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Theoritical Investigaion Of Convection Heat Transfer In Vertical Tubes

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ABSTRACT
Theoretical 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 60 mm. internal diameter and 5mm thickness with length varying from 450mm to 850mm. Ratios of length to diameter of the channel are taken as L/D = 11, 13.11. Wall heat fluxes are maintained at $q^{//}$ = 350 to 3340 W/m^2. The studies are also carried out on intensified channels of the same geometrical sizes with the discrete rings of rectangular section. 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.
Keywords- Heat flux, heat transfer, natural convection, ring spacing, ring thickness

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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 theoretically 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 theoretically is carried out for four different channels of 60 mm internal diameter and 5mm thickness with length varying from 450mm to 850mm. Ratios of length to diameter of the channel are taken as L/D = 11, 13.11. Wall heat fluxes are maintained at $q^{//}$ = 350 to 3340 W/m^2.

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.

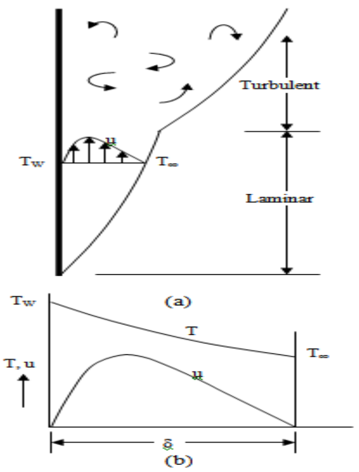


Fig. 1.(a) Boundary layer on a vertical flat plate (b) Velocity and Temperature distribution in the boundary layer

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:

$$\overline{Nu_f} = C\left(Gr_f \ \mathrm{Pr}_f\right)^m \tag{1}$$

where the subscript f indicates that the properties in the dimensionless groups are evaluated at the film temperature, which is given by:

$$T_f == \frac{T_w + T_\infty}{2} \tag{2}$$

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,

$$\frac{D}{L} \geq \frac{35}{Gr_L^{1/4}} \tag{4}$$

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.

$$Nu_f = 0.10\left(Gr_f \ \mathrm{Pr}_f\right)^{1/3} \tag{5}$$

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

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

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

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^* :

$$Gr_x^* = Gr_x\ Nu_x = \frac{g\beta q_w x^4}{k\upsilon^2} \tag{8}$$

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:

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

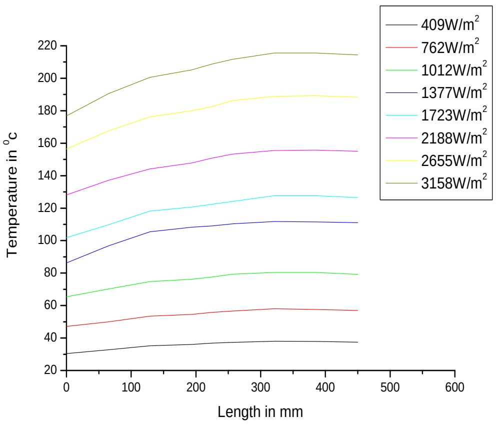
For the turbulent region, the local heat-transfer coefficients are correlated with

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

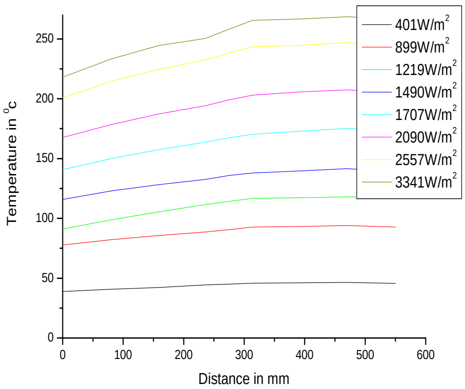
II. RESULTS AND DISCUSSION

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.



