayleigh number based on constant heat flux and average wall temperature can be calculated by using the following two formula:

$$Ra^{\#} = Gr\,\Pr\frac{D}{L} = \frac{g\beta q^{//}D^5}{\alpha\upsilon kL} \tag{16}$$

$$Ra^{\#} = \frac{g\rho^2 c_p \beta\left(\overline{T_w} - \overline{T_b}\right)D^4}{Lk\mu} \tag{17}$$

A vane type anemometer is used to measure the velocity of fluid at the exit of the channel. The modified Reynolds number can be calculated by using the relation:

$$Ra^{\#} = \frac{\rho\bar{u}D^2}{\mu L} \tag{18}$$

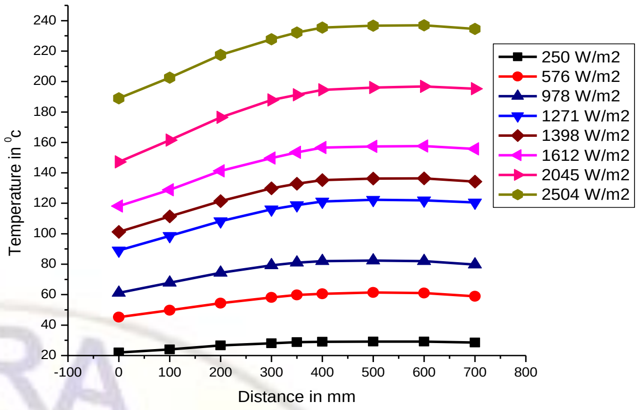
Measured temperature profiles at the channel exit for various heat fluxes and for different channel length are plotted.

A correlation is found for average Nusselt number and modified Rayleigh number. And also another correlation is found for modified Rayleigh number and modified Reynolds number.

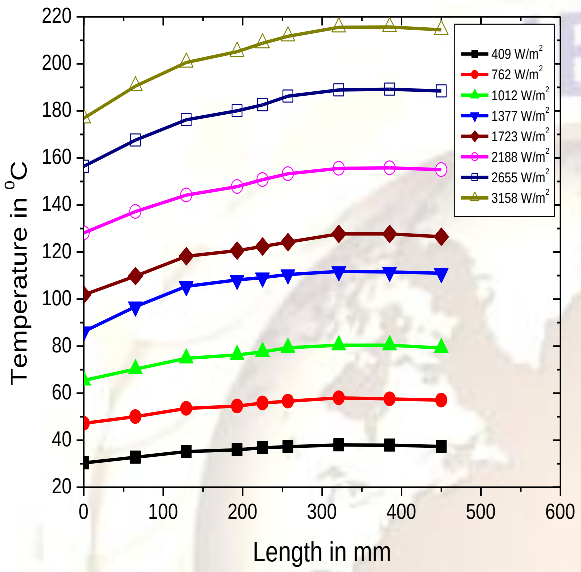
Similarly studies are also carried out on intensified channels of the same geometrical sizes with providing the discrete rings of height t = 2.0, 3.0, 3.8mm; step size (s) of the rings are varying from75mm to 283.3mm and the different non-dimensional ratios are taken as : ratio s / D = 1.67 to 6.29, ratio t / D = 0.044 to 0.084 and ratio s / t =19.74 to 141.65.

III. RESULTS AND DISCUSSION

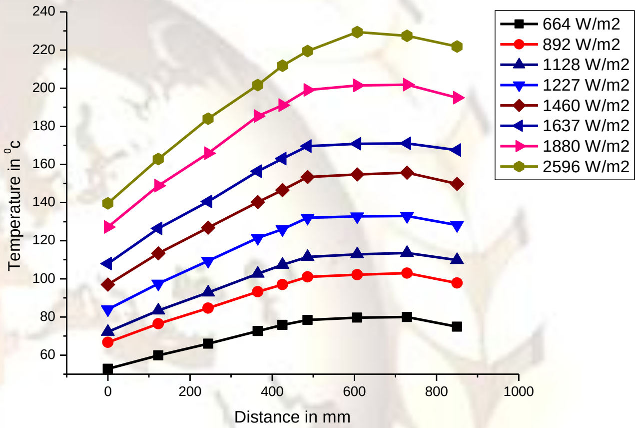
Typical axial variations of local wall temperatures for various L/D ratio and for various heat fluxes are shown in plotted in fig. 4(a) to 4(d) for smooth tubes. It increases along the height of the cylinder, which is in accordance with the theoretical predictions done by various investigators. But it slightly decreases towards the end, which may be due to the heat rejection from the end of the tubes.



