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Fluid flow and heat transfer of mixed convection over heated horizontal plate placed in vertical downward flow

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ABSTRACT

Experimental investigations have been carried out on mixed convective flows induced around upward-facing, horizontal heated plates placed in a vertical downward flow of forced convection. The experiments were performed with water and air and in the ranges of the Reynolds and modified Rayleigh numbers, $300 < Re_W < 5000$, $10^7 < Ra_W^* < 10^{10}$ for water, and $200 < Re_W < 2000$, $10^6 < Ra_W^* < 10^9$ for air. A keen interest was directed to the thermal instability of the stagnation point flows. For the sake of this, the flow fields over plates are visualized with dye and smoke. The results showed that the longitudinal vortices of which axes are parallel to the flow direction appear over the plate when the buoyancy force is beyond critical, and that the onsets of the longitudinal vortices are predicted well with the Richardson number, $Gr_W/Re_W^2 = 2.45$ for both water and air. The local and overall heat transfer coefficients were also measured to investigate the influence of the longitudinal vortices on the heat transfer from the plates. The result showed that the coefficients are increased significantly from those of the stagnation point flows with the occurrence of the longitudinal vortices. It was also found that the overall Nusselt numbers of the plates are correlated well with the above Richardson number.

1. Introduction

When a horizontal plate is placed in a uniform downward flow of forced convection as is schematically illustrated in Fig. 1, a stagnation point flow or so-called Hiementz flow [1] appears over the plate. Such stagnation point flow is encountered in various heat transfer equipment and environmental situations. In particular, the flow attracts growing interest in the formation of thin films by a chemical vapor deposition (CVD) process in recent years [2]. This is because that the stagnation point flows can realize uniform heat and mass transfer rates throughout the base plate, and that they enable the uniform deposition of the films onto the surface. The example is a cold-wall type, thermal CVD reactor. In the reactor, a silicon wafer or a grass base plate heated at a certain elevated temperature is placed horizontally and a material gas is supplied from the top of the reactor vertically downward. The gas impinges onto the wafer or base plate to achieve the stagnation point flow, which enables the uniform deposition of the film. Although it is easy to obtain a stagnation point flow if the gas velocity is high enough, we should remind that the material gas is expensive, so that the velocity should be kept as low as possible to save the flow rate of the gas. While, when the gas velocity is low, it is anticipated that the buoyancy force induced around the heated base plate may exert unfavorable influence on the stagnation point flows. Because the flows will become unstable and highly distorted under the strong buoyancy force, and this will make the uniform deposition of the films difficult.

Despite the above circumstances, very little is known about the thermal instability of the stagnation point flows over heated horizontal plates. To the best of the authors' knowledge, we can cite few analytical works such as by Chen et al. [3] and Amaouche and Boukari [4], who have conducted a linear stability analysis on the laminar boundary layer of stagnation point flow. In those analyses, wave instability was assumed. However, these analyses will give little insight into the actual flow field over the heated plate. We can also refer some analytical works that have treated the mixed convection over heated horizontal plates to simulate a flow field in metal organic chemical vapor deposition (MOCVD) reactors. Calmidi and Mahajan [5] have obtained the numerical results on the mixed convective flows over a heated horizontal plate within a partially open vertical enclosure. Nobari and Beshkani [6] have also reported the analytical Nusselt numbers from the heated horizontal plate placed in partially open vertical channels having convergent or divergent cross-section. However, these analyses will provide limited information on the stability of the flow over the heated plates.

Taking account of these circumstances, the present authors have first carried out the intensive visualization experiments on the stagnation point flows over upward-facing, horizontal plates

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Nomenclature

C	proportional constant
C_p	specific heat at constant pressure
g	gravity acceleration
Gr_W	Grashof number, $g\beta(T_{wm}-T_\infty)\cdot W^3/\nu^2$
Gr_W^*	modified Grashof number, $g\beta q_w W^4/\lambda\nu^2$
h_m	overall heat transfer coefficient, $q_w/(T_{wm}-T_\infty)$
h_x	local heat transfer coefficient, $q_w/(T_{wx}-T_\infty)$
Nu_f	overall Nusselt number of stagnation point flow, $h_{mf}\cdot W/\lambda$
Nu_m	overall Nusselt number of mixed convective flow, $h_m\cdot W/\lambda$
Nu_x	local Nusselt number of stagnation point flow, $h_x\cdot x/\lambda$
Pr	Prandtl number, ν/α
q_w	surface heat flux by convection
Re_W	Reynolds number, $u_\infty\cdot W/\nu$
Re_x	local Reynolds number, $u_b\cdot x/\nu$
Ra_W	Rayleigh number, $Gr_W\cdot \mathrm{Pr}$
Ra_W^*	modified Rayleigh number, $Gr_W^*\cdot \mathrm{Pr}$
T	temperature
u_b	velocity at the outer edge of the boundary layer
u_∞	velocity of downward flow
W	plate width
x	distance from stagnation point

