Pergamon

Int. J. Heat Mass Transfer. Vol. 39, No. 18, pp. 3977–3987, 1996

0017-9310/96 $15.00+0.00

0017–9310(95)00405–X

Penetrative flow due to an isothermal vertical wall in a stable two-layer environment generated by a room fire

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(*Received* 22 *February* 1995 *and in final form* 1 *November* 1995)

Abstract—An experimental investigation has been carried out on the penetrative natural convection flow and thermal transport resulting from an isothermal vertical surface immersed in a stably stratified, two-layer, ambient medium, in which an essentially isothermal heated layer overlies a relatively cooler isothermal layer of the same fluid. Measurements of the thermal field are carried out for several surface temperatures and the corresponding isotherms obtained. The surface temperature is taken as lower than the upper layer temperature so that a downward natural convection flow is generated adjacent to the surface in the upper zone and the penetration of this flow into the lower region is investigated. Velocity and temperature profiles are measured to determine the mass flow rates across the interface between the two regions. The local heat transfer rates are also measured at various locations on the isothermal plate. For the surface temperature lying between the upper and lower layer temperatures, an upward flow is generated in the lower region and a downward flow in the upper region. The two flows collide near the interface, giving rise to transport across the interface. Copyright © 1996 Elsevier Science Ltd.

INTRODUCTION

The natural convection flow arising from bodies submerged in thermally stratified media is of considerable importance in several heat rejection, energy storage and enclosure fire problems. The rejection of the thermal energy to the atmosphere and to water bodies as well as energy removal from heated bodies in enclosed regions often leads to the generation of a stable thermal stratification, as reviewed by Jaluria [1]. In thermal energy storage problems, such as those of interest in solar energy storage as sensible heat, stratification arises and is maintained in the storage tank due to the heat input, see Lavan and Thompson [2] and Jaluria and Gupta [3]. Similarly, at the early stages of fire growth in an enclosure, the fire plume hits the ceiling, spreads out and finally turns downward at the corners. Due to opposing buoyancy effects, the flow reverses its direction, moves upward and finally, a hot upper layer is established above the cooler lower layer, see, for instance, Zukoski [4] and Cooper *et al.* [5]. Such a two-layer, stratified circumstance is of interest in the present study.

Among the earliest studies on natural convection in thermally stratified media was that by Eichhorn [6], who obtained a series solution for the heat transfer from an isothermal vertical surface immersed in a thermally stratified medium with a linear vertical temperature variation. Cheesewright [7] obtained similarity solutions for laminar natural convection transport from a vertical surface to a thermally stratified medium. He calculated the heat transfer rates and the temperature and velocity profiles for a Prandtl number $Pr = 0.708$. However, in many cases, these solutions were physically unrealistic, since the ambient temperature decreases with height, leading to an unstable situation. Gebhart [8, 9] considered the conditions required for a similarity solution in the presence of thermal stratification and also established the conditions needed for a stable stratification in a compressible fluid. Yang *et al.* [10] presented similarity solutions for a nonisothermal vertical surface immersed in a stratified medium and calculated heat transfer rates and temperature profiles for several wall and ambient temperature distributions over a wide range of Pr. Piau [11] considered the case in which both the plate and the ambient temperatures varied as a power of the distance along the vertical plate. The experimental results compared well with the theory, but an arbitrary starting length had to be introduced in his analysis.

Jaluria and Gebhart [12] carried out an experimental and theoretical investigation to determine the effect of a stable ambient thermal stratification on the hydrodynamic stability of a buoyancy induced flow adjacent to a vertical surface dissipating a uniform heat flux. Disturbance growth rates, frequency and amplitude distribution across the boundary region were calculated for Prandtl numbers, Pr, of 6.7 and 0.733, for several levels of ambient stratification. The

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NOMENCLATURE

D	width of an equivalent jet discharged adjacent to the wall
g	magnitude of gravitational acceleration
Gr	Grashof number, $Gr = g\beta(T_u - T_l)H_1^3/\nu^2$, or $g\beta(T_{jet} - T_\infty)D^3/\nu^2$
h	convective heat transfer coefficient
H	height of the experimental enclosure, Fig. 2
H_1	depth of the interface from the enclosure top, Fig. 2
k	thermal conductivity of the fluid
l	height of the far wall of the enclosure used for varying the opening size, Fig. 2
Nu	Nusselt number based on H_1, $Nu = hH_1/k$
Pr	Prandtl number of the fluid
q	local surface heat transfer flux to the plate
Re	Reynolds number, $Re = U_{jet}D/\nu$
Ri	Richardson number, $Ri = g\beta(T_{jet} - T_\infty)D/U_{jet}^2$
u	vertical velocity component in the flow
U_c	convection velocity, $U_c = [g\beta H_1(T_u - T_l)]^{1/2}$
U_{jet}	average discharge velocity of equivalent wall jet
U_o	discharge velocity of heated fluid for stratifying the enclosure, Fig. 2
T	local temperature
T_{jet}	discharge temperature of equivalent wall jet
T_o	discharge temperature of heated fluid, Fig. 2
x, y	vertical and horizontal coordinate distances, respectively, Fig. 2
X, Y	dimensionless vertical and horizontal coordinate distances, respectively.

Greek symbols

α	thermal diffusivity of the fluid
β	coefficient of thermal expansion of the fluid
δ_p	penetration distance
$\Delta T, \Delta\hat{T}$	temperature difference, $\Delta T = T_u - T_l$, $\Delta\hat{T} = T_o - T_\infty$
ν	kinematic viscosity of the fluid
θ	dimensionless temperature, $\theta = (T - T_l)/(T_u - T_l)$
$\hat{\theta}$	dimensionless temperature, $\hat{\theta} = (T - T_\infty)/(T_o - T_\infty)$.