(c)

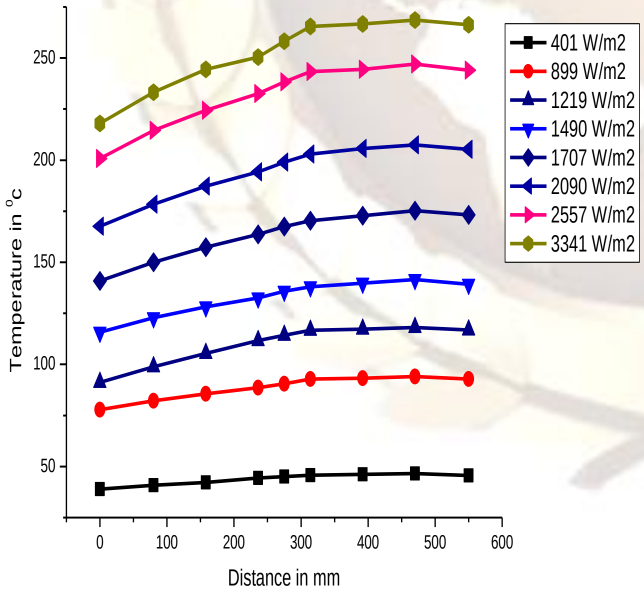


(a)

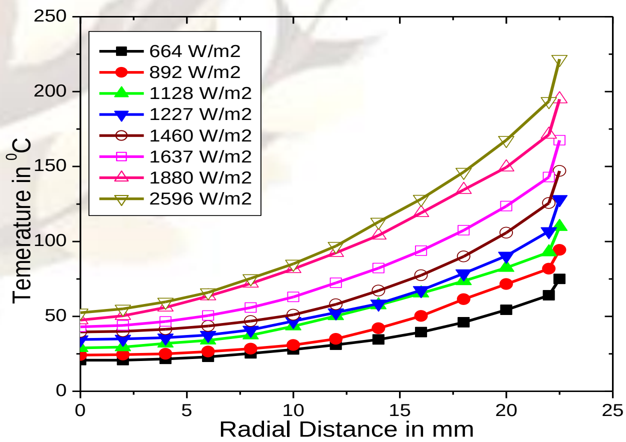


(d)

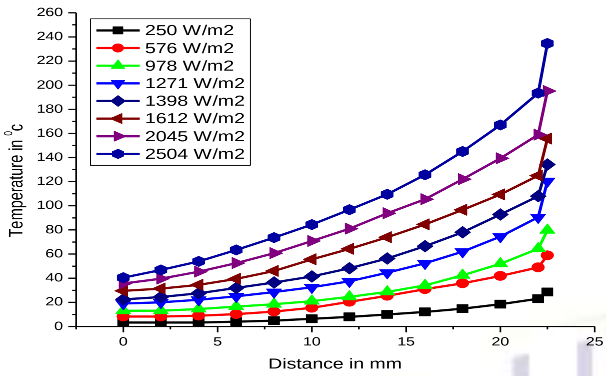
Fig. 4. Variation of wall temperature for different heat fluxes



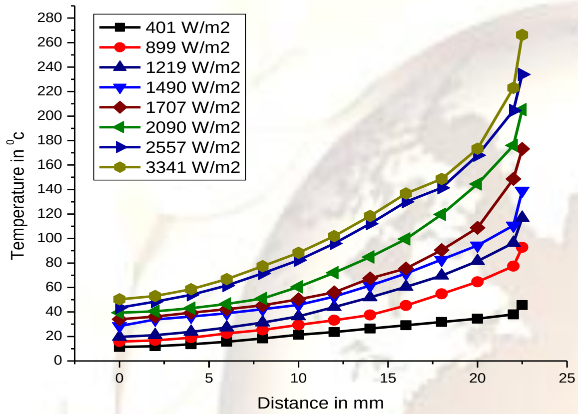
(b)



5(a)



(b)



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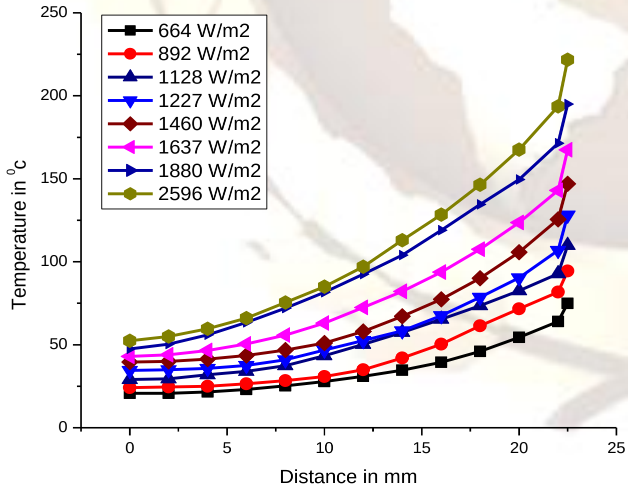

(d)

Fig. 5. Temperature profile at channel exit for different wall heat fluxes for smooth tube

Experimental temperature profiles at the channel exit for different heat fluxes and for different L/D ratio for smooth tubes are indicated in fig. 5(a), (b), (c), (d).