Greek symbols

α	thermal diffusivity
β	volume expansion coefficient
ε	emissivity
λ	thermal conductivity
σ	Stefan–Boltzmann constant
ν	kinematic viscosity
ρ	density

Subscripts

f	film or forced convection
m	mean or average
w	at wall
x	at location x

heated with uniform heat flux. The main concerns were directed to the onset of unstable flows due to buoyancy force. Both air and water were adopted as the test fluid to investigate the effect of Prandtl numbers on the flow instability. The local and overall heat transfer coefficients were also measured to ascertain the influences of unstable flows on the heat transfer from the plates. In order to make the flow configuration simple, the present experiments dealt with rectangular plates of high aspect ratios.

2. Experimental apparatus and measurements

Schematic illustrations of the present experimental apparatuses are given in Fig. 2 together with their dimensions. Fig. 2(a) and (b) shows vertical-type, low-speed, water and wind tunnels. The water pumped from a reservoir was fed into a settling chamber, and, then, was calmed by passing through meshes, honeycombs and a contraction nozzle as shown in Fig. 2(a). Then, the flow of small turbulence entered a test duct of $250\times250\,\mathrm{mm}^2$ cross-sectional area vertically and returned to the reservoir. While, the air in Fig. 2(b) entered a settling chamber of the wind tunnel, then, flowed into a test duct of $300\times300\,\mathrm{mm}^2$ cross-sectional area after going through meshes, honeycombs and a contraction nozzle, and, finally, exhausted to an experimental room by a blower. Rectangular test plates were placed horizontally with their heated side facing upward and at the location 150 mm downstream the nozzle outlet. A flow control valve adjusted the flow rate of water through the test duct. While, by controlling the fan speed with a voltage regulator, the flow rate of air was varied. A LDV measured mean and fluctuating velocities of the forced mainstream in the test duct at the cross-section of x = 150 mm, where the test plates are to be placed. The result showed that the mean velocities in the central portion of the duct were uniform with a deviation of less than ±3%, excluding the near wall regions of 20 mm and of 30 mm from the duct wall for water and air, respectively. The turbulent intensities of the mainstream were also found as less than 5–7% of the mean velocities for both water and air.

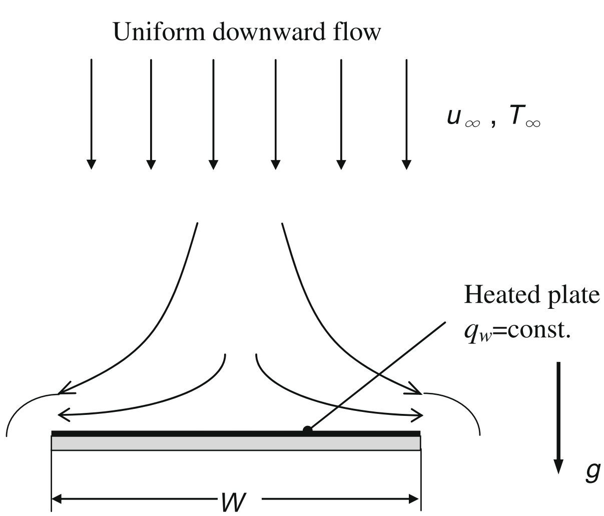

Fig. 1. Present configuration of flow.

Fig. 2(c) shows the test plate utilized in the present experiment. The length of the test plates was fixed as L = 240 and 290 mm for the experiments with water and air, respectively, while the width of the plates was varied as W = 50 and 75 mm for water and W = 50, 70 and 100 mm for air. Making use of the test plates with high aspect ratios, the flows over the plates can be supposed two-dimensional. The test plates consisted with an acrylic-resin base plate of 10 mm-thick and stainless steel foil heaters of 30 μm-thick. The heaters were glued on the front surface of the base plate and were heated with an ac power to attain a constant heat flux condition throughout the surface. While, a foam polystyrene insulation plate of 40 mm-thick was glued on the rear surface of the base plate to minimize the conduction heat loss. For the sake of the heat transfer experiments, Chromel–Alumel thermocouples of 100 μm in diameter were spot welded on the back of the foil heaters to measure local surface temperatures in the flow direction. In the experiments with air, a heat loss by radiation could not be neglected against the heat transfer by convection. Thus, we estimated the radiation heat loss based on the following equation.

$$q_{rad} = \varepsilon\sigma\left(T_{wx}^4 - T_\infty^4\right) \tag{1}$$

where q_{rad} stands for a heat flux by radiation from the surface of temperatures T_{wx} to the environmental air of temperature T_∞,