Subscripts

jet	equivalent wall jet
l	lower region
o	discharge location
s	surface
u	upper region
∞	ambient medium.

experimental results were in good agreement with the theory. The mean velocity and temperature distributions were also calculated, along with the heat transfer rates. Chen and Eichhorn [13] studied the natural convection heat transfer from an isothermal vertical plate to a stable, thermally stratified, environment. Analytical solutions to the problem were obtained by using the local nonsimilarity method. The analytical results were compared with experiments on this configuration as well as with those on a vertical copper cylinder, indicating good agreement. Jaluria and Himasekhar [14] carried out a numerical study on the vertical two-dimensional natural convection flow over a heated vertical surface and on a thermal plume immersed in a thermally stable medium. The temperature and velocity fields were computed and the effect of the thermal stratification on the flow and the heat transfer mechanisms investigated.

Some work has also been carried out on flows rising in an isothermal medium and penetrating into a stably stratified region, see Turner [15]. Mention may be made of the laboratory experiments of Deardorff *et al.* [16], who considered the unsteady penetrative convection in water to simulate the lifting of an atmospheric inversion above the heated ground. They obtained temperature and heat flux profiles. Further work on this problem was carried out by Deardorff *et al.* [17].

It is clear from the above brief review of the relevant literature that very little work has been done on the natural convection flow over an isothermal finite plate immersed in a stable, thermally stratified, two-layer environment, which gives rise to penetrative convection, as shown in Fig. 1 for two different circumstances. Here, $T_u > T_l$ in order to obtain a stable stratification [1]. In Fig. 1(a), the flow in the upper isothermal region penetrates into the lower region where the buoyancy force opposes the flow, restricting the depth of penetration. This figure shows a sketch of the flow that arises when the wall temperature T_s is lower than the upper layer temperature T_u and equal to the lower layer temperature T_l, giving rise to a downward wall flow in the upper layer. Figure 1(b) shows the flow when T_s lies between T_u and T_l. Here, two opposing flows arise in the two regions, collide near the interface and result in transport between the two layers. Similarly, other cases may be obtained for different values of T_s, T_u and T_l. In the present study,

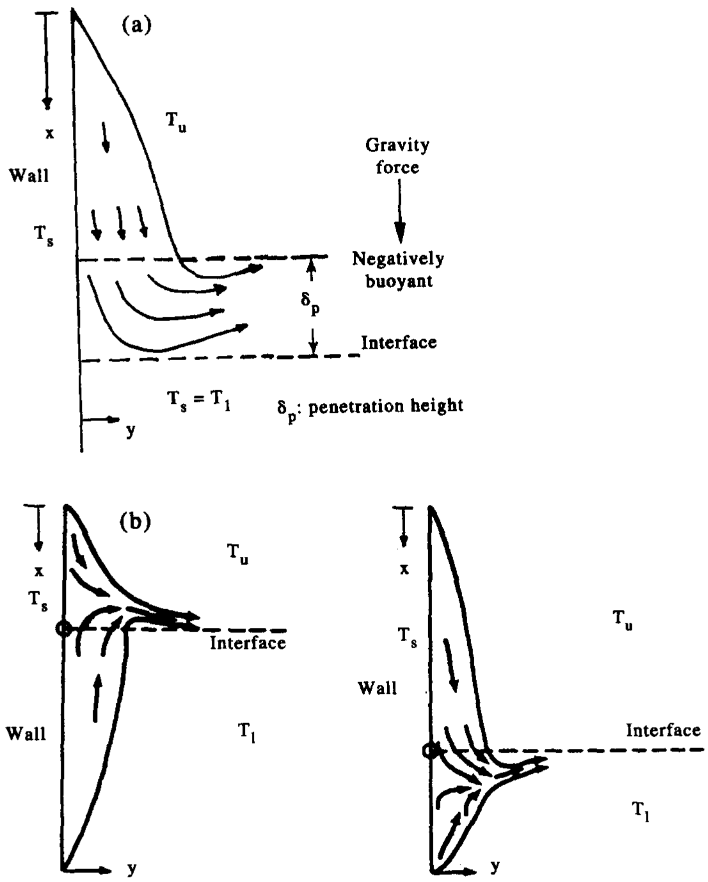

Fig. 1. Sketch of some of the penetrative natural convection flow circumstances considered. (a) $T_u > T_s = T_l$; (b) $T_u > T_s > T_l$.

T_u is always taken as greater than T_s and T_l, with T_l greater or smaller than T_s.

The penetrative convection at the interface of a two-layer thermally stable medium may be considered in terms of the discharge of a negatively buoyant flow into an isothermal medium, i.e. the circumstance where the buoyancy and the flow are in opposite directions, see Gebhart *et al.* [18]. Goldman and Jaluria [19] carried out a detailed experimental investigation to study the basic characteristic of a two-dimensional, negatively buoyant, jet in an isothermal environment. The penetration distance δ_p was measured and was related to the inflow conditions of the jet, in terms of the Richardson number Ri at the discharge location, where $Ri = Gr/Re^2$. Here, Gr and Re are the Grashof and Reynolds numbers, respectively, based on the average inlet temperature and velocity and the width D of the jet discharge slot. The heat transfer characteristics of such flows were studied by Kapoor and Jaluria [20]. The penetrative convection may also be compared to that of a wall jet in a thermally-stable, two-layer, environment. Kapoor and Jaluria [21] studied the penetrative convection of a plane turbulent wall jet in such an environment. For further details on such negatively buoyant flows in enclosure fires, see Jaluria and Cooper [22].

