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# Natural convection heat transfer inside an open vertical pipe: Influences of length, diameter and Prandtl number

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ABSTRACT

We investigated the influences of length, diameter and Prandtl number on the natural convection heat transfer inside open vertical pipes. Numerical calculations were performed using FLUENT 6.3 varying the pipe length from 0.2 m to 1.0 m, the diameter from 0.003 m to 0.03 m and the Prandtl number from 0.7 to 2014. The numerical scheme was validated with mass transfer experiments employed to achieve high buoyancy. In large diameter pipes or high Prandtl number conditions, where relatively thin thermal boundary layers formed near the wall, the $Nu_L$ values were similar to those observed on vertical plates. As the diameter decreased, the length increased, and the Prandtl number decreased, the heat transfer was impaired due to the interactions of thicker thermal boundary layers. The duct flow and chimney effects were analyzed and visualized using the velocity and temperature fields.



## 1. Introduction

Passive cooling systems (PCS's) driven by natural forces have attracted research interests since the Fukushima nuclear power plant accident [1–3]. Many PCS's have been devised as the ultimate heat sink in nuclear power plants. A frequently used design for PCS heat exchangers comprises a vertical pipe geometry with natural convective coolant flow inside [4]. The inner and the outer thermal conditions of the vertical pipe have a significant influence on the cooling capability of the system. The natural convection heat transfer inside the vertical pipe varies with the length and the diameter of the pipe and the material properties of the working fluid. These variables affect the development of momentum and thermal boundary layers.

For a large diameter pipe, the boundary layers, which develop from opposite walls of the pipe, do not interact. Thus the heat transfer phenomena are similar to those observed on vertical flat plates. For a small diameter pipe, the boundary layers interact, merge and fully develop inside the pipe and thus the heat transfer is influenced by the duct diameter. The merged hot plume may be accelerated along the vertical pipe, making it act like a chimney. Due to the duct flow conditions, the bulk mass flow rate at every elevation should be the same.

Complex phenomena in vertical pipes need to be addressed. Most studies regarding heat transfer inside vertical pipes have focused on forced convection conditions [5–7]. Studies regarding natural convection heat transfer in vertical pipes are rare. Most of the research has been concerned with one or two aspects of the phenomena. This study explored the phenomena from various points of view such as development of boundary layers along the pipe, interaction of the boundary layers, duct flow and chimney effects and tried to envisage the complete picture.

In this study, we use both numerical and experimental methods to investigate the natural convection heat transfer inside vertical pipes. Numerical analysis was performed using FLUENT 6.3 [8]. In the numerical analysis, the lengths of the vertical pipes were varied from 0.2 m to 1.0 m, corresponding to a $Gr_L$ of $3.4 \times 10^8$ to $4.2 \times 10^{10}$. The diameters were varied from 0.003 m to 0.03 m and the Prandtl numbers used were 2,014, 20, 1 and 0.7. We validated the numerical schemes by replicating some of these cases experimentally. We took advantage of the analogy between heat and mass transfers. By performing mass transfer experiments we could achieve high buoyancy with relatively short facilities. A copper sulfate – sulfuric acid ($CuSO_4$–$H_2SO_4$) electroplating system was adopted as the mass transfer system.

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**Nomenclature**

| | |
|---|---|
| $C_b$ | Cupric ion concentration in the bulk [$mole/m^3$] |
| $D$ | Diameter of vertical pipe [$m$] |
| $D_m$ | Mass diffusivity [$m^2/s$] |
| $F$ | Faraday constant [96,485 $C/mole$] |
| $g$ | Gravitational acceleration [9.8 $m/s^2$] |
| $Gr_L$ | Grashof number ($g\beta \Delta TL^3/\nu^2$) |
| $Gr_x$ | Local Grashof number ($g\beta \Delta Tx^3/\nu^2$) |
| $h_h$ | Heat transfer coefficient [$W/m^2 \cdot K$] |
| $h_m$ | Mass transfer coefficient [$m/s$] |
| $I_{lim}$ | Limiting current density [$A/m^2$] |
| $k$ | Thermal conductivity [$W/m \cdot K$] |
| $L$ | Length of vertical pipe [$m$] |
| $n$ | Number of electrons in charge transfer reaction |
| $Nu_L$ | Nusselt number ($h_h L/k$) |
| $Nu_x$ | Local Nusselt number ($h_h x/k$) |
| $Pr$ | Prandtl number ($\nu/\alpha$) |
| $q''$ | Heat flux rate [W/$m^2$] |
| $Ra_L$ | Rayleigh number ($Gr_L Pr$) |
| $Sc$ | Schmidt number ($\nu/D_m$) |
| $Sh_L$ | Sherwood number [$h_m L/D_m$] |
| $t_n$ | Transference number |
| $T$ | Temperature [$K$] |
| $U_x$ | Uncertainty of $x$ |
| $y$ | Distance to near the wall [$m$] |
| $y^+$ | Dimensionless wall distance ($\rho\mu_\tau y/\mu$) |