The fig. 6 shows the distribution of temperatures of flow at outlet of channels for various L/D ratio, i.e; for different length of the tube. It is clear from the graph that heat transfer increases with increase in tube length.

The fig. 7 and 8 shows the distribution of temperatures of flow at outlet of channels for various heat fluxes and various thickness and spacing of rings inside the tube. One can see that the average heat transfer rate increases with increasing the thickness of the rings up to a certain limits, beyond which it decreases, which is shown by the fig. 8.

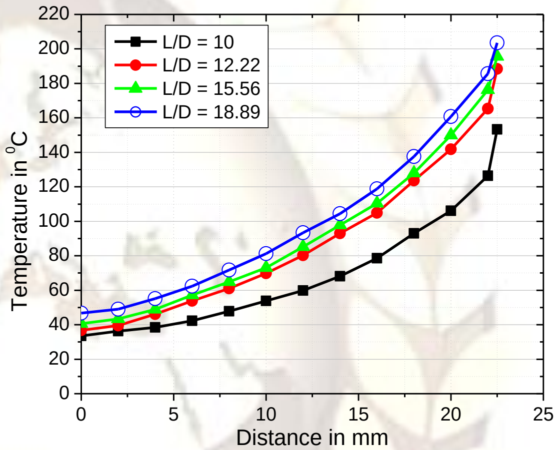

Fig. 6. Temperature profiles at the channel exit for smooth tubes with different length

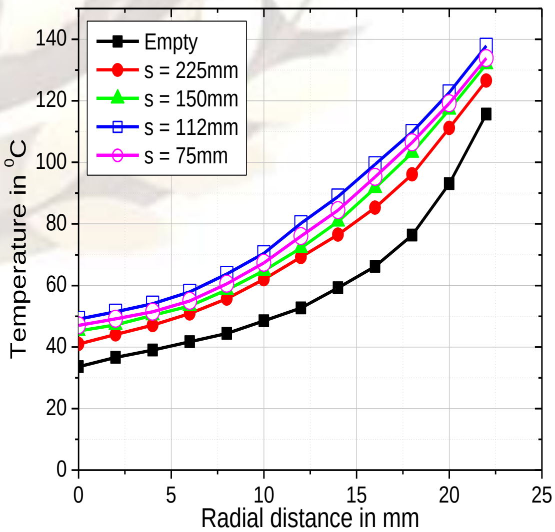

Fig. 7. Temperature profiles at the channel exit for tubes with discrete rings

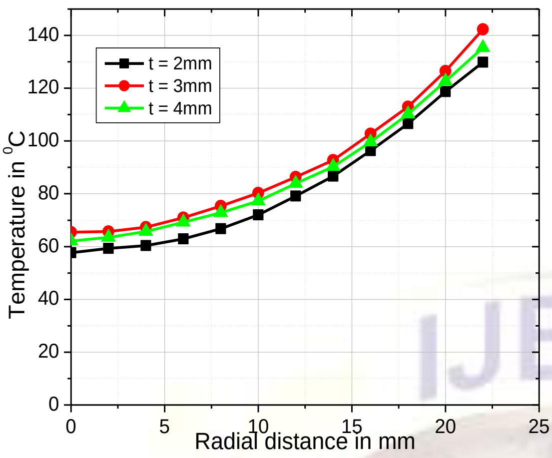

Fig. 8. Temperature profiles at the channel exit for different ring spacing

Fig. 7, shows that when the spacing between the rings decreases beyond a certain limit, the heat transfer rate decreases. This may be due to the fact that with an often arrangement of rings the pulsation arising behind the intensifier will have no time to fade sufficiently on the way to the following intensifier and will defuse to the flow core. Thus intensity of pulsation will increase and consequently the turbulence of flow will occur. This results in significant growth of hydraulic resistance with small increase of heat transfer. As hot layers diffuse to the center of flow, and the cold ones move to wall area, the gradient of temperature in the boundary layer decreases, and thus heat transfer decreases.

IV. CONCLUSIONS

The natural convection heat transfer in a vertical pipe has been studied experimentally, both for smooth tubes and for tubes with discrete rings. The effects of channel length, imposed wall heat flux and also the effects of ring thickness and ring spacing on the characteristics of natural convection heat transfer are examined in detail. The following conclusions can be drawn from the present investigation.

i) Average heat transfer rate from the internal surfaces of a heated vertical pipe increases with providing discrete rings.
ii) Average heat transfer rate increases with increasing the thickness of the rings up to a certain limits, beyond which it decreases.
iii) Average heat transfer rate increases with increasing the number of rings i.e. reducing the spacing between the rings up to a certain value of spacing, but further reduction in spacing, reduces the heat transfer rate from the internal wall to air.

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