An experimental system was developed by Kapoor and Jaluria [23] to obtain a two-layer, thermally stable, environment in an enclosure, so that an essentially isothermal heated layer overlies a relatively cooler isothermal layer. A heated two-dimensional air flow was discharged downward into a glass enclosure at small velocity levels, compared to the wall flow velocities. The location of the interface between two layers could be controlled by varying the temperature and the velocity of the discharged air at the top of the enclosure. The stratification was shown to remain essentially unchanged with time. This experimental system is employed in the present work to generate the thermal stratification, as discussed in detail later.

In this paper, an experimental investigation is reported on the penetrative natural convection flow adjacent to an isothermal surface immersed in a two-layer, thermally stable, environment. The flow generated in one isothermal region penetrates into another region where it is negatively buoyant, as shown in Fig. 1. The thermal field is measured in detail near the isothermal plate for various wall and ambient temperatures. The velocity data at the interface between the two layers are obtained and the convective mass transfer rate penetrating the interface downward as well as that rising upward across the interface are determined. The local heat transfer rate to or from the plate has also been measured for several plate temperatures. The results obtained are considered in terms of the transport processes that govern this flow to bring out the basic characteristics of such penetrative flows.

EXPERIMENTAL ARRANGEMENT

Figure 2 shows a sketch of the experimental system used to generate a thermally stable, two-layer, environment in a glass enclosure. Heated air mass is discharged at very low velocity at the top of the enclosure, which is 1.37×0.3 m in cross-section and 1.5 m in height. A blower is used to send ambient air, over a fairly wide range of flow rates, through a copper duct which is 36 cm in length and 13.2×10.8 cm in cross-section. The copper duct is heated by means of three fiberglass-insulated strip heaters. The electrical energy input to each of the heaters can be varied by means of power controllers. A 10 cm thick honeycomb and three fine screens are used at the exit of the copper duct to ensure uniform temperature and velocity distributions across the duct. The copper duct leads into a diffuser which is designed to keep the total pressure loss at a minimum and the flow at the diffuser exit as uniform as possible. The diffuser has six guide vanes, i.e. seven dividing channels within the diffuser. Thus, the hot air flow from the copper duct is divided into seven streams, each flowing separately and merging with the others near the diffuser exit. This gives rise to a fairly uniform distribution of velocity and temperature along the length as well as the width of the diffuser, as confirmed by actual measurement of the velocity and temperature distributions. A very fine screen is placed just at the exit of the diffuser to further improve the uniformity of the flow and to keep the turbulence level low. The turbulence intensity was measured to be of order 5% at the exit.

The bottom of the enclosure is kept open to allow the ambient air to enter the lower part of the enclosure. The two sides of the enclosure are closed by glass walls

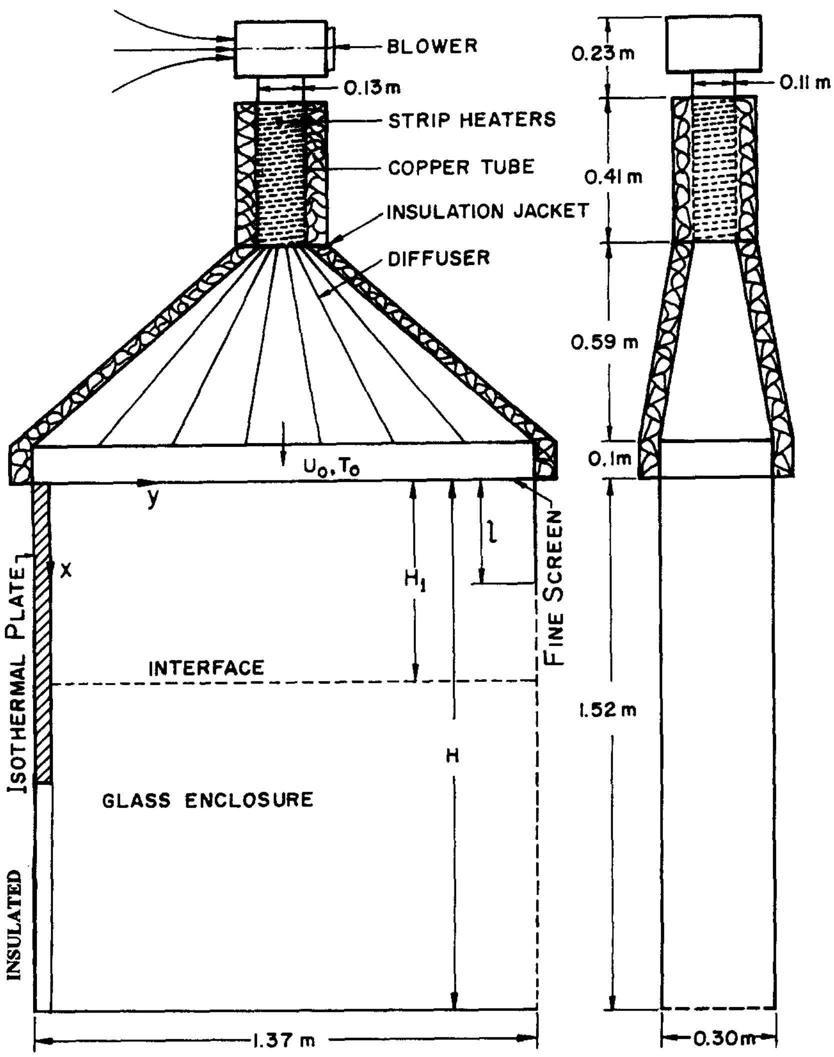

Fig. 2. Experimental arrangement for generating a stably stratified, two-layer, environment in the enclosure.

and the far end, away from the isothermal surface, was kept open to allow the hot air to flow out. The thermal energy loss from the copper tube and the diffuser to the environment is reduced by employing an insulation of fiber glass cloth tape and glass wool. Considerable care was needed to obtain a fairly uniform flow at the top, with a low turbulence level. For further details on the experimental system, see Kapoor and Jaluria [23].