(a)



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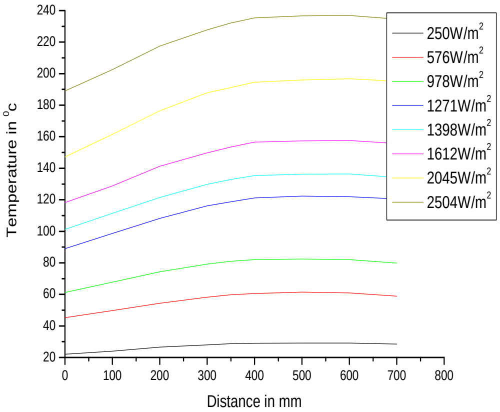
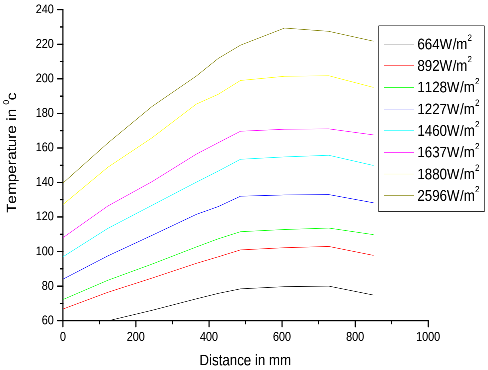
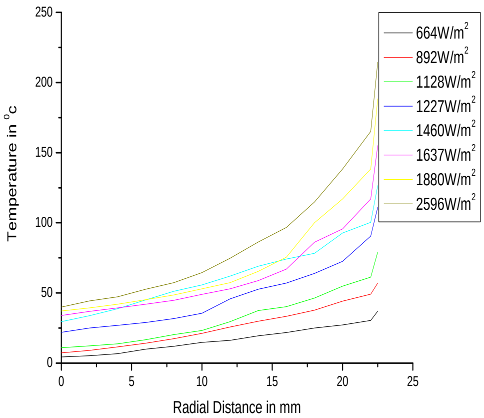


(c)

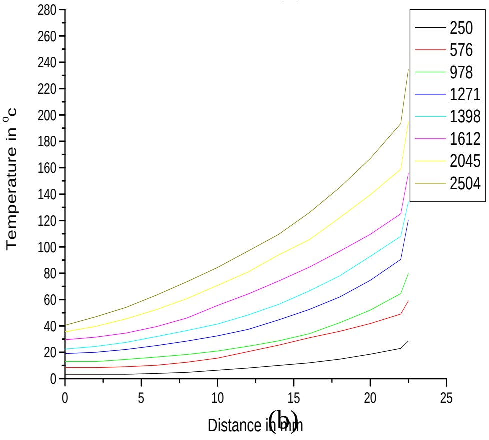


(d)

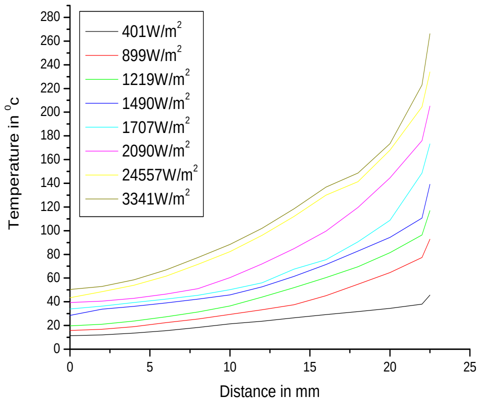
Fig. 4. Variation of wall temperature for different heat fluxes



5(a)



(b)



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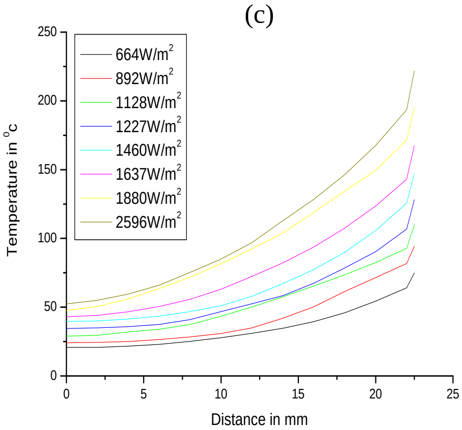

(d)

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

Theoretically 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.

III. CONCLUSIONS

The natural convection heat transfer in a vertical pipe has been studied theoretically, 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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