*Greek symbols*

| | |
|---|---|
| $\alpha$ | Thermal diffusivity [$m^2/s$] |
| $\beta$ | Volume expansion coefficient [$1/K$] |
| $\mu$ | Viscosity [$kg/m \cdot s$] |
| $\mu_\tau$ | Shear viscosity [$kg/m \cdot s$] |
| $\nu$ | Kinematic viscosity [$m^2/s$] |
| $\rho$ | Density [$kg/m^3$] |

## 2. Preliminary analyses

### 2.1. Natural convection in a large diameter pipe

Natural convective flow in a heated vertical pipe is driven by buoyancy. The buoyancy force causes the upward flow and boundary layers to develop along the inner wall of the pipe from bottom to top. Their interactions are affected by the length and the diameter of the pipe [9]. Eckert and Diaguila [10] studied natural convection heat transfer in an open-ended pipe. A high $Gr_L$ was induced when a pipe of diameter 0.61 m and length 4.11 m was used. They reported that the natural convection heat transfer inside the pipe was similar to that observed on a vertical plate. Most studies have also concluded that natural convection heat transfer in large diameter pipes is similar to that on a vertical plate [10–13].

### 2.2. Duct flow

Davis and Perona [14] numerically investigated the natural convection heat transfer in a vertical pipe with a constant wall temperature and a $Pr$ of 0.7. Their results were similar to the experimental results reported by Elenbass [15]. In the case of a large diameter pipe, the velocity profile indicated a natural convective flow typical of a vertical plate, with the velocity peak near the wall. However in the case of a small diameter pipe, the velocity profile was parabolic, with a forced convective flow pattern through the whole pipe even though the flow is driven by buoyancy. Due to the duct flow condition, the mass flow rate at every elevation should be the same. The less buoyant flow velocity near the inlet and the more buoyant flow velocity near the outlet should have had the same mean velocity. Thus the velocity profile near inlet shows the characteristic of forced convection. This also causes the core acceleration [9].

### 2.3. Chimney flow

A chimney, which is an unheated extension of a flow passage, enhances the flow acceleration through buoyancy. It acts like a shroud so that the merged thermal plumes (Thermal boundary layers) inside the pipe are accelerated along the duct without scattering [16]. The height of the chimney determines the acceleration distance of the hot plume, and the heat transfer is enhanced as the flow rate increases [17]. Natural convection in chimney systems has been studied extensively [18–23]. Haaland and Sparrow [18] performed a numerical study of heat transfer in the case where heat sources are located at the bottom of the chimney. The upper, hotter accelerated fluid draws the lower, colder fluid. Similar acceleration is expected in the vertical pipe.

### 2.4. Prandtl number

The Prandtl number, which is the ratio of the kinematic viscosity to the thermal diffusivity, determines the relative thickness of momentum and thermal boundary layers. When $Pr > 1$, the momentum boundary layer is thicker than the thermal boundary layer and when $Pr < 1$, the reverse is observed [9]. Thus, in flows inside pipes, the $Pr$ determines the boundary layer that overlaps first. An overlap of the momentum boundary layer results in a duct flow effect whereas an overlap in the thermal boundary layer results in a chimney effect.

Chae and Chung [24] investigated natural convection heat transfer in a chimney both experimentally and numerically. They varied the length, diameter and $Pr$. $Pr$ was varied from 0.7 to 2014. The heat transfer rates were enhanced, with the chimney effect becoming more pronounced as $Pr$ decreased. Kageyama and Izumi [25] analytically and numerically investigated natural convection heat transfer in an infinitely long vertical pipe with a constant wall temperature. They showed that the temperature and velocity profiles developed differently with regard to the $Pr$. When $Pr > 1$, as the velocity field develops more rapidly. Thus, merged velocity fields near the inlet formed forced convective flow, which velocity peak appears at the center of the pipe, and then that flow last until the outlet. For $Pr < 1$, the temperature field develops more rapidly than the velocity ones. Forced convective flow appeared near the inlet due to the duct flow condition. As the flow proceeded, natural convective flow appeared where velocity peaks appeared near the wall. As the flow proceeded further, velocity profile develops into the forced convective flow again as the temperature field become fully developed.