The thermal field inside the enclosure, far from the isothermal surface, was measured by means of copper–constantan thermocouples made of 7.62×10^{-5} m diameter wires. At regular intervals, 20 thermocouple leads were attached to a side wall of the tank, with the thermocouple junction a few centimeters away from the wall. These thermocouples are used primarily to monitor the vertical air temperature distribution inside the enclosure. The accuracy of these measurements was of order 0.5°C.

A water cooled aluminum plate is located adjacent to the side wall of the enclosure. The plate is 91.4 cm high and 1.25 cm thick, extending over the entire width of the tank. The lower portion of the side wall, beyond the plate, was insulated. Four rectangular copper tubes, each 2.5 cm $\times$ 1.2 cm in cross-section, are located along the length of the plate. Water from an outside tank enters at the top of the plate and the circulated fluid is allowed to drain into a sink. The temperature of water entering the plate is maintained at a desired value by mixing hot and cold water streams from two separate hot and cold water tanks. Nine thermocouples, with an estimated accuracy of around 0.2°C, were located at the plate to monitor the plate temperatures. For details on the plate assembly, see Kapoor and Jaluria [20]. The plate was maintained at constant temperature. A fairly uniform temperature distribution was found to arise at the plate surface. The maximum temperature difference measured between any two surface thermocouples was less than 1.0°C.

The heat transfer to the plate was measured by means of microfoil heat flow sensors (RdF Type 20472-3). Each heat flow sensor had surface dimensions of 1.5×6 mm. These are very thin heat flow sensors (less than 1 mm in thickness) and can easily be attached to the plate surface. The electric output obtained in millivolts from the heat flow sensor was converted into heat flux (in W m^{-2}) with the help of individual calibration curves supplies by the manufacturer (RdF Corporation). These calibration curves were also checked against a known heat flux input to estimate the accuracy. In general, an accuracy of $\pm 5\%$ of the measured heat flux was estimated. Again, a considerable amount of effort was directed at obtaining this level of accuracy by providing good contact between the sensor and the surface and by choosing very thin and sensitive sensors.

The velocity distribution at the interface of the thermally stratified two-layer environment was obtained by using a DANTEC constant temperature hot-wire anemometer. The flow velocities at the interface were expected to be very small (of order 0.2 m^{-1}). Several calibration techniques are available for the high velocity range, see Jaluria [1] and Roshko [24]; but the calibration procedure for such low velocity levels is much more difficult. Due to buoyancy, the thermal plume from the hot-wire rises upward, whereas in the present experiment, the flow to be measured is either downward or upward, resulting in opposing and aiding flow situations, respectively. Since both the plume and the flow velocities are of the same order of magnitude [1], the calibration must be carried out in the same flow configuration as that encountered in the experiment.

A calibration facility was developed to calibrate the hot-wire anemometer for the desired velocity range and flow direction. The calibration facility consists of a slide on which the hot-wire is mounted. The slide is moved at a constant known velocity by a chain which moves between two sprockets with the help of an electric motor and a speed controller. The slide can be fixed at any angular position, from horizontal to vertical. During an experimental run, the output from the hot-wire anemometer is measured by means of a data acquisition system. The average voltage is obtained from this system for different velocities and a calibration curve is obtained. The calibration curves were obtained at different overheat ratios and this information was used to correct for the variation in the ambient fluid temperature. For further details on this calibration system, see Tewari and Jaluria [25].

The calibrated hot-wire sensors were used for velocity measurements in the flow field, particularly in the vicinity of the interface. For this steady state, two-dimensional problem, the two velocity components in the x- and y-directions (Fig. 2) were obtained by aligning the sensor normal to the flow direction of interest. Details of these measurements are not given here for conciseness and the earlier work, as referenced above, may be consulted for further information. The velocity levels arising due to the system for generating the thermal stratification were kept very low compared to the velocities in the wall flow, in order to study the penetration in an essentially quiescent environment.

EXPERIMENTAL RESULTS AND DISCUSSIONS

The experimental data presented here consist mainly of measurements of the mean temperature distribution in the penetrative flow, local surface heat flux distribution along the isothermal plate for various values of the surface temperature, and the mean velocity distribution at the interface of the two-layer environment. Because of length constraints, only a few characteristic results are presented. These bring out the basic trends which were found to be quite similar at other parametric values.

The relatively simpler circumstance of Fig. 1(a), i.e. $T_u > T_s = T_l$, may be considered first to obtain the parameters that govern the flow and heat transfer. Then, the depth of the interface from the top of the enclosure H_1, the convection velocity $U_c = [g\beta H_1 (T_u - T_l)]^{1/2}$ and the temperature difference $T_u - T_l$ may be taken as the characteristic quantities. The results can then be generalized in terms of the following dimensionless variables:

$$X = \frac{x}{H_1} \quad Y = \frac{y}{H_1} \quad \theta = \frac{T - T_l}{T_u - T_l} \quad U = \frac{u}{U_c}.$$

A nondimensionalization of the governing equations and the boundary conditions, for the flow in the upper region, leads to the dimensionless parameters

$$Gr = \frac{g\beta(T_u - T_l)H_1^3}{\nu^2} \quad Pr = \frac{\nu}{\alpha}$$

$$\theta_s = \frac{T_s - T_l}{T_u - T_l} \quad Nu = \frac{hH_1}{k}$$

where Gr is the Grashof number, Pr the fluid Prandtl number, θ_s the dimensionless surface temperature and Nu the local Nusselt number. Additional parameters, such as H/H_1, l/H_1, and W/H_1, where W is the width of the enclosure, also arise in this and other flow configurations. Thus, the Nusselt Nu can be given in terms of the location X and the dimensionless surface temperature θ_s, for given fluid and thermal stratification. Similarly, the temperature distributions can be presented in terms of the dimensionless coordinates, for given stratification and θ_s.

3.1. *Stratified environment without penetrative flow*

Figure 3 shows the vertical temperature distributions measured inside the enclosure for three

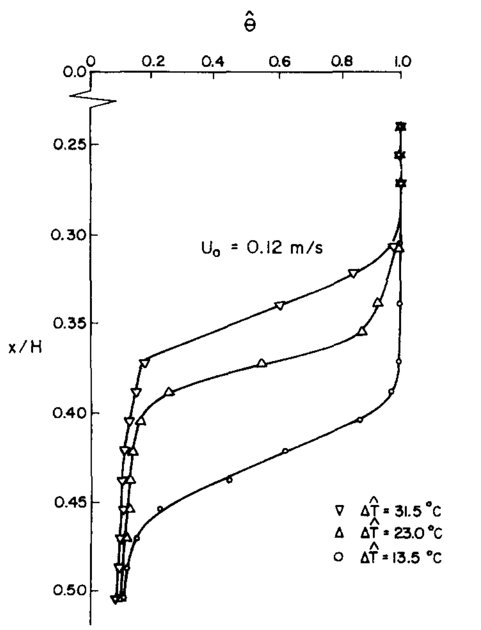

Fig. 3. Effect of the discharge temperature difference $\Delta\hat{T} = T_o - T_\infty$ on the stratification in the enclosure at $U_o = 0.12$ m s^{-1}.

values of $\Delta\hat{T} = T_o - T_\infty$ at $U_o = 0.12$ m s^{-1}, where T_o and U_o are the average discharge temperature and velocity as shown in Fig. 2 and T_∞ is the ambient temperature outside the enclosure. As can be seen from this figure, the temperature is almost uniform in the upper portion of the enclosure. This is followed by a very sharp decrease in temperature. Then the temperature decreases gradually with depth and finally becomes equal to the ambient temperature T_∞ at the bottom of the enclosure. This temperature distribution implies that an upper hot layer of essentially uniform temperature overlies a relatively cooler lower layer, which is also close to isothermal. Here, the dimensionless temperature $\hat{\theta} = (T - T_\infty)/(T_o - T_\infty)$ so that $\hat{\theta}$ is 1.0 at the discharge and 0.0 in the ambient medium, and the vertical distance x is nondimensionalized by the tank height H.

The measured distribution may be replaced by an idealized one with a step change at the interface and essentially uniform temperatures, T_u and T_l, in the two layers. The upper layer temperature T_u was found to be close to T_o in all the experiments carried out. Similarly, the measured lower layer temperature T_l was found to be very close to T_∞. The interface height is obtained from this idealized two-layer distribution and is denoted by H_1, as shown in Fig. 2. Other temperature distributions are obtained by varying the discharge conditions. For further details, see Kapoor and Jaluria [23]. The temperature distribution remains essentially constant with time and, thus, could be maintained indefinitely. Detailed velocity measurements were taken in the upper hot layer of the enclosure by a constant temperature hot-wire anemometer. It was found that these velocities were of order 1.0 cm s^{-1} or less, except near the inlet. The presence of such low velocity levels indicates that the upper hot layer is fairly stagnant and these velocities may be neglected in comparison with the velocities of the flows generated in the penetrative convection experiment. The measured velocities in the flow under consideration ranged from about 10 to 30 cm s^{-1}. Other stratification conditions were also employed in this work, though only one typical case is shown here in Fig. 3 for brevity.

3.2. *Penetration of wall flow across the interface*

Figure 4 shows the mean temperature profiles near the isothermal plate, measured at vertical distances $x/H_1 = 0.68$, 0.87 and 1.05 from the top of the enclosure. The interface is at $H_1 = 0.52$ m in this case. Here, T_s is less than T_u and T_l, though very close to the latter. Therefore, a downward flow is generated in the upper layer and this flow penetrates into the lower layer where it becomes negatively buoyant. The temperature profiles indicate the downward boundary layer flow as well as the flow that returns to the upper region due to opposing buoyancy effects in the lower layer. It is seen from this figure that, as expected, the boundary layer thickness increases as the flow proceeds down the plate. The local heat transfer rate

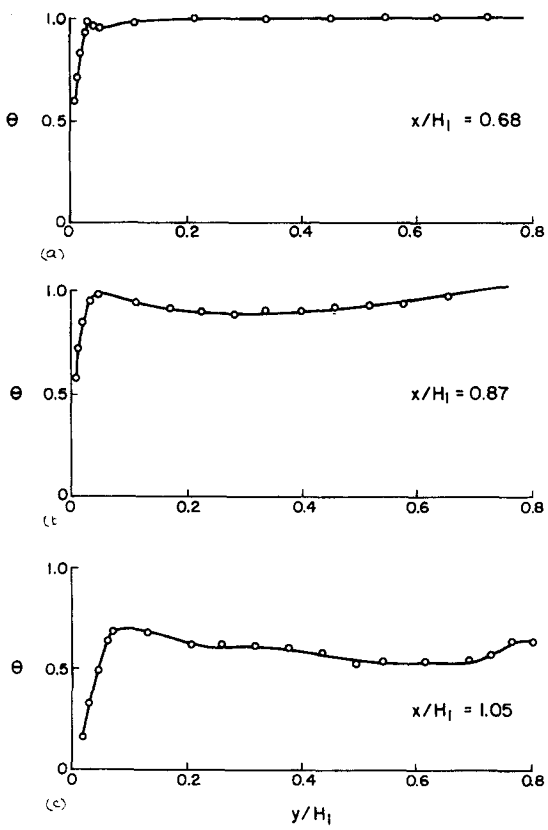

Fig. 4. Measured temperature profiles in the penetrating wall flow at $T_u = 57°C$, $T_s = 22.7°C$ and $T_l = 24.6°C$ ($\theta_s = -0.059$). (a) $x/H_l = 0.68$; (b) $x/H_l = 0.87$; (c) $x/H_l = 1.05$.

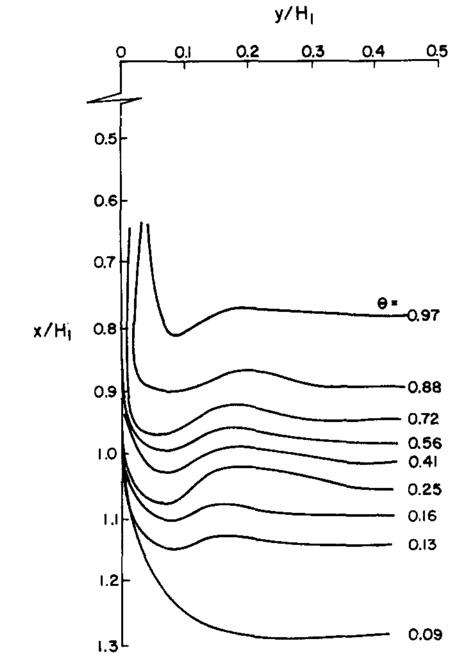

Fig. 5. Isotherms for the buoyancy induced flow adjacent to an isothermal plate in a stably stratified two-layer environment, with $T_u = 59.0°C$, $T_s = 25.2°C$ and $T_l = 27.0°C$ ($\theta_s = -0.056$).

to the plate was also obtained by employing the local temperature gradient near the surface from the measured profiles. The measurements of the surface heat transfer rate using the heat flux sensors and the value obtained from the temperature distributions were found to be in good agreement. It is seen from the figure under consideration that the fluid temperature increases as one moves away from the surface, reaches a maximum value and then decreases again. This indicates that, as expected, the penetrative flow buoys back upward due to opposing buoyancy effects in the lower region. The results are presented in terms of the nondimensional temperature θ and the horizontal distance y, normalized by the interface height H_l. This height is crucial in determining the velocity level and the flow rate of the penetrating flow and is, therefore, used for normalization. Though not shown here, the measured temperature distributions in the wall flow above the interface were found to be close to those predicted by analysis for natural convection transport from an isothermal surface.

Similar detailed temperature measurements were carried out near the isothermal plate for several other surface temperatures. The isotherms obtained for three wall and ambient temperature conditions are shown in Figs. 5–7. Figure 5 shows the isotherms for the case when the plate temperature T_s is slightly lower than the lower layer temperature T_l, but much lower than T_u, as considered earlier in Fig. 4. It is seen that the convective flow penetrates into the lower region, becomes negatively buoyant, rises toward the interface and then flows out of the enclosure at this level. The penetration depth $\delta_p/H_l \approx 0.3$ in this case, with x/H_l at the interface being 1.0. Figure 6 shows the

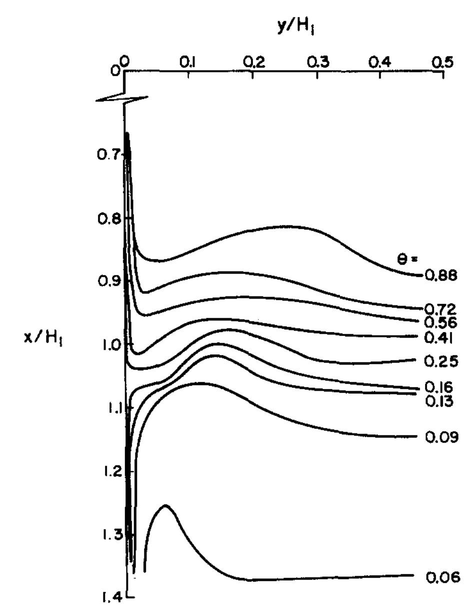

Fig. 6. Isotherms for buoyancy induced flow adjacent to an isothermal plate in a stable stratified two-layer environment, with $T_u = 57.0°C$, $T_s = 32.5°C$ and $T_l = 27.0°C$ ($\theta_s = 0.183$).

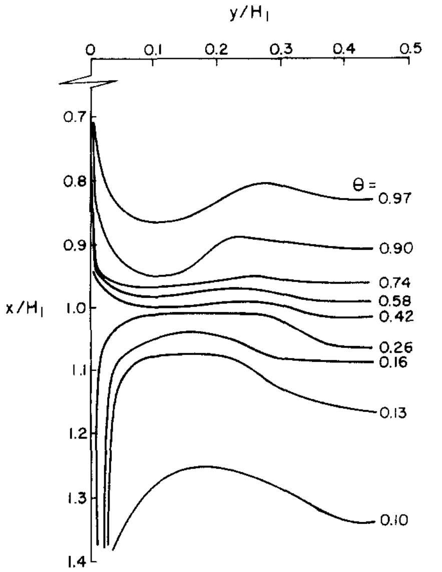

Fig. 7. Isotherms for the buoyancy induced flow adjacent to an isothermal plate in a stable stratified two-layer environment, with $T_u = 58.5°C$, $T_s = 40.0°C$ and $T_l = 27.0°C$ ($\theta_s = 0.413$).

isotherms for the case when the isothermal plate temperature T_s is kept slightly higher than the lower layer temperature. As expected, two boundary layer flows, one downward in the upper zone and the other upward in the lower region, develop and meet near the interface. This results in transport between the two layers as the two opposing flows merge and flow out almost horizontally. Figure 7 again shows the situation when the plate temperature T_s lies between the upper layer temperature T_u and the lower region temperature T_l, which is close to T_∞. However, $T_s - T_l$ in this case is much larger than that for Fig. 6. The trends are similar to those observed in Fig. 6. It is seen from this figure that the lower region boundary layer flow is much more prominent in this case than that of Fig. 6. This is due to the larger temperature difference between the plate and the immediate environment in the lower region. Therefore, these three figures indicate the basic characteristics of the penetrating, buoyancy-induced wall flows in such thermally stratified environments.

3.3. *Convective transport at the interface*

The penetration of the convective wall flows across the interface affects the transport between the upper hot layer and the lower cooler layer. It is important to estimate the mass flow rate which penetrates the interface downward, as well as the mass flow rate which rises upward across the interface. To estimate these mass flow rates, it is necessary to measure the distribution of the vertical velocity component at the interface. The velocity distribution was obtained at the interface, as shown in Fig. 8. The vertical velocity u is normalized by the convection velocity U_c, where $U_c = [g\beta H_l(T_u - T_l)]^{1/2} = [g\beta H_l \Delta T]^{1/2}$, as defined earlier. It is seen that the downward velocity reaches a local maximum at some distance away from the surface and then gradually decreases, becoming zero and then negative, i.e. upward. The negative, upward, velocity keeps on increasing in magnitude up to a certain distance and then starts decreasing gradually to finally become zero. This figure, therefore, shows quantitatively the downward and the upward penetrating flows for this circumstance where T_s is smaller than both T_u and T_l. These flow characteristics were also obtained for other temperature levels.

The corresponding temperature distribution at the interface is shown in Fig. 9. As can be seen here, the temperature gradually decreases as one moves away from the plate and finally becomes equal to that of the surroundings. Such a variation is expected from the diffusion of thermal energy. By integrating the product of the local velocity and the density, as obtained from the measured temperature distribution, over the corresponding area, the mass flow rate can be obtained. For the circumstance shown, the ratio of the upward to the downward mass flow rate is found

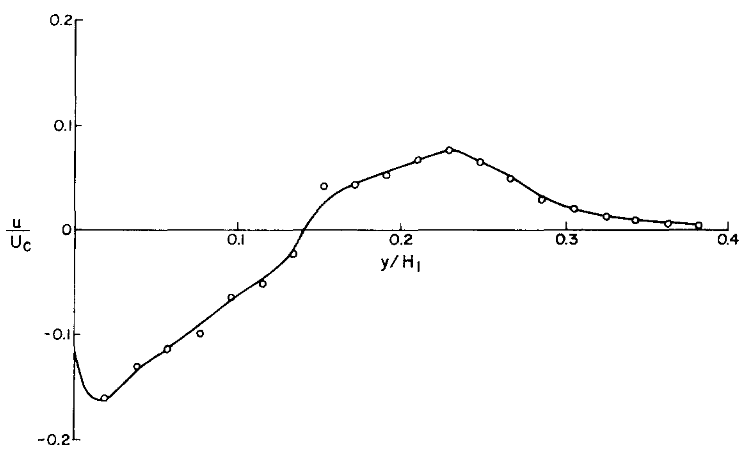

Fig. 8. Measured vertical velocity distribution at the interface of the two-layer environment, with $T_u = 59.0°C$, $T_s = 16.6°C$ and $T_l = 26.0°C$ ($\theta_s = -0.285$).

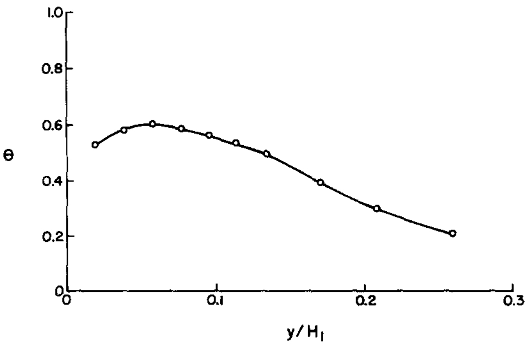

Fig. 9. Temperature distribution at the interface with $T_u = 59.0°C$, $T_s = 16.6°C$ and $T_l = 26.0°C$ ($\theta_s = -0.285$).

to be 1.64. For the parametric ranges considered here, this ratio was found to vary between 1.5 and 1.75, indicating the level of fluid entrainment into the penetrating flow. It is also interesting to compare the measured penetration depth δ_p/H_l of around 0.3 for this case with that obtained from an equivalent negatively buoyant wall jet, with the same momentum and thermal buoyancy. The penetration depth for a wall jet with opposing buoyancy is given by the correlation [19]

$$\frac{\delta_p}{D} = 4.424\, Ri^{-0.389}$$

where the Richardson number Ri is based on the inlet conditions of the jet. Using an equivalent wall jet for the downward penetrating flow, the corresponding penetration depth δ_p/H_l is obtained as 0.28, indicating that the earlier wall jet results may be used to estimate the penetration distance.

3.4. *Heat transfer to the isothermal surface*

The local surface heat flux distribution along the isothermal plate was also measured. Figure 10 shows the variation of the measured local heat transfer flux, in terms of the local Nusselt number Nu, along the isothermal plate for three wall temperatures $\theta_s = -0.03$, 0.32 and 0.65. Here, the Nusselt number Nu is defined as $Nu = hH_l/k$, where k is the thermal conductivity of the fluid and the heat transfer coefficient $h = q/(T_u - T_l)$, q being the local heat transfer flux to the plate. As mentioned earlier, Nu may be expressed as $Nu(x/H_l, \theta_s)$ for given fluid and thermal stratification. It is seen that the local heat transfer rate decreases gradually in the upper zone followed by a sharp drop across the interface. It then remains essentially constant over the remainder of the plate. For a surface temperature of $\theta_s = -0.03$, the local heat transfer to the plate does not become zero, even after the convective flow changes its direction. This is due to the fact that the lower layer temperature T_l is slightly higher than the plate temperature T_s and heat transfer to the plate from the ambient medium occurs in the lower region. The results for plate temperature θ_s values of 0.32 and 0.65 are similar to those obtained for a plate temperature θ_s of -0.03, except that the heat transfer becomes negative in the lower region in these cases, implying heat loss from the plate to the ambient medium in the lower region. The heat flux measurements indicate that the wall gains energy due to the buoyancy induced boundary-layer flow up to the interface. The flow stagnates and reverses direction just beyond the interface. Depending on whether T_s is greater or less than T_l, the plate loses energy to or gains energy from the ambient medium. Therefore, these measurements support the findings discussed earlier on the basis of the velocity and temperature data.

CONCLUSIONS

An experimental study has been carried out to study the basic characteristics of a natural convection wall

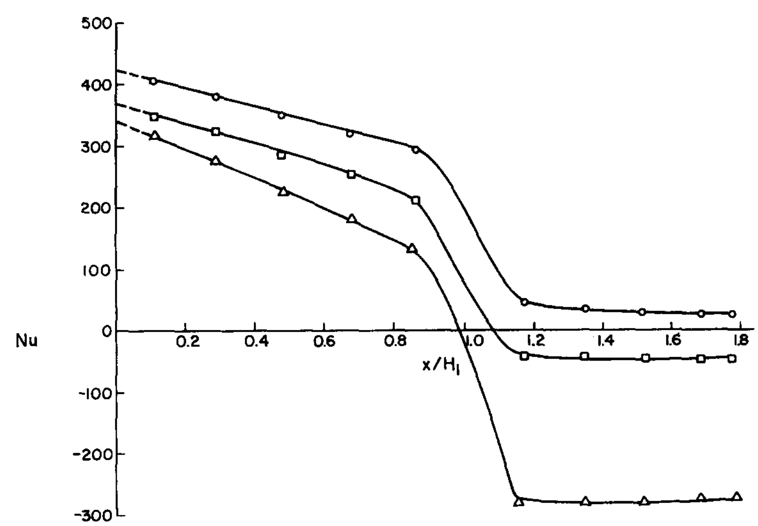

Fig. 10. Variation of the local heat transfer flux in terms of the Nusselt number Nu along the isothermal plate. ○, $\theta_s = -0.03$; □, $\theta_s = 0.32$; △, $\theta_s = 0.65$.

flow penetrating in a thermally stable environment. Such flows are important in many practical problems, such as thermal discharge into the environment, thermal energy storage systems and enclosure fires. An isothermal vertical surface was immersed in a thermally stable, two-layer, medium in which an essentially isothermal heated layer overlies a relatively cooler isothermal layer. Detailed measurements of the thermal field have been taken near the isothermal wall for several values of the surface temperature excess over the lower layer temperature. The upper layer is taken at a temperature higher than the wall temperature. Isotherms were obtained for each case. It was found that the downward natural convective flow generated in the upper layer penetrates into the lower region, buoys back due to the upward buoyancy force and rises toward the interface, finally flowing out of the enclosure at this level. When the isothermal plate temperature is kept between the upper and lower temperatures, two boundary layer flows, one downward in the upper zone and the other upward in the lower zone, develop and meet near the interface. Several interesting aspects in the transport between the two layers are observed.

The mass flow rate which penetrates downward across the interface and that which rises upward across the interface may be estimated from the corresponding vertical velocity and temperature profiles measured at the interface. These are discussed in terms of the conditions generating the wall boundary layer flows. It is found that the penetrating flow entrains about half its mass flow rate from the ambient fluid as it turns upward. The heat transfer flux to and from the surface is measured along the height of the isothermal plate. It was found that the local heat transfer rate decreases gradually in the upper hot layer, as expected for a boundary layer flow, followed by a sharp drop across the interface. It then remains essentially constant over the remaining portion of the plate, with the Nusselt number higher at smaller surface temperatures.

The basic characteristics of this natural convection flow, in terms of entrainment, heat transfer rate and penetration distance, are of particular interest in wall flows generated in room fires. The walls are often isothermal because of their slow response, while the fluid in the enclosure becomes stably stratified. It is important to determine the penetration characteristics of the natural convection flows that arise. Of particular interest are the basic nature of the transport processes and the penetration distances obtained. The results available in the literature on wall jets with opposing buoyancy may be used to estimate the penetration depth, employing an equivalent wall jet. Other quantitative results are obtained on this penetrative flow circumstance to provide inputs for the development of an analytical model for this flow.

Acknowledgements—This research was carried out with support from the Building and Fire Research Laboratory of the National Institute of Standards and Technology, United States Department of Commerce, grant no. 60NANB1D1171. The helpful suggestions of Dr L. Y. Cooper during the course of this work are also acknowledged.

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